THE FLOW OF GASES IN FURNACES

BY

W. E. GROUME-GRJIMAILO

Professor of Iron Manufacturing at the Polytechnic Institute of Petrograd

Translated from Russian into French
By LEON DLOUGATCH AND A. ROTHSTEIN

With a Preface
By HENRY LE CHATELIER

Translated from the French
By A. D. WILLIAMS

With an Appendix upon
THE DESIGN OF OPEN-HEARTH FURNACES

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TO THE MEMORY OF A. ROTHSTEIN

MONSIEUR ROTHSTEIN, A RUSSIAN SUBJECT AND A GRADUATE OF L'ÉCOLE DES PONTS ET CHAUSSÉES, ENLISTED VOLUNTARILY AS A PRIVATE IN THE FRENCH ARMY AT THE OUTBREAK OF THE WAR

HE WAS PROMOTED UPON THE FIELD OF BATTLE TO THE RANK OF SOUS-LIEUTENANT OF ENGINEERS AND WAS DECORATED WITH THE CROSS OF A CHEVALIER OF THE LEGION OF HONOR. HE ALSO RECEIVED THE CROIX DE GUERRE, WITH A NUMBER OF CITATIONS BEFORE AN ARMY

HE WAS KILLED ON THE 1ST OF AUGUST, 1916, AT FLEURY, BY A SHOT THROUGH THE FOREHEAD, WHILE HE WAS LEADING AN ASSAULT UPON A SMALL FORT [illegible] [illegible] UPON THE [illegible] ROAD. HE HAD REQUESTED THE HONOR OF LEADING THIS ATTACK.

PREFACE BY TRANSLATOR INTO ENGLISH

More or less mystery has enveloped the designing and construction of metallurgical furnaces. Blue-prints with the titles removed and tables of supposedly important features have been carefully treasured. When new furnaces were to be designed an extensive analysis was frequently made of what was known of the work of others, and an endeavor was made to "improve" upon them. During construction other plants were visited and further improvements were frequently made, with the result that the finished furnace departed widely from the drawings. Not infrequently the starting up of a new furnace was signalized by an acrid meeting of the lodge of sorrow and by expensive reconstruction and loss of product. During the campaign of the furnace the question, "What can be done to change it?" was a live issue. Certain parts of the furnace had to be rebuilt from time to time, and this afforded a certain amount of opportunity to correct early errors, provided they had been rightly diagnosed.

Some years ago, the writer became convinced that the flow of gases in a furnace was entirely devoid of mystery, being governed by the laws of elementary physics. Several furnace designers were asked, as the occasion offered, whether they based their design upon experience and upon the proportions of working furnaces or used formulas for the flow of the gases. The replies received were interestingly noncommittal. The physical laws of flowing fluids, however, were found to agree very closely with practical observations. They explained, with remarkable clarity, some mysterious troubles.

Later, the writer got in touch with the work of Professors Groume-Grjimailo and Yesmann. Professor Groume-Grjimailo had gone over the ground very thoroughly several years before the writer became interested in furnace work, and had conducted numerous experiments with model furnaces. He had simulated the flow of the gases and the effects of the atmosphere by immersing his models in water and circulating colored kerosene oil through them, the models being built with glass sides to permit observation.

A great many illustrations and drawings of furnaces, with more or less detail, have been printed from time to time. Attention was occasionally called to certain differences, generally to an increase in the size of certain parts above those of some well-known furnace. The reasons for changes not being stated, it was possible to make many inferences.

While the title of this work, "The Flow of Gases in Furnaces," explains its purpose, it is an abbreviation of the title of the French edition, which may be rendered into English as "An Essay upon a Theory concerning Hot Gases in Furnaces based upon the Laws of Hydraulics." The work treats of the development of the flow laws of heated gases and the application of those laws to the rational design of furnaces. Primarily, a furnace, considered as an elementary structure, is merely a hollow structure of refractory material, within which heat may be released. A great many furnaces show that their designers had very elementary ideas concerning the application and utilization of heat. *Damour* has given the following definition of industrial heating, which sums up the question:

"Industrial heating has for its objective *the realization of a temperature more or less high, with economy, within an enclosure of known dimensions, making use of a selected combustible (or form of heat energy), for effecting a certain operation or chemical reaction.*

"High temperature, economy, various combustibles, fixed dimensions of the furnace or its heating chamber, a certain industrial operation—such are the five variables of the problem. Industrial heating operations present an infinite variety of results to be accomplished. The engineer must be able to see all sides of the problem; he must understand the various furnaces required, their proper operation, the correction of their defects, and above all the rational solution of each particular case. Each phase of the question calls for many different forms of knowledge, with each of which the engineer must be equally familiar; but it is essential that he carefully distinguish between them, lest he become confused or fall into those errors which frequently retard the progress of the science of furnaces."

In a preceding paragraph *Damour* states: "All forms of energy may be converted into heat, either directly by a single apparatus or indirectly by the successive use of two forms of apparatus. The only difficulty consists in obtaining this heat in such form, or at a thermal potential (*temperature*), which will permit its utilization, or in finding the methods of recuperation[1] which will avoid the waste of the energy."

According to these conditions the problem may be stated as follows: "*To find the most advantageous means of transforming energy under any of its various forms, chemical, thermal, electrical, mechanical, etc., either singly or in combination, within the chamber of a furnace, into heat, utilizable for effecting a particular industrial operation.*"

All industrial heating operations fall within four principal classes:

"*(1) Those in which the temperature at which the particular reaction occurs is sufficiently low, and the chemical energy of the reaction is sufficient, to release enough heat for the propagation of the reaction and to more than cover all radiation and other cooling effects.*"

This case covers the burning of sulphur to sulphur dioxide, the combination of nitrogen with calcium carbide in the manufacture of cyanamide, the making of steel in the Bessemer converter.

"*(2) Those industrial operations in which a certain amount of heat is released, but whose heat energy is not sufficient to maintain the ruling temperature necessary for the reaction.*"

This case is the most general one: Nearly all operations release some heat. A characteristic example in this class is the open-hearth furnace.

"*(3) The operation possesses no special chemical energy or is inert.*"

This case comprises many reheating and some melting operations.

"*(4) The chemical energy peculiar to the operation is negative.*"

All the heat must be supplied from an external source.

In translating this work, no effort has been made to transform the formulas and tables from the metric to the English system of units. Such a transformation would introduce many complicated constants into the formulas and, in addition, would greatly increase the numerical work required in computation. Moreover,

[1] *Note by translator.*—As used here, this does not necessarily mean regeneration, but the economic utilization of the heat by direct or indirect recovery of waste heat.

the Continental practice of stating metric values has been followed, as it obviates the use of the decimal point. The use of the metric system in work of this character is practically compulsory and practically all of the basic data are available in metric units. The results, if desired, may readily be transformed into English units, the transformation factors, in many cases, being simple and easily memorized numbers.

The translator holds no brief for the metric system. Its logical and simple numerical constants supply a most potent argument for its use in all technical work. It reduces many of the problems of engineering from the tedious computations of the English system to exceedingly simple mental computations.

It is unfortunate that German methods are much more widely known than the extremely simple methods used in France, as a result of German propaganda which has been spread over the world and accepted without due investigation. The Germans have claimed as their own many technical discoveries for which credit is really due to men of other nations.

The purpose of this translation is to convey the ideas of the original author. In order to enhance the usefulness of the work, some additions have been made in the shape of appendices supplying methods not elsewhere available in English.

PREFACE FOR THE ENGLISH EDITION

THIS work was originally published in the Journal of the Russian Metallurgical Society, in 1911.

During the past six years, a large number of furnaces have been designed and placed in operation by my pupils, by the technical bureau working under my direction, and by myself; but I do not know of a single instance in which we have met with a failure.

The hydraulic theory of the flow of hot gases has proved to be correct. This method of establishing the design of furnaces by computations has given good results and may be followed without fear.

This is not all; the new idea, that of examining each furnace as a hydraulic recipient or reservoir, has exercised a deep influence upon the working out of new forms of furnaces. During the last six years I have worked out a number of new forms of furnaces, the greater proportion of which have already been controlled by working experience. There is no doubt that we are on the eve of a radical change in the technical utilization of the heat generated by the combustion of fuel.

When the war is over, I hope to publish my work in regard to the establishment of the design of new types of furnaces. In the mean time, I limit myself to the following remarks:

During the last few years, it has happened more than once that a furnace construction thought out by me, solving a problem placed before me by a client, has proved, after some research, to have been described in an American patent. This tends to show that the technique of the United States has approached very close to the solution of many of the problems connected with the design of furnaces. With a little more theory, the American engineers will fully master the science of building furnaces. I am, therefore, extremely obliged to Mr. A. D. Williams for his offer to translate this work. I am convinced that it will prove very useful to our esteemed allies, the Americans.

W.-E. GROUME-GRIMAILO.

PETROGRAD, May, 1917.

PREFACE TO THE FRENCH EDITION

The researches of Professor Groume-Grjimailo bring to us a new idea. Conceptions of this character are very rarely presented to us in the numerous publications of the day, and we sincerely compliment the author in that he has opened the way for us into a very broad field which merits our careful consideration. He has set forth a principle in regard to the circulation of the hot gases within furnaces—a very simple principle, but one that has not heretofore been recognized. We have always considered that gases, by reason of their absolute elasticity, completely fill the chamber in which they are enclosed. Then, by a process of unconscious induction, without any sound basis, we conclude that in circulating through a series of successive chambers or flues, they fill equally the entire cross-section of those chambers through which they pass and that their current sweeps uniformly through all the passages or flues which are open to them in proportion to the area of these passages. Perhaps we should not formulate this erroneous principle in such exact terms; nevertheless, we proceed as though we firmly believed that it was correct. And it certainly follows, notably in the construction of furnaces, that very serious errors are made. Professor Groume-Grjimailo cites numerous examples of these errors.

A phenomenon is more readily understood when it is compared with something that we have always before us. We see a stream of water flowing in its bed, resting upon the soil and bounded upon three sides by the surface of the ground. Its upper surface is separated from the atmospheric air above by a horizontal plane, the position of which is not fixed and which varies in accordance with the volume of water flowing. The hot gases in a furnace tend to circulate in exactly the same manner, with this single difference, that the plane of separation is below, and the profile of the bed of the stream is formed by the roof and the walls of the furnace. As this comparison shows, we have heretofore erroneously considered that the flame filled the entire furnace and heated it uniformly, when in reality the hot gases may circulate only in the upper portion of the heating chamber, without coming into any contact with the material which has been placed upon the hearth. The utilization of the heat in this case will be very inefficient. The stream of fire which flows against the roof will carry all of its heat to the chimney.

The reason for this lies in the high coefficient of expansion of gases, that is to say, in the fact that their density diminishes very rapidly as their temperature increases. Therefore, the hydrostatic equilibrium of the different layers of gases, which are at different temperatures, is accomplished with great rapidity, and the hottest gases tend to accumulate at the highest point in the furnace. Take, for example, the case of a kiln for bricks, having a height of 5 meters and an average temperature of 1000° and assume that the hot gas issuing from the fireboxes has, where it enters the kiln, a temperature 500° higher, or 1500°. The motive pressure caused by this difference in temperature, acting upon a column of gases 5 meters in height, tends to produce, in the column of hot gases a vertical velocity of 5 meters per second, while the average velocity of the gases, as referred to the total area of the kiln chamber, is perhaps one-fiftieth of this velocity. The gases, therefore, have ample time to arrange themselves in layers according to their relative density. If the opening for the escape of the waste gases is located at the highest point of the roof of the furnace, the greatest portion of the heat will be lost. It is necessary, in order to avoid such heat loss, to employ the downdraft principle of heating, which has been employed for a long time in the manufacture of porcelain and faïence, but which still remains unknown in a great many other industries.

By numerous examples, the author of this volume shows that at least one-half of the furnaces designed for high-temperature heating are arranged in such a manner that the currents of hot gases tend to isolate themselves in the heating chamber, thus giving an extremely poor utilization of the heat released by the combustible.

The formation of these streams of hot gases in the furnace is the cause of another difficulty; they tend to retard the completion of the reaction of combustion. Above a fire burning upon a grate, there are a number of parallel currents of gas, some of which contain an excess of oxygen, while others contain an excess of combustible gas. In order that the latter may burn it must be mixed intimately with the oxygen; this requires a certain period of time, and the mixture must be made at a temperature sufficiently high to permit the gases to react upon each other.

In steam-boiler settings, an arch of refractory materials is frequently built over the fire. But if this arch does not form a pocket in which the hot gases may accumulate and remain for a time, the unburned gases will flow too rapidly from beneath the arch, and the effect of the arch will be only imaginary. The work of Professor Groume-Grjimailo is filled with similar examples, with complete information regarding the works in which the observations were made. This work is not, therefore, purely theoretical; it is a thoroughly practical treatise based upon actual observations and experiments.

Not content with having developed these new ideas, the author has endeavored to place them before his readers in a manner which will make them absolutely clear, just as he has been doing for a long time in the instruction of his classes. In order to present in a visible manner the circulation of the hot gases within the furnace, he has employed small models of furnace sections, enclosed between two plates of glass, and within these models he has arranged to circulate two liquids of different densities, water serving as the heavier liquid and colored kerosene as the lighter liquid, representing the hot gases or flame. The localization of the current of kerosene shows very clearly whether the furnaces are of poor design or not. This method is particularly applicable for presenting this subject before those who may not be well informed regarding the technical principles of the great industries; but it has the inconvenience of giving, to a certain extent, an erroneous view of the actual phenomena. During its circulation, the colored kerosene cannot be changed into water or mixed with it, while the light or hottest gases, having given up their heat, are transformed into colder and heavier gases. This introduces very essential differences which must be clearly understood. The only purpose of this method of representation is to present to the eye a very strong impression which will stimulate the imagination; in practical application it is necessary that we consider these phenomena as they really occur and study the gas itself, which can only be conceived in the abstract and of which no visible representation can be made.

The reading and study of this small volume will present these principles to the eyes of many engineers and should lead to many improvements in the art of heating and utilizing heat. In this it will be of service; this is the highest recognition which can be given to an author.

HENRY LE CHATELIER.

NOTE.—The foregoing was written by Henry Le Chatelier for the French edition, which appeared in 1914.

FOREWORD

In his celebrated *Traité de Métallurgie* in 1875, Gruner gave this definition:

"Metallurgy is the *Art* which treats of the preparation of metals."

To-day, the application of the laws of physical chemistry, and above all the application to metallurgical processes of the principles set forth by M. Le Chatelier, have enabled us to uncover the mysteries of this art. Metallurgy has become a science. The dense fog of empiricism, which formerly enveloped all of the metallurgical processes, has been dispersed, and everything has become simple and clear, as it does in all other branches of human knowledge, when they become part of the domain of science.

It is true that we now require a much more profound theoretical preparation than was required in former times of those who took up metallurgical work. As a result, the young engineer at the termination of his studies enters the workshop with such a clear idea of its processes that after a few months of practical experience he is much better able to make himself the master of the situation than are the older students of Gruner—who acquired the metallurgical *art*—after ten years of practice and assiduous work in the shops.

The young man of our day, who is well prepared theoretically, becomes familiar with his work ten times as rapidly and works to better advantage than those who studied under the old system. Armed with scientific truths, he copes much more readily with the difficulties which arise, and he does not experience the feeling of helplessness which overcame the older metallurgists when they found it impossible to better working conditions and save the plant from a serious loss—a feeling which often paralyzed our efforts at the time when all of our energies were vitally necessary.

The book which we present to our readers is dedicated to the methods of constructing more efficient fuel-fired furnaces. Heretofore, this field has been solely the field of practical experience and of empiricism. How were we able to acquire knowledge of this veritable art? I will speak from my personal experience.

After I had finished my studies at the Institute of Mines, at Petrograd, I left the schoolroom to enter the Salda Works of *M. Démidow*, one of the best plants in the Oural. The supervision of furnace work of all kinds was entrusted to an old employee, Pierre Chicarine, a man of very great intelligence, but illiterate, unable even to sign his own name. This did not prevent him in any way from undertaking to design new furnaces of many types, and he always succeeded in obtaining good working results.

He was unable to state the principles by which he was guided, despite a sincere desire upon his part to make me understand the reasons for the various features of the designs. This, however, was beyond his power of expression. His only argument was, "It must be done in this way; otherwise, you do not get anything."

"Why?"

To this he was never able to reply.

I still remember many of his constructions, and I am persuaded that he really knew nothing of the fundamental principles, or more exactly, that he did not have any very clear comprehension of them. He made many big mistakes, which he finally corrected after many trials. Meanwhile, from time to time, in discussing the various furnaces, and in placing new furnaces in operation, he gave me much important information, calling to my attention the phenomena which took place in the furnace, teaching me not only *where to look,* but also *how to see* that which took place.

I became greatly interested in the furnaces. I studied them very closely, making great efforts to solve their mysteries; but I did not make very rapid progress in so doing. It was only after ten years that I commenced to make suggestions to my instructor for his criticism. It was only after fifteen years that I ventured to design a new furnace for myself. My master reported after some days upon my design; fully half of it had been erased and redrawn. "It is necessary to do it in this manner; otherwise you will not be able to make it work." Thereupon my dignity as director of the works—the position which I occupied at this time—was deeply wounded; at the same time, I recognized that the veteran had reasons for acting in this manner.

After about twenty years I was able to solve the mysteries of furnace construction and discovered how to make furnaces which would work properly. I finally comprehended that each furnace could be represented as a hydraulic recipient or reservoir, that the problem of designing a furnace was nothing more than a problem in hydraulics, and that the circulation of the hot gases in the furnace was similar to the circulation of a light liquid within a heavier liquid.

Having arrived at this knowledge of furnaces, I found much pleasure in conveying it to my co-workers. I was thus confirmed in my belief that the hydraulic theory was clear, and presented to the human mind an exact and precise view of the problem, a true point of attack. The time necessary to acquaint a young engineer with the principles of furnace design was reduced to one month. An old employee who worked in the rolling mill and was well posted in regard to shop practice was able to understand the science of furnaces in a single evening, and I am satisfied that the subject was thoroughly understood by him.

Nevertheless my problem was not completely solved; understanding the mechanism of the circulation of the hot gases was not sufficient. It still remained to establish the mathematical formulas for those laws, in order to compute the dimensions of the furnace. M. J. Yesmann, Professor of Hydraulics at the Polytechnic Institute of Petrograd, came to my assistance at this point. Professor Yesmann's formula for the inverted dam or weir[(1)] has been verified by me by its application to twenty existing furnaces. Having convinced myself that it was correct, I inserted the computation of reverberatory furnaces into the metallurgical course of the Polytechnic Institute of Petrograd.

It is true that the problem has not been entirely solved. A large amount of research work must be done in order to determine the coefficients which are necessary to reduce these formulas to practice. I understand perfectly that my work is nothing more than the first step toward the solution of the problems of furnaces. It has caused and, I hope, will cause many others to undertake the tests which are required to complete the work.

For example, one objection which has been made is that I have made an error in assuming that a current of flame is similar to a current of incandescent gas. It is perfectly true that this is an

(1) Refer to pages 40 and 53.

error; but it is impossible to do everything at the same time and in a single step. It was necessary to establish the fundamental idea firmly, and in doing this the minor details were temporarily neglected. When it has been firmly established that the circulation of the hot gases in a fürnace is a problem in hydraulics, when the fog which has obscured these problems and appeared to make them insoluble has been cleared away, then it will become possible to settle the various details in a suitable manner.

The complete working out of the laws governing the theory of the circulation of the hot gases, which has been commenced by Professor Yesmann, should be continued by experts in hydraulics. It is our province, as men of deeds, to check the working methods of existing furnaces and to deduce therefrom the practical rules which it is necessary for us to know in determining their design.

I am at present engaged in this work, as are many of my students who are constructing furnaces in numerous workshops in Russia. When sufficient progress has been made, it will be possible to give the methods for the design computations of all types of furnaces.

The very simplicity of the conclusions to which I was led were rather disconcerting. But their approval by my colleagues at the Polytechnic Institute—M. Kirpitchow, Professor of Applied Mechanics, and M. Mechtchersky, Professor of Theoretical Mechanics—encouraged me to publish the original of the present volume.

It was only natural that I should desire to submit my work to a high authority, such as M. Le Chatelier; the translation of the work into French afforded such an opportunity, and I am greatly pleased by the honor which he has conferred upon me in introducing me, in such flattering terms, to my new readers.

W.-E. Groume-Grjimailo.

Petrograd,
February, 1914.

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PART I

THE APPLICATION OF THE LAWS OF HYDRAULICS TO THE COMPUTATIONSF OR A REVERBERATORY FURNACE

I. THE FUNDAMENTAL VIEWPOINT

HERETOFORE reverberatory furnaces have been constructed with very little, or without any, computation; and the designer has always met with more or less difficulty in establishing the lines for a new furnace or in improving an old furnace which was defective, and forcing it to work in the right manner. This was due to the fact that the technical considerations governing the design of these furnaces had not advanced sufficiently to give any very exact point of view regarding the mechanism of the circulation of the gases within the furnace. If the viewpoint which is proposed below is accepted, the working processes of these furnaces will become completely clear:

We are surrounded, as is everything else amidst which we live, by a liquid [(1)] ocean, the air; the reverberatory furnace, which is under consideration, is also immersed in this ocean of air. As the density of this liquid, the air, is 770 times less than that of water, its presence is hardly noticeable and is very rarely considered. At the same time, it must be admitted that we do not

[(1)] *Note by the French translators.*—While the word " fluid " would be more appropriate, the word " liquid " has been retained as conveying more exactly the thoughts of the author.

live and exist in a vacuum. It is this neglect of the part played by the air which causes error; it is this which prevents a clear conception of the mechanism of the circulation of the hot gases within a reverberatory furnace. When the presence of the air is taken into consideration, the problem becomes very clear.

What is it that forms the flame? It is a mixture of gases at a high temperature, reacting upon each other (combustion) and releasing in this manner a sufficient amount of heat to raise the products of their combustion to incandescence. The solid particles of carbon, by their incandesence, give to the flames that especial appearance which impresses the imagination and causes the flame to be attributed to some infernal power. But, in reality, the idea which the author desires to convey in regard to the " flame " may be better understood if the flame is considered as a current of incandescent gas. This approximation is sufficiently accurate for the purpose.

The reverberatory furnace is accordingly considered as an apparatus immersed in a liquid, the air, which weighs 1 kg 29 per cubic meter, in the interior of which there circulates a current of incandescent gases, that is to say, a liquid much lighter than the air.

It is known that the coefficient of expansion of gases is $\frac{1}{273}$; if, therefore, the specific weight of air at 0° is considered as unity:

at 273° its specific weight will be	$\frac{1}{2}$
546°	$\frac{1}{3}$
819°	$\frac{1}{4}$
1092°	$\frac{1}{5}$
1365°	$\frac{1}{6}$
1638°	$\frac{1}{7}$
1911°	$\frac{1}{8}$

and as the specific weight of air at 0° is 1 kg 29:

at 273° its specific weight will be	0 kg	645
546°	0	430
819°	0	323
1092°	0	258
1365°	0	215
1638°	0	184
1911°	0	161

A very clear idea of the differences in density which are caused by great differences in temperature may be obtained by a consideration of the air (gases) in an open-hearth furnace, of which the temperature is in the neighborhood of 1638° while that of the air is 0°. If it is assumed that the density of the gases in the furnace is equal to water, the density of the air at 0° will be relatively equal to that of molten iron.

A furnace in its regular working condition may be considered as being immersed in a glass tank filled with water, the heavy liquid, while the interior of the furnace is traversed by a lighter liquid; the action of the flame within the furnace may in this manner be considered as similar to that of the lighter liquid flowing within the heavier liquid.

A complete representation of the circulation of the flame or hot gases within a furnace may be made in the following manner;

A model to scale of the longitudinal section of a reverberatory furnace is constructed and immersed in a tank with glass sides; if a stream of a lighter liquid, as for instance, kerosene, is now passed through the model of the furnace, the movement of this liquid will reproduce exactly the movement of the flames within the furnace.

II. EXPERIMENTS WHICH SERVE TO SHOW THE ANALOGY BETWEEN THE CIRCULATION OF THE FLAME AND THE MOVEMENTS OF A LIGHT LIQUID WITHIN A HEAVY LIQUID.[1]

A white metal model reproducing to scale a brick kiln is placed between two sheets of glass and submerged in a glass tank; by means of pipes a stream of colored kerosene is passed into the model[2] through the firebox from which the gases of combustion enter the furnace.

[1] The photographs for Figs. 1, 2, 4, 10, 19, 24, 29, 30, 31, 32, 116, and 117 were supplied by the *Société russe de métallurgie*, whom the author desires to thank for the same.

[2] The illustration (Fig. 2) shows the general arrangement of the apparatus which has been used in conducting the experiments made before the classes at the Polytechnic Institute of Petrograd. The model is a scale reproduction of a brick kiln of the Motovillikha works, the drawing of which is shown in Fig. 3. This is submerged in a tank filled with water. Tubes with control valves serve to introduce streams of colored kerosene through the fireboxes of the kiln, the kerosene flowing from a large bottle which acts as a high-level reservoir. The kerosene, having passed through the furnace, rises to

First Experiment.—Study of the older, or updraft, type of brick kiln (Fig. 1), which has the opening for the escape of the waste gases at the highest part of the arched roof.

Small streams of colored kerosene are introduced through the fireboxes and flow up to the central orifice, which is wide open. The streams of kerosene may be seen as fine threads flowing up close to the walls of the kiln and are not of sufficient volume to fill the kiln chamber. Increasing the flow of the kerosene, or, as it may be expressed, firing the kiln more heavily, does not affect the result. It is very clear that the burning of the brick in a

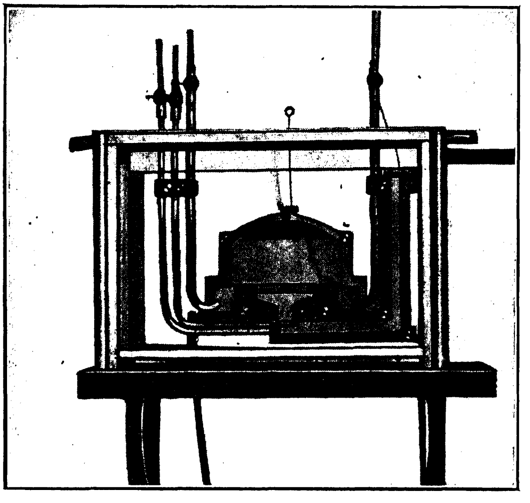


FIG. 1.

kiln working in this manner will be defective. The brick in the lower portion of the kiln will be soft and only partially burned. In order to improve this condition it will be necessary partially to stop up or close the smoke hole. The poor working conditions which exist in the updraft brick kiln are shown in Fig. 1. When the smoke hole is partially closed, the kerosene is forced to accumulate in the upper portion of the kiln; it fills more and more of the

the surface of the water; thence by a trough it flows to the large bottle below. A small pump driven by a motor of $\frac{1}{40}$ hp draws the kerosene from the lower bottle and delivers it to the upper bottle, enabling the kerosene to circulate by gravity as long as desired.

kiln chamber, until an equilibrium is established between its inflow and its outflow. When this equilibrium is established, the lower surface of the layer of colored kerosene assumes a permanent level.

It can be seen in Fig. 2 that only the upper portion of the

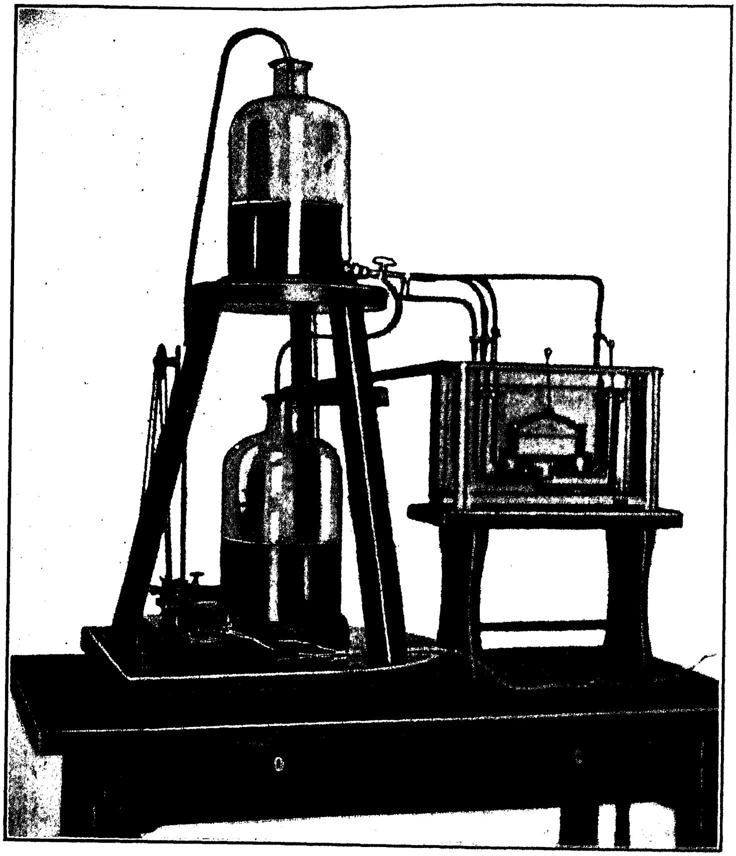


Fig. 2.

kiln, above its mid-height, is filled with hot gases, and that the space between the lower surface of the layer of kerosene and the hearth of the kiln contains none at all. This space, therefore, will only be heated by such eddy currents as form, and these are due entirely to the differences in density which exist between the flame and the cooler gases with which they come in contact. Immediately below the roof of the kiln the hot gases are relatively at rest. The burning of bricks does not permit of sudden or quick changes of temperature, as bricks will crack and spawl if subjected to such changes; therefore it would not be advantageous to use a kiln which worked in this manner.

By still further obstructing the smoke hole of the kiln, the lower surface of the layer of hot gases can be driven downward

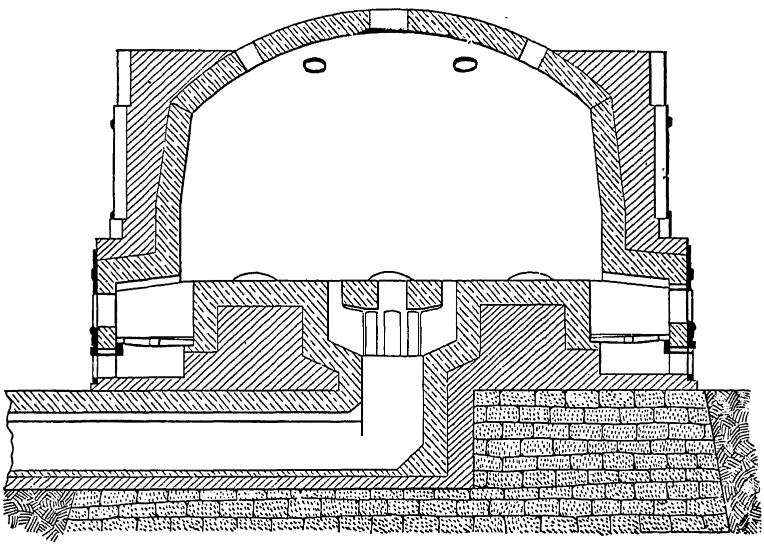


FIG. 3.

until it comes into contact with the sole of the kiln, thus giving the maximum efficiency which is possible with kilns working on the updraft system. Under these conditions, the brick can be burned to an extent which is fairly satisfactory, but they will not be of extra good quality, because, in spite of all, the currents of hot gases have a tendency to flow directly to the highest opening. The equalization of the temperature will be affected very little by the eddy currents which will be formed. These eddy currents will be of very slight intensity when the hot gases in the kiln are as shown in Fig. 1, but nevertheless they do not disappear entirely while the temperature at the sole is less than that which exists immediately below the roof of the kiln.

Note by translator.—A considerable portion of the equalization of the heat in the updraft kiln is due to conduction of the heat, through the kiln structure and the brick set in the kiln. This heat is carried downward in this manner and imparted to the cooler layers of gases at the bottom of the kiln, heating them and promoting eddy currents. The updraft kiln, however, heats very slowly at the bottom, and the upper portion of the setting will be overburned while the lower portion is underburned. As compared with the downdraft kiln, the updraft kiln consumes a larger amount of fuel per unit of output and requires a longer time to complete a kiln round.

Direct or updraft brick kilns were the only ones built up to about twenty years ago. They are still found in many potteries,

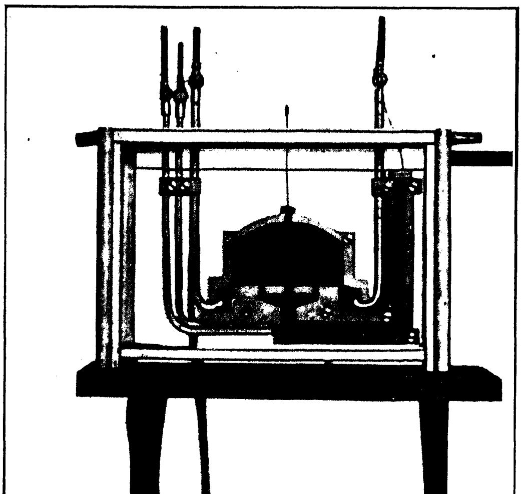


Fig. 4.

as in the works of Korniloff Brothers, at Petrograd. A brick kiln of this type is still in use at the Oboukoff works, and it is only a short time since one was in use at the Poutiloff works. These are not the last updraft kilns in use, but most of them have been replaced by *downdraft* kilns.

Second Experiment.—Continued study of updraft kiln. Method of operating downdraft kiln. For this experiment the smoke hole at the top of the kiln is completely closed. The chimney of the apparatus, as shown in Fig. 4, is filled with kerosene, which is also introduced as before, through the fireboxes. It follows that when the damper or stopper at the top of the chimney is opened slightly, the model will represent a downdraft brick kiln in operation.

It is seen that in the downdraft kiln the hottest gases rise to the highest point under the roof, where they accumulate, forcing the cold gases to the chimney through the ports in the hearth of the kiln. Descending little by little toward the sole of the kiln, the flames or hot gases finally fill the entire kiln chamber and maintain themselves throughout it, only passing to the chimney as they are displaced by hotter gases. In this manner the free lower surface of the hot layer of gases is very nearly stationary, which insures a practically uniform burn to the brick. In this atmosphere, which varies very little, the reactions of combustion are readily effected until only very slight traces of the combustible elements and free oxygen can be found in the gases. That is, combustion takes place with very nearly the theoretical supply of oxygen. The flames of this combustion traverse the entire mass of the gases and there are no definite points at which high temperatures may be found. For this reason the downdraft kiln is successfully employed when it is desired to obtain slow and uniform heating.

These experiments with a model of a furnace immersed in water confirm, with sufficient clearness, the fundamental principle that the circulation of the hot gases within a furnace is similar to the circulation of a light liquid within an enclosure filled with a heavy liquid.

III. THE CURRENT OF THE HOT GASES MAY BE COMPARED TO A STREAM OF WATER TURNED UPSIDE DOWN OR INVERTED

Streams composed of a heavy fluid in motion within a lighter fluid are seen everywhere. Do not all rivers represent the displacement of a light fluid—the air—by a heavy fluid—the water? In this case, it is very well known that the stream is confined on the bottom and the sides.

If the flame and the hot gases within the furnace were fluids heavier than the air, it would be found that they flowed in the same manner as the stream of water. But as they are much lighter than the air, it is found necessary to confine them upon the top and the sides.

This may be more clearly comprehended by means of the following laboratory experiment:

It is possible to pour a gas from one container to another by employing a sloping trough to guide its flow. This may be done with carbon dioxide gas, which is heavier than air, and also with hydrogen, which is lighter than air. When the carbon dioxide is being poured, the stream of gas must be confined below and upon its sides (Fig. 5). The hydrogen, on the other hand, must be confined upon the top and the sides (Fig. 6).

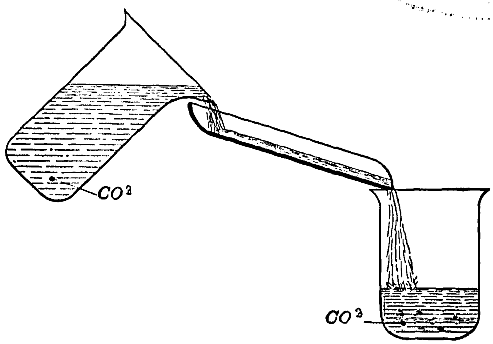


FIG. 5.

There is evidently nothing which confines the current of carbon dioxide upon the top and the stream of hydrogen on the bottom. These experiments require care, but are easy to make if the surrounding air is absolutely still and free from currents.[1] The gases may be poured equally well whether there is a fourth wall or not.

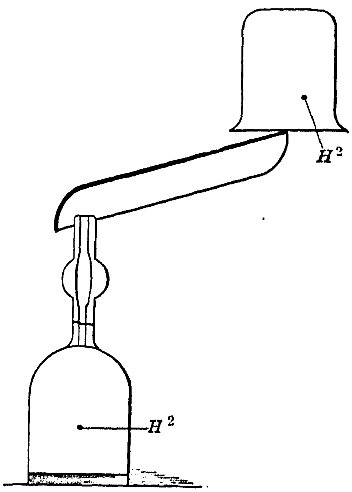


FIG. 6.

These experiments lead to the following conclusions:

Streams of incandescent gas need be confined only upon the top and sides, and, in effect, all reverberatory furnaces confine the stream of hot gases in this manner, at the top (the roof) and

[1] A condition which is neglected in the above experiment, is the tendency of all gases to form homogeneous mixtures by diffusion. In both experiments there will be a slight mixing with the air, as a result of this tendency.

upon the sides (the walls of the furnace) (Fig. 7). The confining boundary upon the bottom may be present or not. In the same manner, a stream of water may be confined by walls below and upon both sides, while its upper surface is entirely free. In a similar way the gaseous stream has to be confined upon the top and the two sides, while upon the bottom its surface is entirely free.

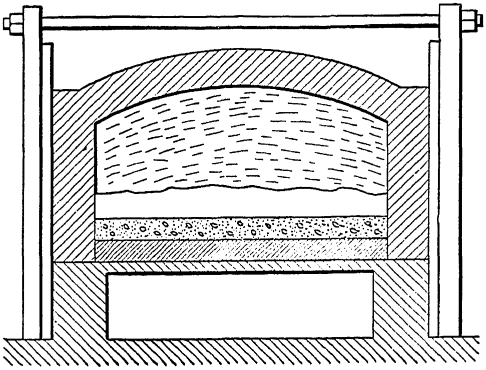


FIG. 7.

It would reveal a serious error in this theory, if, for example, the breeching, or smoke flue, common to four boilers was found to be completely filled with gases without regard to the number of boilers which were in service. This, however, will not be the case. In a correctly proportioned smoke flue the free lower surface of the gaseous stream will be found in the neighborhood of the bottom AA (Fig. 8). If the volume of the gases flowing decreases to one-fourth its former volume, the thickness of the stream decreases two and one-half times [1] or to 0.40 its former thickness, and the lower free surface of the gases will be found at the level BB; within the space $AABB$, no circulation of the gases will be found.[2]

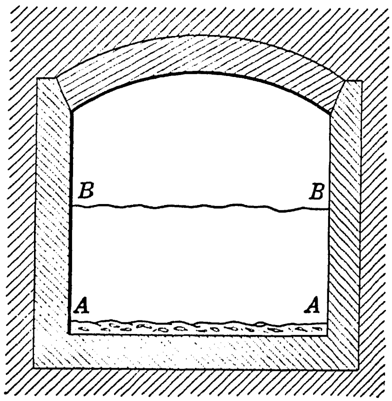


FIG. 8.

[1] Where $2.5=\sqrt{4^2}$, as may be seen further on, p. 40.

[2] This statement is not absolutely correct, as eddy currents will exist in the space $AABB$ due to the cooling effect of the walls upon the flowing stream of heated gases.

Just as a river has a depth which is a function of the volume of water flowing, it is evident that a stream of gases will have a thickness which is a function of the volume of gases flowing. If, on this basis, water is used to simulate the cold air and colored kerosene to simulate the hot gases flowing in the flue, it would not be difficult to show a smoke flue filled or partially filled with a stream of kerosene. This demonstration is considered useless as it has been thoroughly established that flowing streams of hot gases do not require anything to confine their lower surface and that the thickness or depth of the stream is a function of the volume of the gases which are flowing.

Therefore, when currents of hot gases are dealt with, they will always be represented as *inverted streams of water*.

IV. APPLICATION OF THE LAWS OF HYDROSTATICS TO HOT GASES.

The weight per cubic meter of the products of combustion of the ordinary combustibles varies from 1 kg 29 to 1 kg 33 at 0° and 760 mm. The computations may be simplified, and will be sufficiently exact, if the first of these values is assumed as the weight of the products of combustion, because this value is also the weight of a cubic meter of air. At any temperature t the weight of 1 cu m of the hot gases will be, therefore, $\frac{1.29}{1+\alpha t}$ kg, in which $\alpha=\frac{1}{273}=0.00367$.

By reason of this large coefficient of expansion of gases, the difference between the weight of a cubic meter of atmospheric air (1 kg 29) and a cubic meter of the gases of combustion, taking, for example, those in an open-hearth furnace (0 kg 17) is quite large, being equal to $1.29-0.17=1$ kg 12. It is this difference between the weight of the air and the weight of the flame or hot gases which causes the hydrostatic pressure of the latter.

The following experiments will serve to make this clear (Fig. 9). The upper surface of the water in a beaker is at *aa*; and *B* is a lamp chimney into which kerosene has been poured until its lower surface is at the bottom of the lamp chimney. It can be seen that the upper surface of the kerosene *bb* in the lamp chimney is higher than the surface of the water in the beaker.

The difference in level between *aa* and *bb* can be computed in the following manner. The column of water h is balanced by the column of kerosene H. The specific weight of water is 1.0 and that of kerosene is 0.80; it follows that:

$$H = \frac{h}{0.8} = 1.25h,$$

and the difference in level will be $0.25h$, equivalent to a column of water of $0.25h \times 0.8 = 0.2h$.

This experiment may be modified as follows: take a beaker

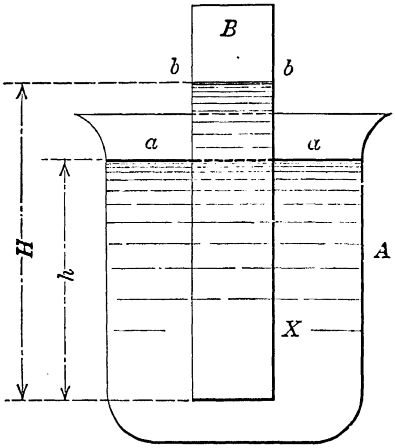


FIG. 9.

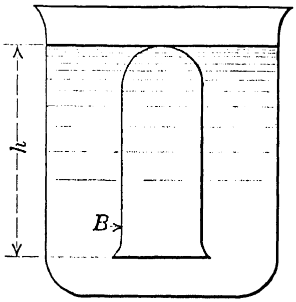


FIG. 10.

of water and immerse therein a test-tube filled with kerosene, until the bottom of the test-tube is even with the surface of the water, the test-tube being inverted (Fig. 10), and determine the hydrostatic pressure which it supports. It is evident that at the level B the pressure exerted by the water upon the kerosene and the pressure reciprocally exerted by the kerosene upon the water are equal, because they are in equilibrium.

The pressure of the water per unit of surface in millimeters of water column is that given by the height h:

$$P_{\text{water}} = h \times \text{density of water} = h \times 1 = h \text{ mm.}$$

The pressure of the kerosene is measured, first, by the weight of the column of kerosene, and, second, by a certain hydrostatic pressure δ to be determined. From which:

$$P_{\text{kerosene}} = h \times \text{density of kerosene} + \delta = 0.8h + \delta \text{ mm of water.}$$

But since $P_{\text{water}} = P_{\text{kerosene}}$, it follows that:

$$h = 0.8h + \delta \quad \text{and} \quad \delta = +0.2h \text{ mm of water}$$

If it is considered that the water in this experiment represents the cold air and that the kerosene represents the incandescent gases in the furnace, the following law may be established with regard to the hydrostatic pressure which will be produced at the different parts of a furnace chamber containing hot gases:

The hydrostatic pressure δ in kilograms per square meter at a point in a chamber bathed by the incandescent gases, located at a distance H above the free surface of those gases, is equal to the difference Δ between the weight in kilograms of a cubic meter of the external air and a cubic meter of the incandescent gases, multiplied by the height H, from which

$$\delta = H\Delta.$$

Example.—If $H = 0$ m 70 and the weight[1] of 1 cu m of hot gases at 1200°,

$$P_{1200} = \frac{1.33}{1 + \frac{1200}{273}} \text{ kg} = 0 \text{ kg } 25,$$

from which

$$\delta = 0.7\ (1.29 - 0.25) = 0 \text{ kg } 728 \text{ per square meter,}$$

or 0 mm 728 of water, since the pressure of 1 kg per square meter is equal to the pressure exerted by a column of water 1 mm in height.

Experiments which may be readily made will show that the light hot gases which fill the furnace are actually exerting a pressure greater than that of the atmosphere.

Open the register connected with any hot-air house-heating system. A jet of hot air escapes with some force. What is it that sets this air in motion? What is it that provides the energy necessary for this motion?

Open the sight hole located at the upper part of an open-hearth regenerator chamber. If the regenerator is not connected

[1] Refer to Appendix II.

with the chimney, a jet of incandescent gas or air will escape with considerable force. What is it that sets this in motion?

Open the bell of a gas producer working with natural draft; the producer gas will escape. What is the force which causes the air to pass through the bed of wood or coal, where it is transformed into gas, and which, in addition, has sufficient pressure to produce the jet of gas?

From the fact that enclosures filled with a cold gas, but one which is lighter than the air, always exert a pressure higher than that of the atmosphere, it follows that the preceding phemonena are due *not to the temperature of the gas*, but to the fact that it weighs less than the air.

Take, for example, a balloon 10 m in diameter, filled with hydrogen (Fig. 11). Compute the hydrostatic pressure of the gas at the top of the sphere. This balloon being open at the bottom, at the level AA the gas is in equilibrium with the air, their pressures being equal.

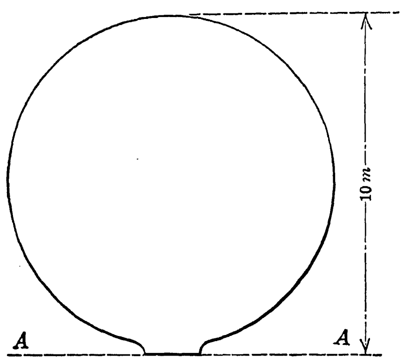


Fig. 11.

Consequently the weight of a column of cold air 10 m in height (1 kg 29×10 = 12 kg 90) is held in equilibrium by the weight of a column of hydrogen 10 m in height plus a certain hydrostatic pressure δ which may be determined.

The weight of a cubic meter of hydrogen being equal to 1.29× 0.06927, the following equation is obtained:

$$1.29\times 10 = 1.29\times 0.06927\times 10 + \delta; \quad \delta = 12 \text{ kg } 006,$$

which is in the neighborhood of 12 mm of water.

It is on account of this hydrostatic pressure of the hydrogen at the top of the balloon that it is necessary to employ a very strong material in the making of this envelope. It is this pressure which causes the balloon filled with hydrogen to ascend and which furnishes the energy for the flow of the hydrogen when the valve at the top of the balloon is opened to permit it to escape. Furthermore, children are often amused by making small hot-air balloons of paper, which are then filled by the hot air rising from a samovar or a lamp. Many readers will undoubtedly recall the circumstances attending the launching of such a balloon; the hot air enters the balloon, expanding its envelope, and the balloon tends to fly. This would not happen if the pressure in the balloon were not higher than the atmospheric pressure.

The question of this hydrostatic pressure, existing in all enclosures filled with a gas which is hot and for that reason lighter than air, is the basic one of the hydraulic theory of reverberatory furnaces. It is therefore desirable to give another example which occurs in daily life.

In a building piped for both gas and water it is very evident that the hydrostatic pressure of the water will be greater upon the lower floors than it will be upon the upper floors. Will it be the same for the gas? If, for example, the gas upon one floor has a pressure of 25 mm of water, what will be the gas pressure 10 m higher?

Assume that at the stopcock R (Fig. 12) the gas pressure is equal to that of the atmosphere. The pressure of the column of air per unit of surface, which is equal to (1.29×10) kg, will be held in equilibrium by the weight of the column of gas 10 m in height, increased by the hydrostatic pressure δ of the gas; the weight of a cubic meter of the gas is equal to 1.29×0.4, from which

$$1.29\times10=1.29\times0.4\times10+\delta,$$
$$\delta=7 \text{ kg } 74 \text{ per square meter.}$$

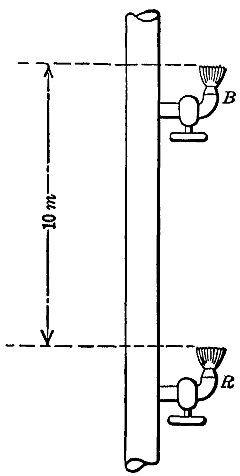


FIG. 12.

But the pressure of the gas at the point R is not only equal to the atmospheric pressure, but exceeds it by 25 mm.

It is therefore necessary to add to both sides of the equation +25 mm. It follows that the gas pressure at the point B (since 1 kg per square meter is equivalent to 1 mm of water) may be expressed as follows:

$$\delta+25=32 \text{ mm } 74,$$

that is to say, the pressure of the gas at the higher floors of the building is always greater than it is on the lower floors. The increase of the pressure in the case of gas is the inverse of that which occurs with water.

In *Hütte,* in the chapter upon illuminating gas, the following statement occurs: " A rise (or a drop) of 10 m in height in the mains corresponds to an increase (or a loss) in pressure in the neighborhood of 7 mm of water column." (1)

The following rule, to be followed in the location of a gas works is given in order that there may be a uniform pressure in the holder and throughout the distributing system: " The gas works, should be installed at the lowest point in the system and should be so located that the highest point in the supply system is furthest from the works, and the lower parts are closer to the works; under such conditions the loss of gas pressure due to friction in the mains is compensated for by the increase of the hydrostatic pressure in the piping, in proportion to its distance from the works and its elevation above their level."

The existence of hydrostatic pressure in the gas having been established, it is now possible to take into exact account those phenomena which are presented in metallurgy.

(*a*) **The Draft Fallacy of Metallurgical Furnaces.**—It was believed for a long time that furnaces having a natural current of air through them operated through the effect of their chimneys. This is an error due to a poor interpretation of the facts. One of the first principles which must never be forgotten is that the only furnaces which operate by the draft provided by the chimney are those which have no working doors or openings for the charging of material to be heated. Furnaces of this sort include boilers, entirely enclosed in a setting built of brick or other material, crucible melting furnaces which are very nearly hermetically closed up, iron tube air heaters (formerly used for heating the air at blast furnaces), the Cowper hot blast stove, etc.

Furnaces which have working doors to their laboratories or heating chambers, such as reverberatory furnaces, melting furnaces, puddling furnaces, brick kilns, Siemens furnaces of all kinds (open-hearth and others), do not operate by the draft of the chimney. The chimney connected with these furnaces is only an apparatus for removing the products of combustion from the laboratory. The colossal chimneys which are very often seen are

(1) Vol. II, p. 869, French edition of 1911.

frequently unnecessary for this purpose. Therefore all of those furnaces which are provided with working openings or doors operate without "draft." This fact will be established.

Consider any reverberatory furnace (Fig. 13) when in operation. The doors are never hermetically tight to the walls, and a flame or sting escapes from the interstices at the top of the door. If a

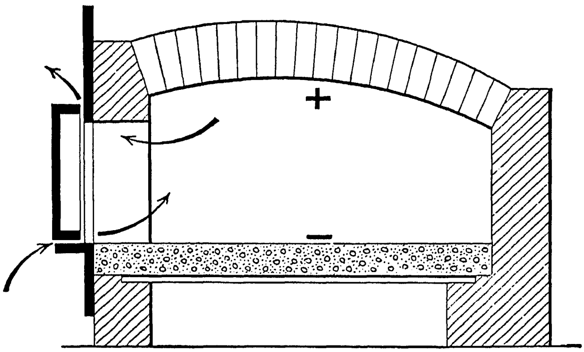


FIG. 13.

lighted torch is held close to the upper part of the door its flame will incline away from the crack. On the contrary, if the torch is held close against the crack at the lower part of the door, it will be noticed that the flame will be drawn into the furnace. It can be stated, therefore, that immediately below the roof of the furnace the pressure of the hot gases inside the furnaces is higher than the atmospheric pressure at that level; whereas, at the level corresponding to the lower part of the furnace the pressure of the hot gases is less than the atmospheric pressure, that is to say, *negative* with regard to the pressure at the roof of the furnace.

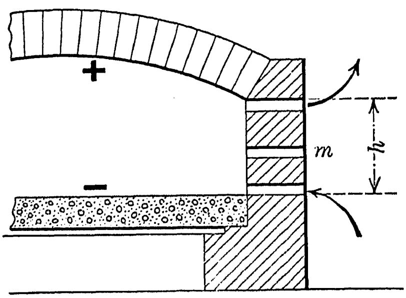


FIG. 14.

If two openings are made in the walls of this furnace (Fig. 14), one at the level of the sole, the other at the level of the roof, the phenomena will be the same. By the lower opening the external air will be drawn into the furnace, whereas from the upper opening a tongue of flame will escape. From this it might be concluded that if an opening *m* were made halfway between the top and the bottom of the wall of this furnace, there would be no tendency for the air to be drawn into the furnace nor for tongues of flame to escape.

In reality the phenomenon which occurs is slightly different. At this opening *m*, the outer air will sometimes be drawn into the furnace and at other times small jets of flame will escape, as the pressure within the furnace varies. That is, the level at which the pressure in the furnace is in equilibrium with the atmospheric pressure shifts vertically, now above and now below the level of the opening *m*.

These simple observations show that the pressure of the hot gases within the laboratory of a metallurgical furnace provided with working openings or doors is, on the average, equal to atmospheric pressure. These pressures may be directly measured by the use of a manometer.(1)

Let it now be considered whether it would be possible for a metallurgical furnace to work in a regular and uniform manner if the pressure of the hot gases within the furnace were less than the atmospheric pressure. If the foregoing occurred, an enormous quantity of cold air would be drawn in through the working doors. When this occurred, the depression due to the chimney draft would be entirely overcome. And, further, by reason of this inrush of cold air, the temperature of the furnace would be lowered to such an extent as to produce a very bad effect upon the working of the furnace. In addition, this would be likely to damage the brickwork of the furnace.

In order to understand the working of those furnaces which operate with a natural current of air it is not necessary to take into account what is called the "draft of the chimney," the only function of the chimney being to remove the burned gases from the heating chamber, in order to provide space for the new or burning gases. By placing this construction upon the question of chimney

(1) Extensive observations covering the temperatures and the pressures at various points in an open-hearth furnace have been made by E. Juon, at the Donetz-Jurjewka works, Russia. M. Juon's paper appeared in Stahl und Eisen, Oct. 24 and Nov. 7, 1912. It was abstracted in *The Iron Age*, Dec. 26, 1912.

draft, it becomes possible to take into account the manner in which metallurgical furnaces operate.

The pressure of the gas within the laboratory of the furnace is zero. The firebox acts to *pump* the elements necessary for the formation of the flame into the furnace, and the chimney acts to remove the products of combustion. This is the case with all such furnaces and with all systems of heating, whether simple coal-fired or using producer gas with or without the recuperation of the heat. This rule is also applicable to coke ovens, to kilns for the burning of brick or for the calcining of ores.

Each firebox acts as a *force pump*, functioning in the following manner: assume that the firebox A (Fig. 15) is filled with gas at a temperature of 1200°, that is to say, the specific weight of this

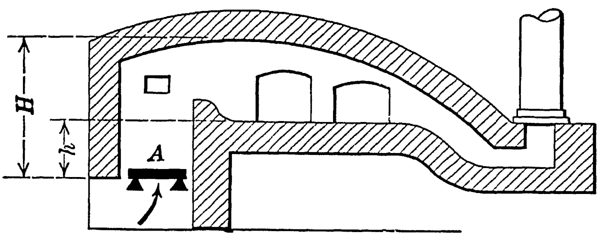


FIG. 15.

gas will be about one-fifth the weight of air. Under these conditions each particle of air which is near the grate is acted upon, on the side of the furnace, by the pressure of a column of gas h, which is only one-fifth the weight of that which acts upon it on the side exposed to the outside air. It is evident that a current of air will be established by the action of this positive pressure, which will be equal to the difference between the weight of a column h of the outside or cold air and the same column of hot gases.

The height of the column has been taken as h and not as H, because, in the furnace chamber, the current of hot gases rising from the firebox must always be taken with regard to the hearth level.

The positive pressure which is measured by the height of the column h is expended: (a) in overcoming the resistance of the bed of fuel to the passage of the air and the gas; (b) in creating the velocity with which the gases and the flame flow, and which also acts as a reserve force through their inertia.

In a number of types of furnaces this live force or velocity is utilized to direct and force the flame or hot gases from the top down upon the hearth of the furnace. It has been shown that the cold external air has a tendency to enter the furnace through the cracks or openings below the working doors; since this cold air settles upon the hearth it acts as a heavy liquid flowing within a chamber filled with a light liquid, the hot gases. However, the material to be heated (ingots, faggots, billets, bars, etc., in the furnaces for the reheating of small pieces of metal, the molten metal of the open-hearth furnace, etc.) is always placed upon the

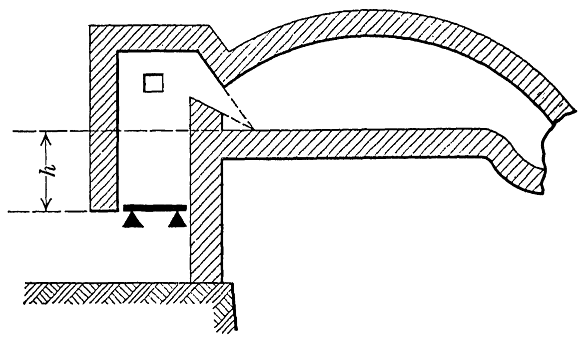


Fig. 16.

hearth. Therefore it is necessary to overcome the formation of such a current of cold air. There are two means of doing this:

1. Increasing the velocity of the flame by increasing the height h.

2. Bending the flame sharply down upon the hearth as it issues from the firebox, and in this way pushing back the cold air and preventing it from entering the furnace through the openings below the doors.

A furnace constructed to employ this second method is shown in Fig. 16. The increasing of the head h provides a possibility of increasing the thickness of the bed of fuel and of obtaining in the firebox a producer gas at a slightly lower temperature. This increase in the head of the column of gases also permits the mixing of the producer gas from the firebox with a supply of preheated secondary air, supplied through channels in the bridge wall.

Boetius and Bicherou have utilized this method of firing in a furnace invented by them and bearing their names.

When many of the various existing types of furnaces, such as the continuous furnaces, Siemens furnaces, brick kilns, etc., are analyzed, it will be noted that their construction is such as to preclude, in reality, the draft action of the chimney in drawing in outside air and to limit the action of the chimney to the removal of the waste gases. The drawing in of the cold outside air would have an injurious effect upon the working of the furnace, particularly in the case of brick kilns. These furnaces, therefore, constitute a perfect illustration of the fundamental principle of the method in which metallurgical furnaces operate. It is well known that the pressure of the hot gases in the heating chamber of such furnaces is practically equal to the pressure of the atmosphere. The gases coming from the firebox and the supply of secondary air, if such is necessary for the operation of the furnace, enter the heating chamber under a pressure head which may be measured by the difference in level between the grate bars and the hearth of the furnace. The chimney for a metallurgical furnace should be designed to provide for the removal from the heating chamber of the burned gases, but it should not, in addition, provide any further draft depression. On the contrary, it is always desirable that there should be a slight positive pressure in the heating chamber of metallurgical furnaces.

(*b*) **Hydrostatic Pressure within Metallurgical Furnaces and their Flues.**—1. All reverberatory furnaces comprise a chamber of some fixed height filled with incandescent gases. At each point in the interior of this chamber there will be a different pressure, according to the height of the point. Therefore, in a reheating furnace, having working doors with a height of 700 mm, with a pressure equal to that of the atmosphere acting at the hearth level, there will be a pressure +0 mm 728 of water, at the level of the top of the door, as has been shown previously on page 13.

This pressure, in excess of that of the atmosphere, causes, as is well known, the formation of a great aureole or "sting" of flame due to the hot gases which escape through the crevices at the top of the door. In order to diminish the loss of gases through these crevices, the following means may be employed. By a slight increase in the chimney draft, the level at which the pressure in the furnace is in equilibrium with the atmospheric pressure may be changed to about the mid-height of the working door. This will produce the following results: the pressure at the level of the hearth of the furnace will be reduced below that of the atmosphere and will be (−0 mm 36), and air from outside the furnace will commence to flow into it under the door; the pressure at the top of the door will be diminished by one-half, becoming (+0 mm 36), and the loss of hot gases will be diminished correspondingly.

2. Hydrostatic pressure readily explains the irregular working of sloping grates in gas producers fired with wood. There is a current of gas with such a grate, as indicated by the arrow at the right of Fig. 17. The negative pressure which exists at the level of the horizontal grate bars varies gradually until it becomes positive at the top of the sloping grate (compare with page 141), under which some of the gas commences to escape and burn, a pure loss of energy.[1] The decrease in the draft produced in this manner also occurs with the sloping grates used under boilers.

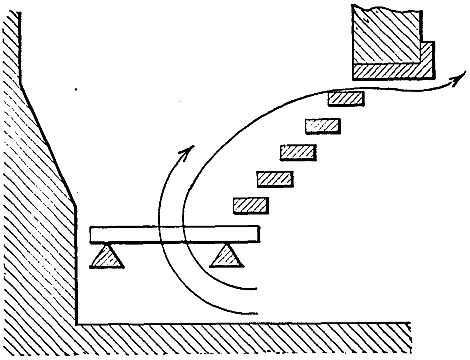


Fig. 17.

This is the reason why this type of grate has never been considered satisfactory with natural draft.

3. The existence of this hydrostatic pressure readily explains the draft of gas producers which operate without forced draft, with the Siemens siphon (Fig. 18). The gas from the producer passes up through

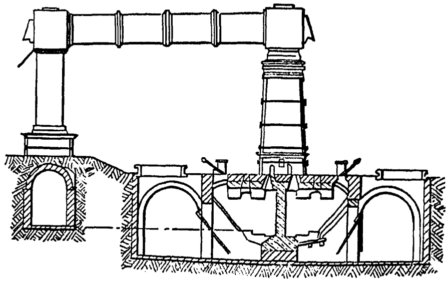


Fig. 18.

[1] With forced draft and a closed ash pit, sloping grates work comparatively well.

the masonry gas uptake with an average temperature of 600°, thence into a long horizontal main constructed of steel plates, where its temperature drops to approximately 300°. The gas then flows down a vertical gas downtake to the underground gas flue, which it reaches with a temperature of 200°.

Assuming that the height of the masonry gas uptake is 9 m above the grate of the producer and that the producer gas has a weight of 1 kg 07 per cubic meter at 0°, the hydrostatic pressure of the gas at the top of the uptake will be

$$\delta_{600}=9\left(1.29-\frac{1.07}{1+\frac{600}{273}}\right)=8 \text{ mm } 64 \text{ of water.}$$

The average temperature of the gases in the downtake will be

$$\frac{300°+200°}{2}=250°.$$

In order *to force the hot gas down* through the vertical downtake there must be an initial pressure at the head of the downtake sufficient to overcome the hydrostatic pressure of the gases in the downtake with an average temperature of 250°.

The resistance due to the hydrostatic pressure in the downtake is

$$\delta_{250}=9\left(1.29-\frac{1.07}{1+\frac{250}{273}}\right)=6 \text{ mm } 57 \text{ of water.}$$

According to the foregoing the hydrostatic pressure of the gas within the underground flue will be equal to

$$\delta=\delta_{600}-\delta_{250}=8.64-6.57=+2 \text{ mm } 07 \text{ of water.}$$

As the frictional resistance of the gas mains and flues to the passage of the gas is generally less than 2 mm 07 of water, the Siemens " siphon " favors the " draft " of the gas producer. It is evident that the draft on the gas producer would be stronger if the length of the conduit were increased, as in this case the drop in temperature in the horizontal section would be greater.

The position of the horizontal gas main, at a considerable height above the grate of the gas producer, assures a pressure in the main somewhat higher than that of the atmosphere, and this constitutes a guarantee against the risk of explosions with gas producers operating without forced draft. On the contrary, a construction in which the gas flows directly from the producer into the underground conduit or gas flue will always possess latent possibilities of an explosion, because the gas in the producer will be at such a low pressure that there will be a negative pressure in the underground flue, with producers working without forced draft.

A gas producer with mains connected in this manner was installed at the Alapayevsky works. The producer was worked without forced draft and the gas main was long. When placed in operation this proved so dangerous that it became necessary to replace the gas main with a metal main forming a Siemens siphon. When this was done there were no more explosions.

4. This method of computing hydrostatic pressure may be applied to the gas mains of a blast furnace producing charcoal iron. These furnaces have very large bells for charging and the volume of gas is relatively small. The percentage of the gas lost by the bells is considerable and increases rapidly as the pressure of the gas in the mains increases. For this reason it is very important that the mains should be designed to *require the minimum of gas pressure* to overcome the unavoidable resistance to the flow of the gas. It is also possible to determine the excess of pressure required to overcome the resistance created by poorly designed gas main, too small for the volume of gas flowing, having sharp bends, etc.

Assuming that the height of the furnace is 20 m, the height of a cubic meter of the gas being 1 kg 22 and the temperature in the main 200°, the hydrostatic pressure required at the top of the furnace can be determined. Assuming, on the other hand, that the gas is to be burned under boilers located 3 meters below the hearth level of the blast furnace, the pressure in the gas mains or the burners at the boiler after regulation and control, must not be less than the atmospheric pressure.

The gas mains form an enclosure or chamber 23 meters in height, filled with a light gas which has the following weight per cubic meter:

$$\Delta_{200} = \frac{1.22}{1+\frac{200}{273}} = 0 \text{ kg } 705$$

The hydrostatic pressure will be, therefore,

$$\delta = 23\ (1.29 - 0.705) = 13 \text{ kg } 45 \text{ per square meter,}$$

or 13 mm 45 of water. This pressure of 13 mm 45 is absolutely necessary, as well as a slight additional pressure to overcome the friction in the mains and impress upon the gas the necessary velocity of flow. In the case of a blast furnace producing charcoal iron, the pressure at the head of the furnace was 35 mm of water, the gas mains being too small and poorly designed. This bad construction of the gas mains could not be corrected without a general reconstruction. The method of arriving at the resistance caused by friction in the gas mains will be given later.

(*c*) **Hydrostatic Pressure in Open-top Chambers Filled with a Light Gas.**—If an inverted test-tube is held vertically (Fig. 19)

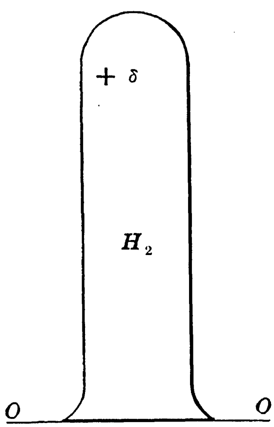


FIG. 19.

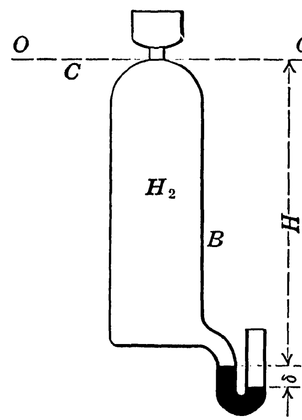


FIG. 20.

and filled with hydrogen the pressure at the level *OO* will be equal to that of the atmosphere, and at the upper part there will be a positive pressure $+\delta$. This experiment may be modified by using a vessel filled with hydrogen (Fig. 20) which has a capillary strangulation at the top surmounted by a thistle, and a U tube at the bottom. The capillary tube is sealed with a drop of water, permitting the hydrogen to escape, when its pressure exceeds that of the atmosphere, without, at the same time, giving free communication between the interior of the vessel and the air. It necessarily follows that the pressure in the interior of the vessel at the level *OO* is equal to that of the atmosphere. What will be the pressure indicated by the manometer U tube filled with water at *K* connected with the lower part of the vessel? Upon the free surface of the water the pressure will be equal to the pressure of a column of air whose height is measured by $H+\delta$ and whose weight is $1.29\times(H+\delta)$ kg. This column of air is in equilibrium with a column of hydrogen whose height is H and with a column of water with a height δ, the existence of which can only be explained by a rarefaction of the hydrogen in the bottom of the vessel B. It is on account of this rarefaction that the equilibrium at the top may be maintained by the drop of water.

It is not difficult to compute the value of this depression by means of the general formula:

$$-\delta=H\cdot 1.29\cdot 0.06927-(H+\delta)\cdot 1.29.$$

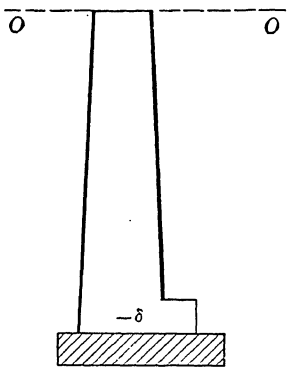


FIG. 21.

It may be noted that this is what takes place in all chimneys (Fig. 21), which are simply open-topped enclosures filled with a light gas. As in the preceding case, the waste gases at the base of the chimney have a pressure *less than* the pressure of the atmosphere. This phenomenon is well known as the *depression* produced by the chimney.[1]

The following problem remains to be considered:

In the heating chamber of an open-hearth furnace the pressure of the hot gases is in equilibrium with the atmospheric pressure. From the heating chamber the gases at a temperature of 1600° pass to the regenerators. There they give up a portion of their heat and are finally reduced to a temperature of 400° at the chimney. The weight of a cubic meter of these gases being 1 kg 30, what is the hydrostatic pressure which will be found at the level of the bottom of the regenerators, when the distance down from the heating chamber is 6 m? From what has gone before it may be seen that the pressure at the level of the bottom of the regenerators will be negative. In order to set the hot and light gases in motion and cause them to descend through the regenerator, the chimney must draw these gases from the heating chamber and accordingly

[1] *Note by English translator.*—Very often the waste gases in the chimney do not fill the entire area of the stack, due to their high ascensional velocity.

produce a depression at the bottom of the regenerator which will be equal, neglecting the resistance to their flow, to

$$\delta = -6\left(1.29 - \frac{1.20}{1+\frac{1600+400}{2\times 273}}\right) = -6 \text{ mm } 08.$$

To sum the matter up, when a chamber is filled with a light gas and the line at which its pressure is in equilibrium with the atmospheric pressure is at or below the lowest point of the chamber, the pressure in the upper part of the enclosure will be higher than atmospheric pressure. If the line of equilibrium with the atmospheric pressure passes through or above the upper part of the chamber, the pressure at the lowest point in the chamber will be negative. In other words: when a light gas ascends it will create a pressure; conversely, to force a light gas to pass downward, it is necessary to create a depression.

V. THE APPLICATION TO HOT GASES OF THE LAWS FOR THE FLOW OF LIQUIDS THROUGH AN ORIFICE IN THE BOTTOM OF THE RESERVOIR.

For water flowing into air the formula is:

$$Q = \kappa_1 \kappa_2 \omega \sqrt{2gh},$$

in which Q = the volume of liquid flowing;
h = the head of the liquid above the orifice;
$\sqrt{2gh}$ = the theoretical velocity of flow;
ω = the area of the orifice;
κ_1 = the coefficient for the contracted vein, that is, the ratio between the area of the contracted vein and ω (generally $\kappa_1 = 0.64$);
κ_2 = the coefficient of velocity, that is, the ratio between the actual velocity at the contracted vein and the theoretical velocity (generally $\kappa_2 = 0.97$).

If it is desired to apply this formula to the flow of kerosene through an orifice in the top of a bell glass immersed in a vessel of water, it is necessary to determine exactly what is meant by the head h.

Assume that a bell glass with an orifice of an area ω, within which the kerosene is constantly maintained at a height H (Fig. 22), is immersed in water. Neglecting, for the moment, the coefficients of contraction κ_1 and of velocity κ_2, which, for this case, have not been determined, the formula for the flow will be simplified. The hydrostatic pressure at the orifice ω will be equal to the weight of a column of water H less the weight of a column of kerosene of the same height, that is to say:

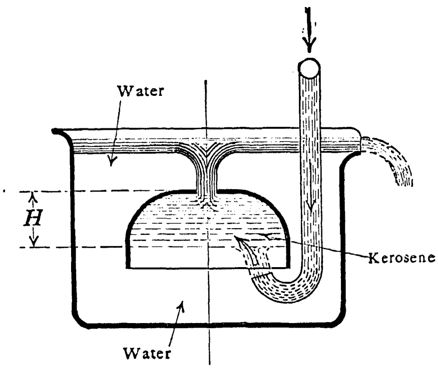


FIG. 22.

$$\delta = H(\Delta_{\text{water}} - \Delta_{\text{kerosene}}),$$

Δ designates the specific weights corresponding to the indices.

But as this head must be given in meters of height of the liquid which is flowing, the expression for the head will be

$$h_{\text{kerosene}} = H\,\frac{\Delta_{\text{water}} - \Delta_{\text{kerosene}}}{\Delta_{\text{kerosene}}}.$$

Introducing this value of h in the formula for the volume flowing, will give, for the volume of a light liquid (kerosene) flowing through an orifice into a heavy liquid (water), the following expression:

$$Q = \kappa_1\kappa_2\omega\sqrt{2gH\,\frac{\Delta_{\text{water}} - \Delta_{\text{kerosene}}}{\Delta_{\text{kerosene}}}}. \quad . \quad . \quad . \quad . \quad (A)$$

Passing to the case of the air and the gases in furnaces, this expression will become

$$Q = \kappa_1\kappa_2\omega\sqrt{2gH\,\frac{\Delta_{\text{air}} - \Delta_{\text{gas}}}{\Delta_{\text{gas}}}}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (B)$$

in which H designates the distance from the lower free surface of the gas to the opening in the roof, or, as it may be said, the thickness of the layer of gas.

This may be clearly seen by a reference to Fig. 2, in which is given the orifice at the highest portion of the roof of the kiln having an area ω, such that the lower free surface of the layer of kerosene in the model has become stationary at a distance H below the smoke hole or orifice.

It is evidently desirable that the values of the coefficients κ_1 and κ_2 should be determined in order that they may be used in the equation (B). The values of Q, ω and h may be determined experimentally. Unfortunately this has not been done up to the present.

For gases, in the case of comparatively large orifices, these coefficients will probably approximate unity. This is the value which has been assigned to these coefficients for the purpose of solving the problems presented in this work.

The case shown in Fig. 2 may be especially called to the attention of those who are operating reverberatory furnaces. The author had the opportunity to verify this in the works of Korniloff Brothers, at Petrograd, accompanied by Professor J. G. Yesmann.(1) In the second story of the porcelain kiln (Fig. 85, page 114) where the waste gases passed out through the roof, the lower free surface of the gases was clearly seen at a height of from 1 to $1\frac{1}{2}$ meters above the hearth of the kiln. This lower free surface of the layer of gases evidently might be forced down as far as the hearth of the upper chamber by the obstructing of the smoke hole in the roof of the kiln, as with the experimental models which have been shown.

When the lower free surface of the layer of gases is brought down into contact with the hearth of the kiln, it gives the nearest approach to a uniform burn that it is possible to obtain with a furnace operating in this manner, which is old and irrational. Complete regularity of the burn cannot be attained, as has been shown, except by the use of the *downdraft system* (Fig. 4).

Getting rid of the waste gases of combustion by means of an opening in the highest part of the furnace was a method very much in favor in former times;(2) but in later furnaces this method has been completely superseded. However, an example is given

(1) Professor of hydraulics at the Polytechnic Institute of Petrograd. Refer to p. 153 and following.

(2) Updraft furnaces are frequently employed in the manufacture and smelting of copper, lead, and certain other metals.

of the method of computation which is used for an updraft furnace.

From a reheating furnace at the Lougansk works (Russia) the products of combustion, at a temperature of 600°, pass to a boiler by an outlet port in the roof. The volume of these gases, based upon the actual coal consumption per second, is computed as being equal to $Q_{500} = 8$ m^3 58 per second, with an air supply equal to one and one-half times the theoretical amount required. The port in the roof of the furnace had a section of $2.5 \times 0.6 = 1$ m^2 50. With these conditions fixed, at what distance below the roof will the free surface of the layer of gases in the furnace be located?

$$Q = \kappa_1 \kappa_2 \omega \sqrt{2gH \frac{\Delta_{\text{air}} - \Delta_{\text{gas}}}{\Delta_{\text{gas}}}}$$

$$8.58 = \kappa_1 \kappa_2 \cdot 1.5 \sqrt{2gH \frac{1.29 - \frac{1.33}{1 + \frac{600}{273}}}{\frac{1.33}{1 + \frac{600}{273}}}}$$

Assuming $\kappa_1 \kappa_2 = 1$, $\sqrt{2g} = 4.43$, it will be found that

$$H = 0 \text{ m } 896.$$

And therefore, according to these calculations, the distance from the free lower surface of the gas layer to the port, in the case of the gases flowing into the air, would be 896 mm. In the furnace this distance actually was 800 mm.

This shows that the designers of this furnace had taken into account a very old rule which says that if a reverbatory furnace is to work well, the flame must *lick* the hearth of the furnace. The height of 896 mm not only assures the contact of the flame with the sole of the furnace, but also permits, by the increase or diminution of the draft through the boiler, of the regulation of the thickness of the gaseous current; that is to say, it provides a means of drawing the flame up from the hearth or of forcing it down against it.

Although the above computation has already been given, it is repeated here that the port for the waste gases should never be placed in the roof of the furnace. It is a bad system and one that is now out of date. In the operation of those furnaces where the waste gases are taken off by a port through the roof, the hot gases cannot distribute themselves uniformly; the hottest currents tend to rise and pass out of the waste gas port in the roof, and there will be currents of cold and therefore heavy gases in the neighborhood of the ingots on the hearth, which accordingly will not be well heated.

Therefore *the waste gases should always be carried away from the heating chamber by a port at the level of the hearth of the furnace;* the colder currents of gases will then pass out of the heating chamber and their place will be taken by currents of hot gases; under these conditions the ingots will be well heated. These remarks apply to furnaces of all kinds, and are a general rule for the construction of all furnaces of rational design. It has been well established that the port for the waste gases must be located at or below the level of the hearth of the furnace.

Returning to the computations, the equation (B) may be considerably simplified, if it is assumed that the weight of a cubic meter of waste gases is the same as the specific weight of air (1 kg 29). This is very nearly true. Therefore,

$$\frac{\Delta_{\text{air}}-\Delta_{\text{gas}}}{\Delta_{\text{gas}}}=\frac{1.29-\dfrac{1.29}{1+\alpha t}}{\dfrac{1.29}{1+\alpha t}}=\alpha t.$$

$$Q=\kappa_1\kappa_2\omega\sqrt{2gH\alpha t}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (C)$$

The expression $H\alpha t$ is that of the velocity head h, because

$$v=\sqrt{2gH}.$$

Therefore $$h=H\alpha t,$$

and $$\frac{h}{H}=\frac{t}{273}.$$

Therefore *the ratio of the head required to generate the velocity to the height H is equal to the ratio of the temperature t to* 273.

This expression will be found used later in connection with jets of gas, page 65.

VI. RELATION BETWEEN THE HEAD, THE PRESSURE AND THE VELOCITY OF CURRENTS OF LIQUIDS AND GASES

If the velocity of flow of any current is considered, as in a canal or a river, it is a function of the head expended. As the bottom or bed of a torrent of water is never smooth, the velocity of flow is only maintained by the expenditure of a velocity head sufficient to compensate for the velocity lost at the obstructions. The water in a stream flows only when its free upper surface has a sufficient slope or hydraulic gradient to cover the loss of velocity due to the friction against the bottom and sides of the stream.

Therefore, *there will be no flow or current unless there is a corresponding loss of velocity head, and if there is a flow there will be a corresponding loss of velocity head in impressing this velocity upon the stream and in maintaining it.*

If the velocity of the flowing current is v, then

$$v = \sqrt{2gh} \quad \text{and} \quad h = \frac{v^2}{2g},$$

in which h represents the velocity head expressed in terms of the liquid in motion (water, kerosene, liquid iron, mercury, etc.).

For example, when any liquid is impressed with a velocity of flow of 4 m 43 per second, there will be required, according to the formula, the expenditure of a velocity head h of 1 m. This will be 1 meter in height of water, kerosene, liquid iron, mercury, or of the particular liquid which is in motion.

If this meter of velocity head is expressed in kilograms per square meter, it will be found that, for each of these different liquids, a different pressure is required to produce the same velocity of 4 m 43 per second, according to their density, as follows:

Liquid	Kilograms per square meter	Atmospheres	Millimeters of water
Kerosene	800	0.08	800
Water	1,000	0.10	1,000
Liquid iron	6,900	0.69	6,900
Mercury	13,595	1.3595	13,595

From this it may be seen that without regard to the kind or density of the liquid which is in motion, any particular velocity will correspond to a velocity head, which may be absolutely determined and which will be obtained by the formula

$$v = \sqrt{2gh}.$$

But the pressure necessary to impress motion upon a liquid will be different for each liquid and is proportional to its density.

In considering the movement of gases, the same fundamental principles must be observed:

1. Motion or velocity of flow cannot be initiated or maintained without a corresponding loss of head or pressure, as the loss of head and the velocity impressed have the relationship of a cause and an effect.

2. Any particular velocity of flow corresponds to a velocity head expressed in terms of the gas which is in motion, which may be determined by the formula

$$v = \sqrt{2gh}.$$

3. Equal velocities will be impressed upon gases of different densities by equal velocity heads, but by different pressures. These pressures are proportional to the specific weight of these gases or to the weight of a unit volume.

As the gases appear to act as a form of liquid having the particular property of considerable variation in its specific weight according to changes in its temperature, the third principle is of great importance in all computations relative to the circulation of the flame or hot gases.

The following examples will illustrate this point:

A. In the heating chamber of an open-hearth furnace the hot air emerges from the port with a velocity of 18 m per second at a temperature of 1000°. What is the pressure required to impress this velocity upon the air?

To impress a velocity of 18 m per second upon a fluid will require a velocity head $h = 16$ m 51.[1]

In this case it is air at 1000° which is in motion; the weight of 1 cu m of this gas is

$$\Delta_{1000} = \frac{1.29}{1 + \frac{1000}{273}} = 0 \text{ kg } 277.$$

[1] Refer to Appendix III.

A column of air 16 m 51 in height at a temperature of 1000° produces a pressure per square meter of

$$\delta = 0.277 \times 16.51 = 4 \text{ kg } 57 \text{ per square meter}$$

or $$\delta = 4 \text{ mm } 57 \text{ of water.}$$

Therefore, in order to impress upon the hot air emerging from the port a velocity of 18 m per second it is necessary to provide a pressure acting upon that air of 4 mm 57 of water column. This is the hydrostatic pressure created in the regenerator chamber and the vertical uptakes, or flues leading to the port, which are filled with hot air.

If it is assumed that during the passage of the air through the regenerator checkerwork and through the uptake there is no loss of pressure due to friction or other causes the vertical height required to produce this pressure may be computed. The value found in this manner will evidently be too small, since actually the resistance and friction through the regenerator and uptake cannot be neglected.

Assuming that the air at the bottom of the regenerator has a temperature of 50° and the temperature at the heads is 1000°, the average temperature will be 525°. The hydrostatic pressure which this will give is

$$\delta = 4.57 = H\left(1.29 - \frac{1.29}{1+\frac{525}{273}}\right),$$

from which the value of $H = 5$ m 37.

Therefore the vertical height of the regenerator and the gas or air uptake should not be less than 5 m 50 or thereabouts, as only a regenerator and uptake having a height greater than this will be able to supply the hydrostatic pressure sufficient to impress upon the air in the heads a velocity of 18 m per second.

The fundamental points of the mechanics of the circulation of the gases in furnaces have now been stated. The following is a brief summary of the principles involved:

The chambers of furnaces filled with hot gases create in their upper parts a positive hydrostatic pressure which expends itself or is absorbed in impressing upon the hot gases the velocity necessary for their circulation in the furnace.

There are two causes which act to retard or absorb the velocity of the hot gases:

1. The friction of the gaseous currents against the walls of the chambers and flues, which has not been considered in these computations, as sufficient data have not been available to establish the value of the coefficient.(1)

2. The effect of changes in the direction of flow.

In the computations made for ventilating ducts it is considered that a change in direction of 90° introduces a resistance equal to the velocity head, or that all of the velocity of the flowing gases will be lost or absorbed. If this is correct a change in direction of 180° may be considered as equivalent to two changes in direction of 90°, that is to say, it may be assumed that the resistance offered will be equal to twice the velocity head. A change in direction of 45° may be assumed as offering a resistance equivalent to 0.6 of the velocity head.(2)

The following method is employed in approximating the resistance occasioned by changes in the velocity of the gases by reason of their passing from a larger to a smaller or a smaller to a larger flue or passage.

If a gas flows from a small channel into a larger channel or chamber there will be a change in velocity from V_{max} to V_{min}, and there will be no loss of head; but, when the gas passes from a larger channel into a smaller channel a sufficient velocity head will be required to impress upon the gas the increase in velocity required.

The computation of the resistance offered by the checkerwork of a regenerator is given below as an example. This checkerwork is composed of vertical channels, those in one tier being 120 mm × 60 mm and in the alternate tiers 60 mm × 60 mm.

The velocity of the gas in the smaller channels is V_{max} and in the larger channels V_{min}, so that

$$V_{min} = \frac{V_{max}}{2}.$$

(1) This lack has recently been supplied, in part by W. A. Mojarow, in a paper on the "Friction of gases in brickwork flues" in the *Rev. de la Société russe de Métallurgie, 1913, No. 3, pp. 325–370; Revue de Métallurgie, XI bis, May, 1914, p. 320. Appendix IV.*

(2) Considerable data covering the flow of air may be found in, *Le Chauffage Industrielle by H. Le Chatelier, p. 477 et seq. D. Murgue "On the loss of head in Air Currents in Underground Workings. Trans. A.I.M.E., 1893, Vol. XXIII. Appendix V.* (*Notes et Formules p. 1090 et seq.*)

When the gas passes from a larger channel into a smaller channel the velocity changes from V_{min} to V_{max}. If the velocity V_{min} corresponds to a velocity head h_{min} and V_{max} to a velocity head h_{max}, then to produce the increase in velocity from V_{min} to V_{max} a velocity head will be required equal to $h_{max}-h_{min}$.

This quantity represents the resistance, or the velocity head required by one tier of the checkerwork.

If there are N tiers, their total resistance or velocity head required will be

$$\delta = N(h_{max}-h_{min})\Delta_t,$$

in which Δ_t is the weight of 1 cu m of the gas in motion at the temperature t.

B. What will be the velocity head required to overcome the resistance offered by the changes of velocity of the air in passing through a checkwork 6 m in height (52 tiers)? The average temperature of the air is

$$\frac{50+1050}{2}=550°.$$

The velocities are $V_{min}=0$ m 50 per second, $V_{max}=1$ m 00 per second.

The weight of 1 cu m of air at a temperature of 550° is

$$\Delta_{500}=\frac{1.29}{1+\frac{550}{273}}=0 \text{ kg } 43.$$

$V_{min}=0$ m 50 per second corresponding to $h_{min}=0.01274$,
$V_{max}=1$ m 00 per second corresponding to $h_{max}=0.05097$,

from which

$$h_{max}-h_{min}=0 \text{ m } 03823;$$

$$\delta_{regen}=0.03823\times 52\times 0.43=0 \text{ mm } 85 \text{ of water.}$$

And therefore the resistance due to changes of velocity in the checkerwork is equal to 0 mm 85 of water column, while the regenerator 6 m in height produces a hydrostatic pressure of

$$\Delta=6(1.29-0.43)=5 \text{ mm } 16 \text{ of water.}$$

So that, in the regenerator chamber, neglecting the resistance due to surface friction, the pressure of the air will increase to

$$5.16-0.85=4 \text{ mm } 31 \text{ of water.}$$

The foregoing considerations being kept in mind, the working conditions within the open-hearth furnace may be analyzed.

Fig. 23 shows a model reproducing the longitudinal vertical section of an open-hearth furnace. This model is placed between two sheets of glass and immersed in a glass tank filled with water; colored kerosene is circulated through the model. The mechanics of the circulation of the gases within this type of furnace are clearly shown: the regenerators by heating the air and the gas produce a positive hydrostatic pressure. It is this pressure which impresses their velocities upon the gases in motion; it is completely absorbed in overcoming the friction in the uptakes, etc., and in producing the velocity with which the gases enter the heating chamber of the furnace. In the heating chamber, the gases have a very small velocity, and their hydrostatic pressure is, on the average, equal to the pressure of the atmosphere, which must be the case, by reason of the number of working doors and openings.

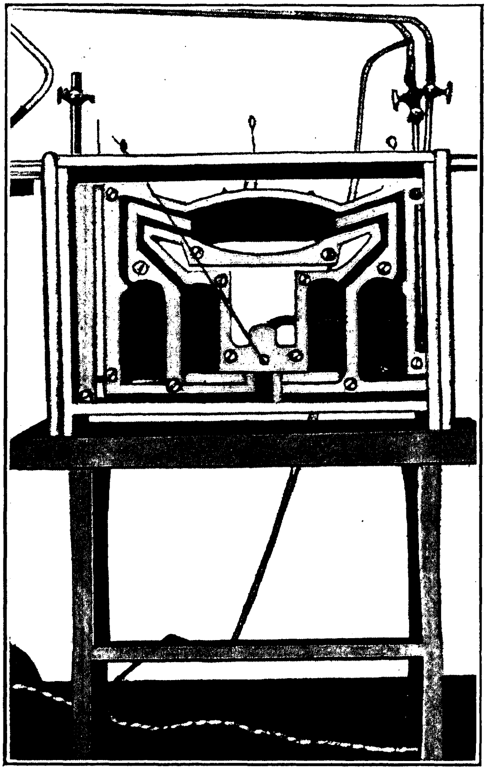


FIG. 23.

The descent of the light gas through the uptakes and regenerators, and its passage through the flues and reversing valves to the base of the chimney, commences at the heating chamber where the hydrostatic pressure is equal to that of the atmosphere. In order to draw the light gases downward to the bottom of the regenerator chamber, it is necessary to establish a negative hydrostatic pressure or draft depression at that level.

This is the function of the chimney, which, therefore, has a double purpose:

1. To draw the hot gases down to the bottom of the regenerator.
2. To impress upon them the necessary velocity and to overcome the losses of head due to changes in the direction of flow, changes in velocity and the friction against the walls of the flues, etc., in this portion of the furnace.

The computations for reverberatory furnaces, based upon the laws of hydraulics, were included, some time ago, among the obligatory exercises for the students in the metallurgical section of the Polytechnic Institute of Petrograd; as a result of this, a very large number of these furnaces in the workshops of Russia have been recalculated. Ordinarily the problem, as given, involved the computation for furnaces actually in use within the different plants, based upon the daily consumption of fuel. The volume of producer gas being determined, the volume of the products of combustion was calculated. This was usually based upon a secondary air supply one and one-half times as great as the theoretical air supply. The analysis of these furnaces actually in service has shown the following:

1. That open-hearth furnaces of defective construction were subject to a rapid filling up of the regenerators, and for this reason served only during comparatively short campaigns; the velocities were too great, and the hydrostatic pressure of the hot air and gases was barely sufficient to produce the necessary velocity. Sometimes these computations showed negative results, that is to say, the calculations showed that forced draft was necessary for operation of the furnace. But, nevertheless, the furnace worked without forced draft, though in a defective manner which caused trouble. It follows, therefore, that these methods of computation have in themselves a certain factor of safety.

2. The gas regenerators and the air regenerators should receive different volumes of the waste gases. In computing these two quantities of gas, it was found that the draft depressions necessary

to cause these volumes of the gases to pass into their corresponding regenerators were, in a great many furnaces, nearly of equal value. On the contrary, for American furnaces, where the air passes through the port with a velocity of about 8 m per second or a little more, and the gas attains a velocity of about 58 to 60 m per second, the chimney drafts required for the regenerators for air and for gas have very different values.

This, therefore, explains *the necessity of having different dampers or regulating valves for controlling the draft upon the regenerators for gas and those for air.*

In a general way all the computations made have confirmed this conclusion.

3. Occasionally, but very rarely, there has been found an open-hearth furnace in which the air and the gases had a hydrostatic pressure which was greater than was necessary (Lysva Works, Oural). This is due to the low velocities in the furnace and to the large size of the reversing valves employed.[1]

When furnaces of this kind are new it is necessary to work with the air and gas regulating valves partially closed. After the furnace has been in use for some time the cinders and dust commence to obstruct the checkerwork, and the supplementary resistance due to the regulating valves may be decreased. This permits the furnace to be operated with the regenerators badly obstructed.

This also explains why, in certain works, the checkerwork has to be renewed after one hundred to five hundred heats, while in other plants, the regenerators last six months or more. When there is an excess of hydrostatic pressure of the gases, there is the possibility of controlling the operation of the furnace, and of utilizing the air and gas valves as rheostats, which permits the prolonging of the campaign of the furnace. Therefore, the computation of *the hydraulics of the gas* in the open-hearth furnace supplies the criteria for their construction. These calculations make it unnecessary to have extensive tabulations of the dimensions and proportions of actual furnaces, with which so many metallurgists burden their memories and notebooks.

[1] Strangulation of the furnace due to the small size of the reversing valves is one of the principal defects in reversing furnaces.

VII. THE APPLICATION TO HOT GASES OF THE THEORY OF HYDRAULIC FLOW OVER THICK-CRESTED WEIRS.

In a large reservoir from which there is a flow over a weir of Q cu m of water per second there will be established at a constant height H above the sill of the weir, a water surface which supplies the hydraulic gradient necessary for the flow (Fig. 24). The

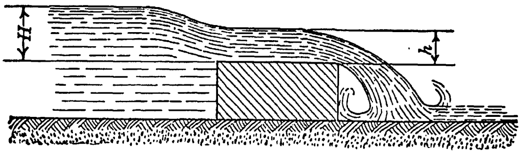


FIG. 24.

depth of the water flowing over the weir will be determined by the formula:

$$h=\tfrac{2}{3}H.$$

There is also the formula

$$Q=0.35b\sqrt{2gH^3},$$

from which

$$h=0.50\sqrt[3]{\frac{Q^2}{b^2}}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (D)$$

in which b signifies the length of the weir.

This formula makes possible the determination of the depth of the channel connecting two basins of water, and applies equally to the depth of all streams whose bed presents a grade sufficient to compensate for the inevitable friction losses of flow.

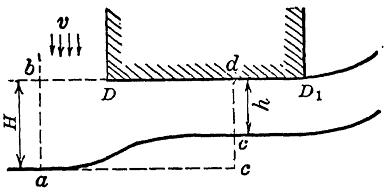


FIG. 25.

Each reverberatory furnace having a horizontal roof forms the bed of an inverted stream, in which there flows, not water, but hot gases (Fig. 25). Reverberatory furnaces, in order that they may function properly, must be so designed as to take account of the following facts:

All of the material which is to be reheated in the furnace is placed upon its hearth. Much heat will be lost in the reheating of this material, as well as by radiation from the walls. The furnace must, therefore, be strongly heated by the hot gases; and this is not possible unless the nappe or free lower surface of the current of hot gases is in contact with the hearth of the furnace, or, as it is expressed in the shop, unless the flame *licks the hearth.*

This fact supplies the method by which reverberatory furnaces may be proportioned by computation. It is evident that, in furnaces working in the most satisfactory manner, the depth of the inverted gaseous stream must be equal to the height of the roof of the furnace above the hearth. Referring to Fig. 25, h represents this height.

The formula giving the depth of the gaseous stream has been established in a brilliant manner by J. C. Yesmann,[1] to whom we are indebted for the mathematical study of this case. In his work,[2] it is shown that the formula (D) cited for water, becomes for gases

$$h_t = A\sqrt[3]{\frac{Q_t^2}{B^2 \cdot t}}, \quad \ldots \ldots \quad (E)^{(3)}$$

in which h_t represents the depth or thickness of the layer of gas in motion; Q_t, the volume of gas flowing in cubic meters at the temperature t; B, the length of the weir over which the gas flows, the *inverted weir* (the width of the furnace).

The coefficient A is different for each value of h_t and the length B of the inverted weir. The values of these coefficients are given in the following table: [4]

$h_t=$	0.30			0.50			0.75			1.00		
$B=$	1.00	2.00	5.00	1.00	2.00	5.00	1.00	2.00	5.00	1.00	2.00	5.00
$A=$	3.42	3.54	3.62	3.29	3.46	3.57	3.13	3.37	3.54	2.97	3.28	3.53

According to this, in making the computation for a reverberatory furnace it is considered as an *inverted weir* of which the depth above the crest is determined by Yesmann's formula.

For the motion of the gases flowing through horizontal flues and under the straight horizontal roofs of continuous ingot heating furnaces, this formula will give results which very closely approach those obtained in practice. For ascending roofs, the actual thickness of the layer of gas will be less than the value obtained

(1) *Revue de la Société russe de Métallurgie, 1910, pp. 319–343, and Ann. de l'Inst. Polyt. de Petrograd, 1910.*

(2) *Refer to Appendix I.*

(3) For this formula in English units refer to p. 193.

(4) For coefficients in English units refer to p. 257.

by the Yesmann formula. For descending roofs, it will be greater. In his work, Professor Yesmann has given the mathematical expression for the first case, but he has introduced certain coefficients of friction for which no accurate data exist at present.

In what measure is actual usage found to accord with this formula. For verifying Yesmann's formula, a test furnace has been built at the Polytechnic Institute of Petrograd; but during the time that it has been in service it has been impossible to obtain a gaseous current with a lower free surface or nappe which

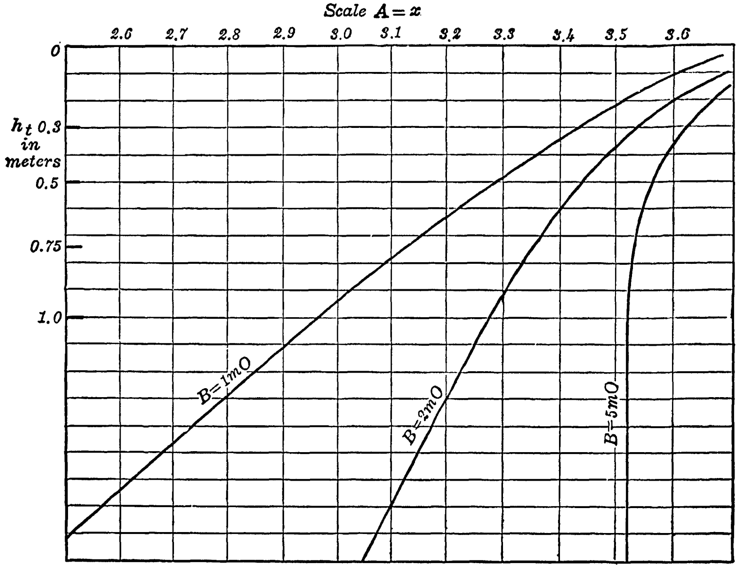


FIG. 26.—Values of Coefficient A.

was free from eddies. In this current of gases the surface is mixing with air. For this reason the measurement of the thickness of the gaseous stream is difficult. It is hoped that this difficulty may be overcome, and that it will be possible to effect the verification of the formula for the flow of gases under inverted weirs by accurate methods. In the meantime, Yesmann's formula has been tested by application to furnaces which were in use. The tests have been based upon the assumption that since the furnace works well and the shop is satisfied, the hot gases which flow in the inverted channel must lick the hearth of the furnace in a satisfactory manner.

The reverberatory furnace itself forms a true inverted channel for the flow of the gases. In analyzing each of these furnaces, the research has been concerned with the principal dimensions. It has been found in the course of this work that, in many furnaces, the dimensions which seemed to be the least important have the most important effect, and conversely, that those to which great importance had been attached were of no significance.

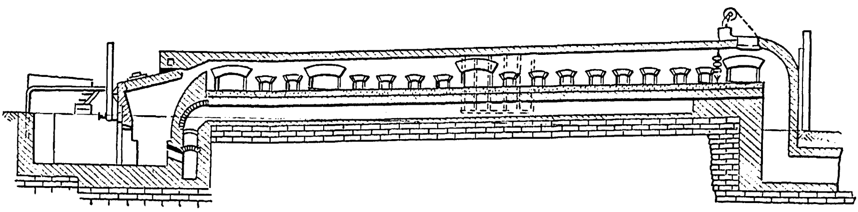


Fig. 27.—Furnace N.

It is also clearly apparent that these methods make it possible to regulate accurately the quantity of fuel consumed and diminish it accordingly, in many cases, by slight corrections applied to the furnace.

As an example of the method of computation and of its impor-

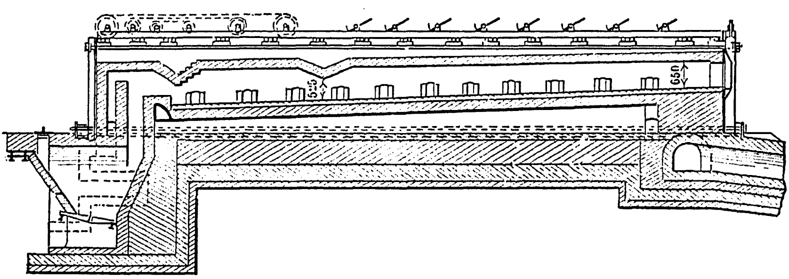


Fig. 28.—Lyswa Furnace.

tance, the calculations are presented for a continuous heating furnace in the N[1] works, using a very good grade of English coal, and a similar furnace at the Lyswa works (Oural), using a coal from the Kisélow mines, having 25 per cent of cinders or ash and 3 per cent of sulphur. The defects in these furnaces and the methods used in improving them were plainly shown by these computations. (*Refer to Figs.* 27 *and* 28.) *Computations tabulated on page* 45.)

It may be seen, from the computations for the furnace *N*

[1] A large Russian Works.

(Fig. 27) that the flame follows the hearth of the furnace, if the furnace does not hold any ingots, and this is very nearly what may be actually seen in this furnace.

In the furnace at the Lyswa works (Fig. 28) there has been found to be a complete identity between the actual and calculated heights of the strangulating ridges in the roof above the hearth of the furnace (calculated value 539 mm; actual dimension 525 mm), but the dimension computed for the height at waste gas ports was considerably less than that in the furnace. When these conclusion had been reached, A. J. Onoufrowitch, managing director of the works in the Lyswa district, was asked the following questions regarding the details of the work of this furnace:

1. Have the strangulations in the roof any effect in forcing the hot gases down on the hearth of the furnace?

2. Do the gases seem to remain at a short distance above the hearth of the furnace at the waste gas end?

A reply was received giving the following facts concerning the record of this furnace:

The furnace had been originally constructed without strangulations and worked very badly. The managing director had then dropped portions of the roof a sufficient distance to bring the hot gases down to the hearth. He stated that he had been led by trial to give this lowered section a height of 525 mm (Fig. 28), the thickness of the layer of hot gases in the furnace chamber.

To the second question he replied that the furnace had not worked well at the end connected with the chimney (the rear end of the furnace), the hot gases seeking the roof. He proposed to correct this condition later by the removal of the present roof, replacing it by a horizontal roof 150 mm lower; h_{700} would then be equal to 500 mm.

This shows that Yesmann's formula has received a complete confirmation in the computations verifying these two furnaces. It is accordingly permissible to make some other comparisons.

The "N" furnace heated $\frac{3800}{2900}=1.31$ times the weight of ingots, but it consumed per second $\frac{2.16}{0.78}=2.77$[1] times the volume of fuel burned at Lyswa; that is to say, per unit weight of ingots, the consumption of fuel was $\frac{2.77}{1.31}=2.11$ times greater in the

	N Furnace. Fig. 27	Lyswa Furnace. Fig. 28
Dimensions of Hearth	$16 \times 2.1 = 33$ m² 6	$13 \times 2 = 26$ m²
Production per 24 hours	3,800 pouds 62,300 kg	2,900 pouds 47,400 kg
Per square meter	1,860 kg	1,850 kg
Coal consumed per 24 hours	800 pouds 13,000 kg	409 pouds ½ 6,700 kg
Coal consumed per second	0 kg 154	0 kg 079
Ratio between weight of coal and weight of ingots	21.1 per cent	16.8 per cent
Gas volume per kilogram of coal burned with 50 per cent excess air supply	14 m³ 04	9 m³ 89
Gas, volume burned per second reduced to 0°	$Q_0 = 2$ m³ 16	$Q_0 = 0$ m³ 78
Gas, volume burned per second at $t = 1200°$	....	$Q_{1200} = 4$ m³ 2
Height of roof above hearth at the right of the strangulation, computed by Yesmann's formula for a temperature of 1200°	....	$h_{12000} = 0$ m 530
Effective height	....	*0 m 525*
Gas, volume per second at $t = 700°$.	$Q_{700} = 7$ m³ 697	$Q_{700} = 2$ m³ 780
Height of roof at chimney or waste gas opening by Yesmann's formula $t = 700°$	$h_{700} = 0$ m 91	$h_{700} = 0$ m 479
Effective height	$h_{700} = 0$ *m 90*	$h_{700} = 0$ *m 650*

"N" works. To what may this greater fuel consumption be attributed. Evidently to the greater height between the roof and the hearth of these furnaces.

Why was it found necessary at the "N" works to raise the roof to a certain height above the hearth, while at the Lyswa works it was found necessary to lower the roof to accomplish the same result? An examination of the design of the "N" furnace (Fig. 27) leads to a negative conclusion. Indeed, according to Yesmann's formula a furnace will require less combustible when the roof is closest to the hearth. Accordingly the roof should be brought down as low as possible.

The hot gases consist of a mixture of air, combustible gases and the products of their reaction upon each other. The combustion takes place while the hot gases are passing through the furnace;

(1) The *quality* of the coal is not considered in this computation.

the heat released, therefore, serves to heat the ingots and to maintain the hot gases at a high temperature. If the flames, that is to say, the mixture of combustible gases and air, in which the reaction is taking place, impinge against the cold ingots, a deposit of soot will be formed, and carbon, carbon monoxide and hydrogen will pass off unburned through the waste gas port. It is evident that such a method is not advantageous, as it is necessary for combustion to occur in the heating chamber in order that the heat may be utilized. From this effect it is evident that the flaming gases should not be cooled while the reaction of combustion is taking place or until it has been completed. On the contrary, if the reactions of combustion are very nearly completed, the hot gases produced may be cooled very rapidly.

The Lyswa furnace has a large firebox, allowing the producer gas to mix perfectly with the secondary air. On the other hand, in the " N " furnace there is a very small firebox with very poor mixing of the producer gas and the secondary air. The flaming gases from this firebox are forced directly down upon the ingots without any precautions; the production of soot is inevitable. It is for this reason that these gases sweep along under the roof, from the strangulation which is located immediately over the bridge wall, to obtain a better mixture; and from this point on the velocity of the gases is increased, thus diminishing the thickness of the layer of gases below the arch. As a result of these conditions, the furnace works very poorly.(1)

A furnace at the *Zavertsé* works, where the height from the hearth to the roof is only 350 mm, will serve as an example of a continuous reheating furnace with a very low roof. This furnace gives off enormous quantities of smoke when coal is placed in the firebox. There is no flame in the heating chamber and the heat there is not sufficient to ignite the gaseous products which are given off in great volume by the fresh charge of coal as it is distilled on the fire.

Continuous reheating furnaces, therefore, will not work well with low roofs; on the other hand, the fuel consumption increases very rapidly as the roof is raised. From Yesmann's formula it can be deduced that

$$Q_t = B\sqrt{\left(\frac{h_t}{A}\right)^3}\cdot t.$$

(1) The formation of soot by the cooling down of the flaming gases is particularly noticeable in open-hearth furnaces working with a charge of cold pig

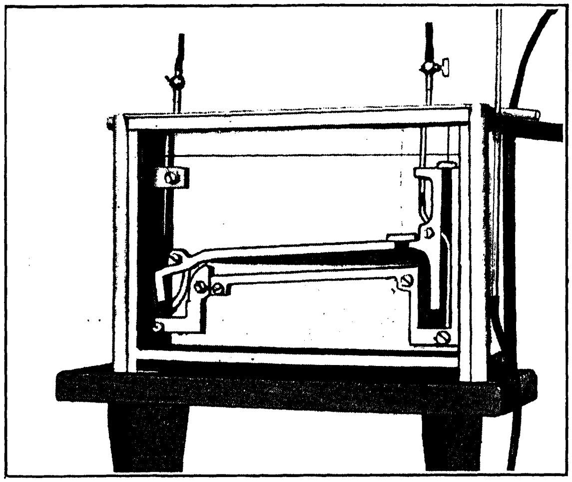


Fig. 29.

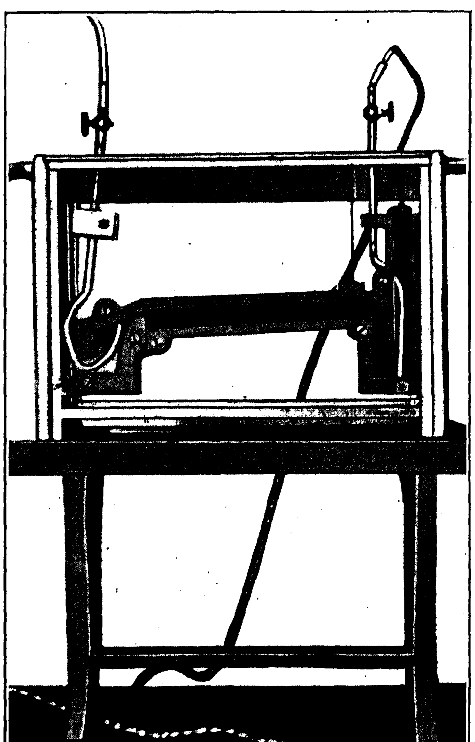


Fig. 30.

Moreover, according to this principle, the use of a roof sloping upward tends to increase the velocity of flow of the hot gases and in this manner to decrease the thickness or depth of their stream. They are thus kept away from the hearth, and the fuel consumption is increased, as it is necessary to increase the quantity of gases flowing in order to force them to lick the hearth. At the same time, these statements should not lead to the conclusion that contemporary continuous reheating furnaces cannot be reconstructed so that they will work in a satisfactory manner.

Figs. 29 and 30 show a model of the " N " furnace immersed in water; a stream of colored kerosene flows into the heating chamber from the firebox. The two half-tones show the outflow of the waste gases through openings pierced in the hearth of the furnace. Fig. 29 shows that with this method of outflow of the waste gases there remains upon the sole or hearth of the furnace a layer of cold air. The rise of the hearth forms a *pocket* from which the colder and heavier gases cannot flow by gravity. This is a very serious defect of continuous reheating furnaces having the roof and the hearth rising from the firebox end toward the charging end. The ingots or billets heating on the hearth of the furnace chill the gases, and these chilled gases cannot by any possibility reach the chimney, since they are heavier than the hot gases.

In Fig. 30 the current of colored kerosene has been increased, but it has not been possible to force out the water from the pocket on the hearth, as the pressure has not yet become higher than the water pressure, that is to say, the atmospheric pressure. When this point is reached a portion of the kerosene escapes through the interstice or small opening between the sheets of glass forming the sides of the model. It is evident that this is not a desirable manner of working as the outflow of the chilled gases from the hearth will take place through the working doors, instead of through the waste gas opening to the chimney.

In practice, therefore, a furnace of this design will work under the conditions shown in Fig. 29, and the ingots or billets will not be well heated except on their upper portion, since their lower portion will rest in the layer of chilled gas on the hearth of the furnace and the cold air drawn in through the working doors. This prolongs the time required to heat the ingots, increases the defects in the metal, etc.

The problem of the rational construction of the continuous

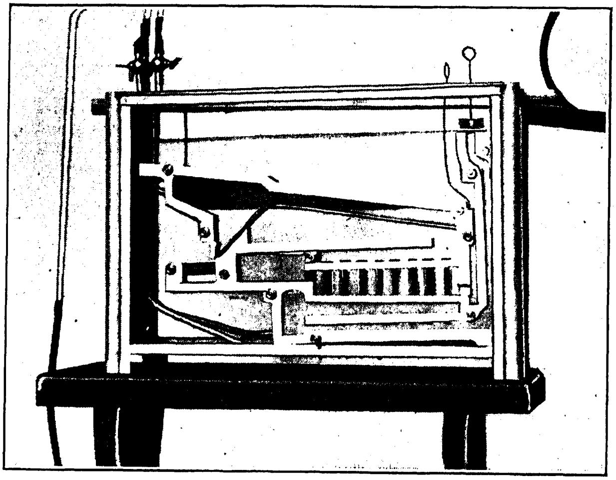


Fig. 31.

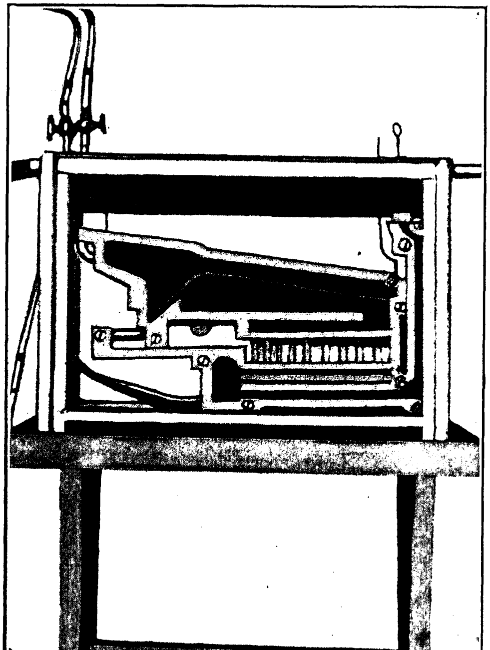


Fig. 32.

reheating furnace for ingots and billets was solved in a very satisfactory manner by the designer of the American Morgan furnace, which has a descending roof (Fig. 31, 32 and 33). It is rather interesting to know that Morgan was not led to construct his furnace in this manner by consideration of the circulation of the heated gases, but simply because it was found more convenient to push the ingots up a sloping skid. The ingots remained in contact with one another when heated on skids with an upward slope, a condition which was difficult to maintain when the skids were level or sloped downward. It has been found that this construction with a roof sloping downward from the firebox end of the furnace to the end at which the waste gases are taken off and where the cold ingots enter the furnace is the very best solution of the continuous reheating furnace problem.

It may be seen in Fig. 33 that producer gas at a high temperature (1000° or thereabouts) and secondary air heated in recuperators enter the combustion chamber. Here the mixture is burned, and the products of combustion, as they give up their heat, pass under the descending roof between the ingots resting on the water-cooled pipe skids, and descend into the recuperator tubes.

In this furnace, the ingots are literally plunged into the hot gases. The model which has been made shows, when immersed in water and traversed by the stream of colored kerosene, that there will be no drawing in of cold air at the working doors, and that there will be no pockets of chilled and stagnant gases. On the contrary, the chilled gases fall upon a hearth a sufficient distance below the skids to insure the immersion of the ingots in a layer of hot gases, and from this hearth the colder gases drain into the recuperator. The photograph, Fig. 31, was taken while the model was being filled with kerosene. It shows extremely well how the hot incandescent gases from the flame fill the upper portion of the heating chamber in a horizontal layer which increases in thickness, and how these gases, as they cool, drop lower by reason of their increase in weight. The cool gases fall away from the ingots as they give up their heat to them.

The idea of the descending roof for *continuous reheating furnaces* has come into general use in many plants. In central Russia there may still be found a number of older furnaces which have roofs ascending from the firebox end, and some of which also have a downward inclination of the roof at the opposite end.

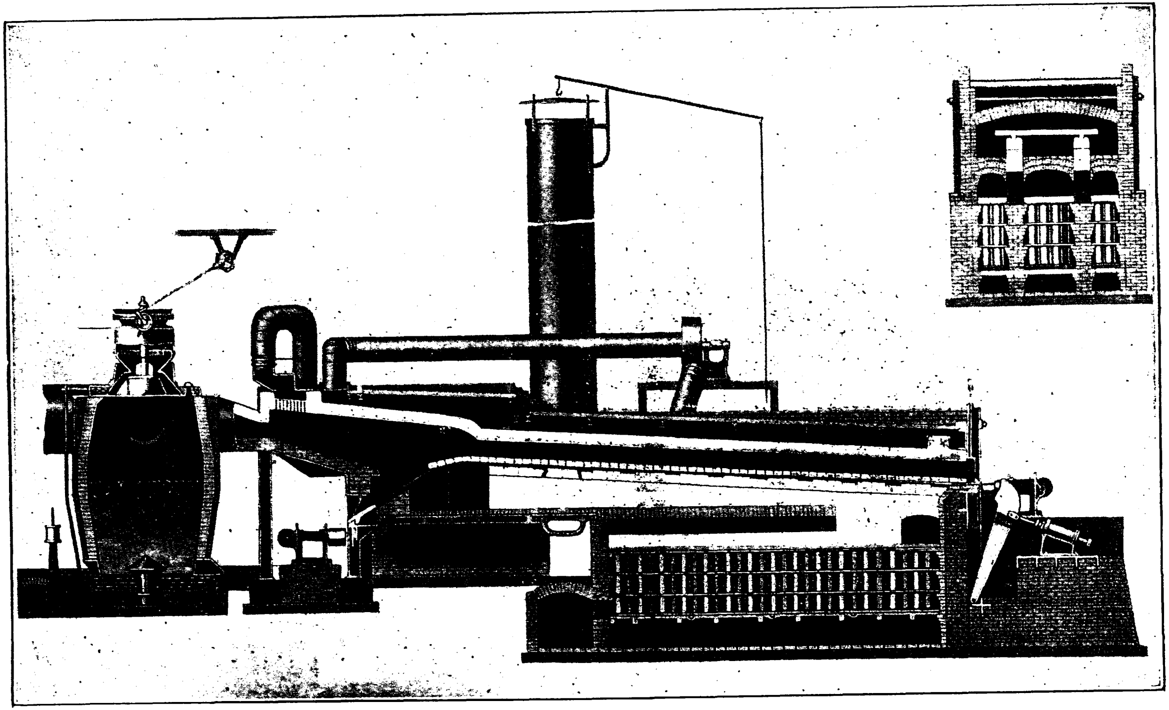


FIG. 33.

The type of roof which should be chosen—ascending, descending or horizontal—will depend upon the result which is desired from the furnace.

1. A horizontal roof will give a high temperature at the firebox end of the hearth and a low temperature at the rear end of the furnace.

2. If a higher temperature is desired near the firebox end of the furnace, and a diminishing temperature toward the rear end, the descending roof should be used. The firebox end will form a high chamber, in which the flaming gases will remain for a short period of time and where they will complete their combustion. The hot products of combustion will drop below the hotter flame, and as they gradually give off their heat and become cooler, will descend in horizontal layers. The coolest gases will pass from the heating chamber by means of the waste gas port in the hearth of the furnace.

3. In certain cases it is necessary to maintain a uniform temperature throughout the length of the furnace; it is then necessary to use an ascending roof.

In the author's personal practice the following case came up for analysis: two steel boxes for the annealing of steel sheets were heated upon the hearth of a furnace. The first box heated much more rapidly than the second box, and the annealing of the sheets was accomplished at a temperature which was much too high. This peculiarity of the furnace caused considerable trouble to the plant, and it was impossible to hold back the annealing of the first box so that the annealing of the second box would be completed at the same time. The furnace was changed so that the annealing of both boxes was completed at the same time and temperature, by replacing the descending roof of the furnace by a horizontal roof. This was done because the author had a groundless fear, at this time, of making an ascending roof, which would have carried the total mass of the hot gases to the rear end where they would have distributed themselves very effectively and uniformly throughout the full length of the furnace.

VIII. VELOCITY IN INVERTED WEIRS. RATIONAL CONSTRUCTION OF FURNACES OF THIS TYPE

Yesmann's formula provides a method of computing the normal velocity of the current of hot gases in an inverted weir according to the depth of the layer or stream of gases. It is

$$h_t = A\sqrt[3]{\frac{Q^2}{B^2 t}} \quad \text{or} \quad Q = B\sqrt{\frac{h_t^3 \cdot t}{A^3}}$$

but $Q = \omega \cdot v = B \cdot h_t \cdot v,$

in which ω = the area of the cross-section;

B = the length of the weir or width of the furnace;

h_t = the depth or thickness of the stream of gases under the inverted weir or the head at the crest;

t = the temperature of the hot gases;

Q = the volume of gases flowing;

A = a coefficient.

From which

$$B \cdot h_t \cdot v = B\sqrt{\frac{h_t^3 \cdot t}{A^3}} \quad \text{and} \quad v = \sqrt{\frac{h_t \cdot t}{A^3}}$$

NORMAL VELOCITY OF GAS FLOW UNDER INVERTED WEIR

B = 1 m 00 = width of furnace, corresponding to length of weir.

h_t =	1 m 00	0 m 75	0 m 50	0 m 30
A =	2.97	3.13	3.29	3.42
t	Velocity, Meters per Second = v			
500°	4.368	3.497	2.650	1.936
600°	4.785	3.830	2.913	2.121
700°	5.168	4.138	3.135	2.291
800°	5.526	4.423	3.352	2.450
900°	5.861	4.691	3.555	2.598
1000°	6.178	4.945	3.747	2.739
1100°	6.479	5.186	3.930	2.872
1200°	6.768	5.417	4.105	3.000
1300°	7.046	5.638	4.272	3.122
1400°	7.310	5.852	4.434	3.240
1500°	7.566	6.056	4.589	3.354
1600°	7.814	6.256	4.739	3.464
1700°	8.055	6.458	4.885	3.571
1800°	8.288	6.634	5.044	3.674
Q_t =	1.00v	0.75v	0.50v	0.30v

(For the values of A refer to page 41.)

NORMAL VELOCITY OF GAS FLOW UNDER INVERTED WEIR

B=2 m 00=width of furnace, corresponding to length of weir.

h_t=	1 m 00	0 m 75	0 m 50	0 m 30
A=	3.28	3.37	3.46	3.54
t	Velocity, Meters per Second = v			
500°	3.764	3.131	2.462	1.839
600°	4.123	3.428	2.691	2.014
700°	4.454	3.704	2.823	2.176
800°	4.761	3.959	3.108	2.326
900°	5.050	4.199	3.296	2.467
1000°	5.323	4.427	3.474	2.600
1100°	5.584	4.643	3.677	2.727
1200°	5.832	4.849	3.806	2.849
1300°	6.069	5.047	3.961	2.965
1400°	6.299	5.238	4.140	3.077
1500°	6.520	5.422	4.255	3.185
1600°	6.736	5.599	4.394	3.289
1700°	6.941	5.772	4.530	3.390
1800°	7.142	5.939	4.661	3.489
Q_t=	2.00v	1.50v	1.00v	0.60v

These tables shows that, for a horizontal roof, the velocity of the hot gases ordinarily varies from 1 m 94 to 8 m 29 per second. And, as in reverberatory furnaces, it is necessary that the hot gases should remain in the furnace a sufficient time to give up heat to the material being heated, in certain industrial heating applications there will be found to be a relationship between the circulation velocity of the hot gases and the application. But it is evident, on the other hand, that there should be no necessity for accelerating the velocity of the hot gases, because this would lead to the construction of extremely long furnaces.

A number of attempts have been made to determine the time which is required for the transmission of the heat in the hot gases to the materials being heated and to the walls of the furnace. Observations and computations have led to the establishment of two limits

NORMAL VELOCITY OF GAS FLOW UNDER INVERTED WIER

$B=5$ m 00 = width of furnace, corresponding to length of weir.

$h_t=$	1 m 00	0 m 75	0 m 50	0 m 30
$A=$	3.53	3.54	3.57	3.62
t	Velocity, Meters per Second $=v$			
500°	3.386	2.908	2.344	1.778
600°	3.709	3.185	2.568	1.948
700°	4.006	3.440	2.774	2.104
800°	4.283	3.678	2.965	2.249
900°	4.543	3.901	3.145	2.386
1000°	4.788	4.112	3.315	2.515
1100°	5.022	4.312	3.477	2.638
1200°	5.246	4.504	3.631	2.755
1300°	5.460	4.688	3.779	2.86
1400°	5.666	4.865	3.923	2.976
1500°	5.864	5.036	4.060	3.080
1600°	6.057	5.201	4.193	3.181
1700°	6.243	5.361	4.322	3.279
1800°	6.424	5.516	4.448	3.374
$Q_t=$	$5.00v$	$3.75v$	$2.50v$	$1.50v$

for the drop in temperature of the gases per second. The method of determining the rate of the drop in temperature is as follows:

The difference between the theoretical calorific intensity, or temperature, of the combustible and the temperature of the products of combustion leaving the heated zone is divided by the time in seconds which the products of combustion remain in the heated zone. This gives the average drop in temperature of the gases per second.

For open-hearth furnaces this drop in temperature has been found to be from 200° to 250° and for large brick kilns with a spherical roof—downdraft—from 70° to 80° per second.

Therefore, taking 1500° as the temperature produced by the combustion of a coal with an air supply 1.50 times the theoretical requirements, and 600° as the temperature of the products of combustion leaving the heating chamber, there is a temperature drop, or cooling of the gases in the heating chamber, of 900°, that is to say, the gases remain in the chamber only four or five seconds.

If these data are considered in connection with an ordinary type of continuous heating furnace for ingots, in which the lower free surface of the gases is in contact with the hearth of the furnace, and the height from the hearth to the roof is 0 m 75, the velocity of the gases entering the heating chamber will be 5 m 41 per second and of those leaving the chamber 3 m 42 per second. The average velocity will be

$$\frac{5.41+3.42}{2}=4 \text{ m } 41 \text{ per second;}$$

from this it may be determined that the length of such a furnace will be from 17 m 00 to 22 m 50.

Therefore, when the velocity of the current of hot gases must be limited to very small values the result will be a long heating chamber.

One of the many defects in the older types of reverberatory furnaces was the constriction of the opening over the bridge wall where the hot gases entered the furnace. This resulted in an impinging current of burning gases and the rapid cutting away of the brickwork owing to the high velocity of the gases. This high velocity was considered necessary in order to effect an intimate mixture of the combustibles and the comburent. This method of mixing is now considered undesirable and ineffective. In the later types of reverberatory furnaces it has been completely abandoned. An intimate mixture of the producer gas from the firebox and the secondary air is essential, but this mixture should not be made at the bridge wall.

The height of the opening over the bridge wall may be computed by Yesmann's formula

$$h_t = A\sqrt[3]{\frac{Q^2}{B^2 \cdot t}}.$$

IX. REVERBERATORY FURNACES CONSIDERED AS INVERTED WEIRS WITH A CISTERN OR RESERVOIR. DIMENSIONS UPON WHICH THEIR CORRECT OPERATION DEPENDS

As a general proposition, very long furnaces are not common and in many cases are undesirable. Nevertheless, it is sometimes necessary to retain the hot gases in the heating chamber as long as possible, by reducing their velocity. This effect is obtained under the crest of an inverted weir for gases, in the same manner in which it would be done on a weir carrying water, by providing

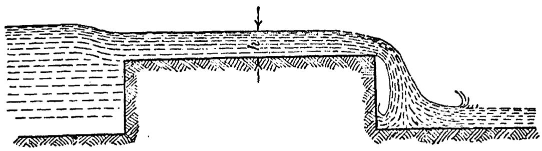


FIG. 34.

a cistern or cutting a deeper reservoir in the weir, making an upstream and a downstream crest.

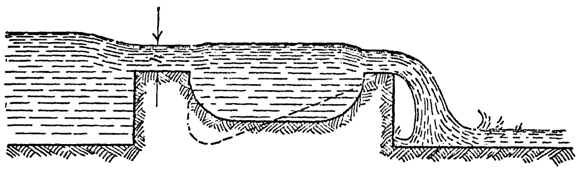


FIG. 35.

When a weir having a thick crest (Fig. 34) is transformed into a weir with a cistern or reservoir (Fig. 35), the velocity of flow of

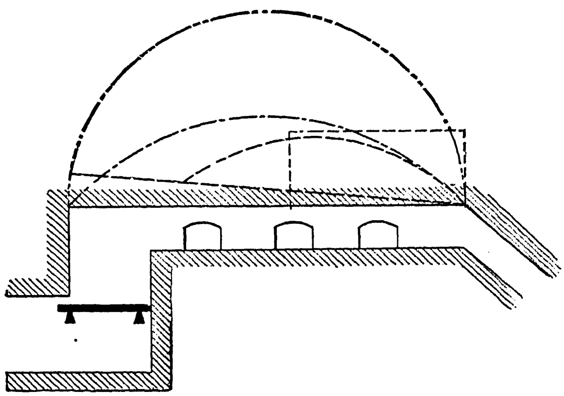


FIG. 36.

the water in the reservoir is reduced, while the level of its free surface remains the same.

In securing this result with an inverted weir for gases, all of the dimensions of the furnace heating chamber are affected (Fig. 36), but the level of the lower free surface of the stream of gases remains at the hearth of the furnace, and the furnace continues to work well. The gases, however, remain in the heating chamber a longer time and a better transfer of their heat to the material being heated is obtained. The temperature of the gases leaving the furnace and, accordingly, the temperature of the material being heated, are different.

In this manner it is not difficult to pass from the consideration of the continuous or gradual reheating furnace to that of the chamber type of furnace. Fig. 37 shows a type of furnace intermediate between the ordinary reheating furnace and the chamber

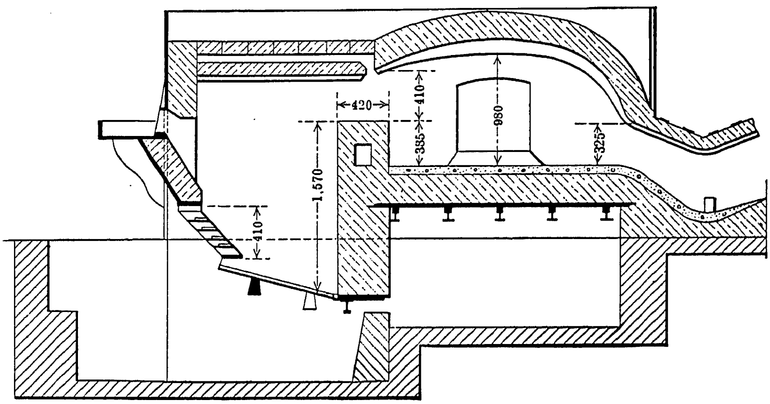


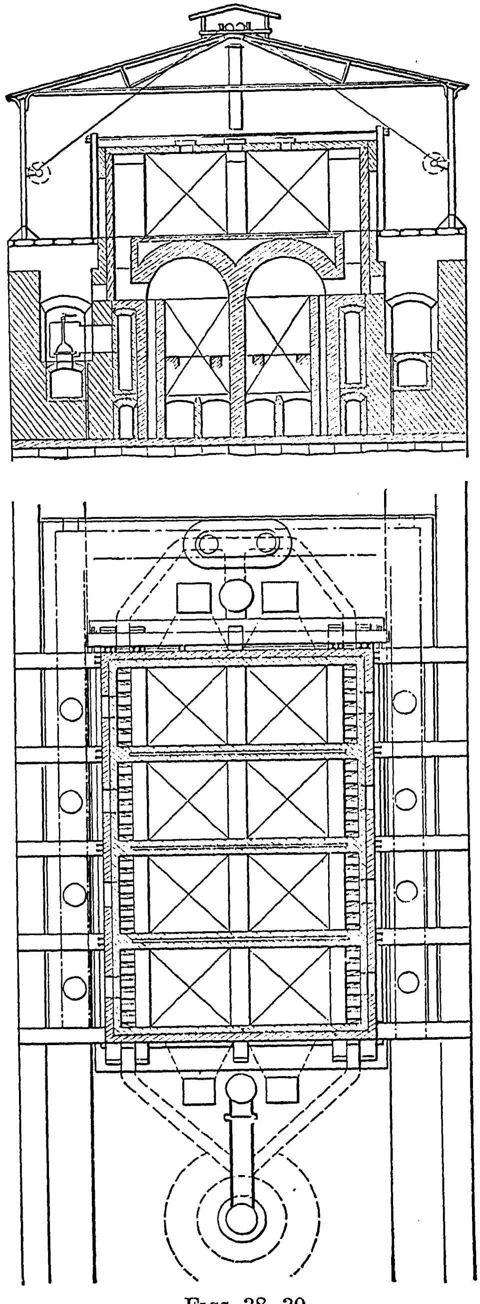
Fig. 37.

type of furnace. This is a reheating furnace at the Pouliloff works; it is very short and has a high roof permitting very large ingots to be placed upon its hearth.

Figs. 38 and 39 show the arrangement of a kiln for the burning of silica brick, designed by the author. This is a multiple-chamber furnace (Siemens system); the gas and the air enter and the products of combustion leave the heating chamber at the hearth level. The hot products of combustion rise to the roof of the chamber, and, as they gradually lose their heat, descend towards the hearth and then pass through the flues leading to the regenerators. The furnace chamber is constantly filled with flame, and the hearth is always in contact with the hot gases. The design may be modified in reconstructing existing brick kilns to suit this method of firing.

As the reader may see, the transition from the consideration of the gradual or continuous heating furnace, the simple inverted weir, to the consideration of the chamber furnace with down-draft is readily made. The chambers of these furnaces are computed upon the basic idea of the time during which it is necessary for the hot gases to remain in the chamber, and this in turn is fixed by the drop in temperature of the heated gases per second. But, in designing and constructing these furnaces, it is necessary to depend upon Yesmann's formula in all those cases where the flow of the gases in motion takes place under an inverted weir.

For example, in a furnace with working openings closed by doors, the height of the door openings should not exceed h_t. If the waste gas flues leave the fur-



FIGS. 38, 39.

naces laterally, the ports in the wall should have their sills at the hearth level, and their height should not exceed h_t. If, on the contrary, the waste gases are taken off from a chamber furnace

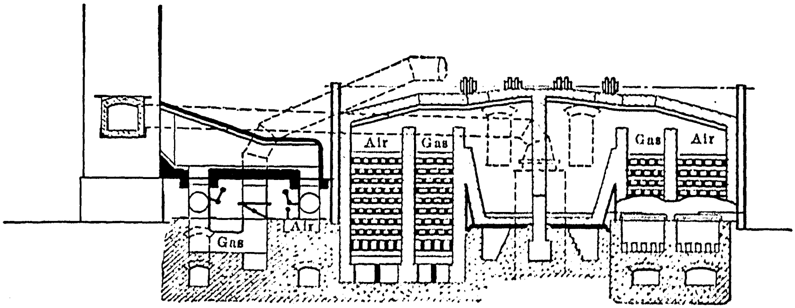


FIG. 40.

by ports having a sill at a higher level than that of the hearth of the furnace, the hearth can never be brought up to a high temperature.

As an example of the experiments made to obtain satisfactory

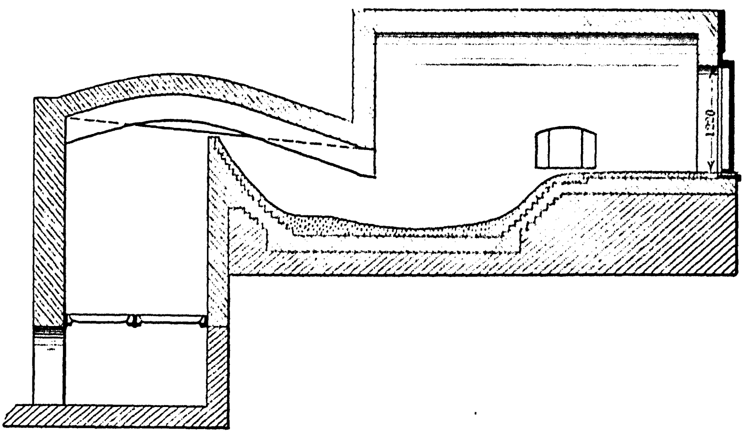


FIG. 41.

operation, without any realization of the necessary conditions, a kiln designed for the burning of silica bricks and using regenerative firing methods is shown (Fig. 40). In order to force the hot gases to act on the silica brick set on the hearth of the furnace, the designer placed a wall in the center of the heating chamber and connected the two halves of the chamber by ports in the lower portion of this wall. Nevertheless the bricks on the hearth were not sufficiently well burned. It therefore became necessary to add a supplementary waste gas flue with ports in the hearth of the heating chamber.

A very interesting type of chamber reverberatory furnace has been installed at the Lyswa works for the remelting of scrap and large broken castings. It was built by M. Onoufrowitch after an American design (Fig. 41). The rear portion of an ordinary reverberatory furnace has been replaced by a large chamber, *the waste gas flue having been left in its former place.*

Computations for this furnace may be made in the following manner: the consumption of coal is 0 kg 31 per second. Assuming that the air supply is 1.40 times the theoretical requirements of combustion, and that the temperature t is equal to 1300°, the volume of gas $Q_0 = 1$ m^3 82 per second and $Q_{1300} = 10$ m^3 48 per second. For a furnace width of 1 m 60, therefore, $h_{1300} = 1$ m 05.

That is, the normal thickness of the *layer of gas below the inverted weir* for this furnace is equal to 1050 mm and, for this furnace to work, the distance from the hearth to the top of the waste gas port should not exceed 1050 mm. Otherwise the hot gases would not touch the hearth of the furnace and it would not heat well. The vertical distance from the tapping hole to the top of the waste gas port actually is 1100 mm, which agrees very well with the calculated distance. The charging door of this furnace is very large and high, but it is hermetically closed when the furnace is in operation, and on this account it occasions no loss of the hot gases.

The volume of the heating chamber of the Onoufrowitch furnace is 11 m^3 00, of which 8 m^3 70 are comprised in the chamber. By reason of this, the gases remain in the heating chamber slightly longer than one second, and therefore the furnace gives good results.

A very interesting type of furnace is that used in the heating of steel plates. In this case the necessity of having the hot gases at the level of the hearth is still more absolute than in ordinary furnaces. In effect, the presence of air upon the hearth of the furnace causes considerable damage, as it oxidizes the iron and in this manner produces a quantity of surface defects. It is for this reason that these furnaces must be so constructed as to prevent the least possibility of free air getting in on the hearth.

These are chamber furnaces, and the height from the hearth to the roof may be great. The best furnaces are constructed with a high arch; the space immediately below it serves as a combustion chamber, and is very effective in promoting an intimate mixture of the combustible gases and the air; the combustion of the gas, therefore, takes place with very nearly the theoretical air supply.

These furnaces are operated with the doors constantly or very frequently opened. Therefore the height of the charging opening is the most important dimension and fixes the working quality of the furnace. This height must be less than the height h_t as indicated by Yesmann's formula.

The Sud-Kama works show an example of a plate-heating furnace upon the hearth of which there is no inflow of air and with which, accordingly, they have succeeded in solving the difficulty of utilizing the waste heat from the furnace in a boiler.

The data for its computation are as follows: the coal consumption per second for one chamber is 0 kg 023. The quantity of gas per second for theoretical combustion is

$$Q_0 = 4.01 \times 0.023 = 0 \text{ m}^3\ 092 \text{ per second,}$$
$$Q_{1000} = 0 \text{ m}^3\ 429 \text{ per second.}$$

The width of the charging opening is 0 m 85.

The height calculated for this opening (according to Yesmann) $h_t = h_{1000}$, is 0 m 216 and the effective value is 225 mm.

It may be seen by this that the working opening, which is closed by a door, has a height very well proportioned to the volume of the gases circulating in the heating chamber. It is for this reason that the opening of this door, which is operated by a pedal, does not cause an inrush of cold air into the furnace, as occurs in furnaces of other systems, for example, in the gas-fired furnace of the type used at the Alapayewsky works.

The following are the data for these furnaces: the quantity of wood consumed per compartment or heating chamber per second is 0 kg 0185. The volume of gas per furnace at 0°, $Q_0 = 0 \text{ m}^3\ 082$; at 1000°, $Q_{1000} = 0 \text{ m}^3\ 382$.

The height of the working door, computed according to the theory of the inverted weir $h_{1000} = 0$ m 164. Nevertheless, the charging opening is 400 mm in height. This explains why the hearth of this furnace is constantly covered with a layer of air, and why the waste gases, as analyzed by M. Asseyew, showed 3.85 per cent of CO_2 and 16.65 per cent of O_2. This was the cause of the abandonment of these furnaces at the Alapayewsky works.

The conclusion that the good or bad working qualities of plate-heating furnaces depended upon the height of the working opening was entirely unlooked for at this time. Since then this conclusion has been completely verified. There is no doubt whatever that this is only one of the numerous surprises which will be revealed by the application of the hydraulic theory to furnaces.

X. THE APPLICATION TO HOT GASES OF THE THEORY OF JETS OF WATER

If the resistance due to friction in the pipes and in the air is neglected the vertical height to which a vertical jet of water will rise will be equal to the head and will be

$$H=h=\frac{v^2}{2g}.$$

The jet from an inclined fountain describes a parabola whose middle ordinate (Fig. 42) will be

$$H=\frac{v^2 \sin^2 \delta}{2g}.$$

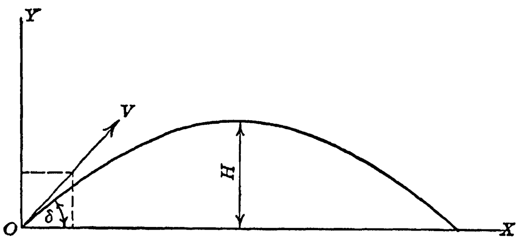


FIG. 42.

In applying these formulas to hot gases, that is to say, to a light liquid in motion within a heavy liquid, it is necessary to turn the diagram upside down, as the particles in motion are acted upon, not by their weight and gravity in a downward direction, but by the difference between the weight of the particles in motion and a corresponding volume of the medium within which their motion takes place. The resultant force will act in an upward direction, thus inverting the diagram, as is indicated in Fig. 43.

Yesmann has developed the following expression for an infinitely thin jet of hot gases, projected from the port of an open-hearth furnace:

Let v = the initial velocity of the fluid in motion;
Δ_m = the specific weight of the fluid in motion;
Δ_t = the specific weight of the fluid at rest.

The weight of each unit of volume of the fluid in motion is less than Δ_t. The upward force which acts upon each unit volume of the fluid in motion is, therefore, equal to $\Delta_t - \Delta_m$. The upward acceleration due to this force is

$$\frac{\Delta_t - \Delta_m}{\Delta_m} g$$

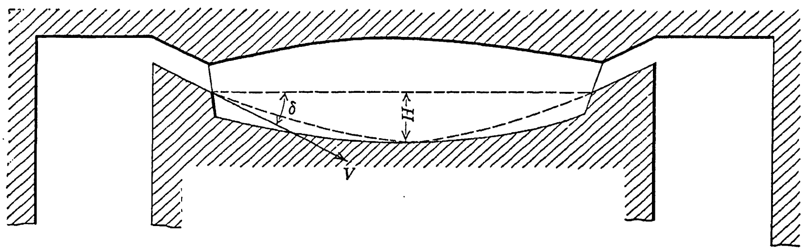


FIG. 43.

The depth to which the jet will descend for this case is

$$H = \frac{v^2 \sin^2 \delta}{2g} \times \frac{\Delta_m}{\Delta_t - \Delta_m}$$

For hot gases

$$\Delta_t = \frac{\Delta_0}{1+\alpha t_t}, \qquad \Delta_m = \frac{\Delta_0}{1+\alpha t_m}$$

from which

$$H = \frac{v^2 \sin^2 \delta}{2g} \times \frac{\dfrac{\Delta_0}{1+\alpha t_m}}{\dfrac{\Delta_0}{1+\alpha t_t} - \dfrac{\Delta_0}{1+\alpha t_m}} = \frac{v^2 \sin^2 \delta}{2g} \times \frac{1+\alpha t_t}{\alpha(t_m - t_t)}$$

$$H = \frac{v^2 \sin^2 \delta}{2g} \times \frac{\dfrac{1}{\alpha} + t_t}{t_m - t_t} = \frac{v^2 \sin^2 \delta}{2g} \times \frac{273 + t_t}{t_m - t_t} \quad . \quad . \quad . \quad (F)$$

For vertical jets, when $\delta = 90°$, this formula becomes

$$H = \frac{v^2}{2g} \times \frac{273 + t_t}{t_m - t_t}$$

When $t_i=0°$, that is to say, when the jet enters cold air, the formula becomes

$$H=\frac{v^2}{2g}\times\frac{273}{t_m}$$

and as $\frac{v^2}{2g}$ designates the head h the formula becomes

$$\frac{H}{h}=\frac{273}{t_m}$$

a proportion which has already been given on page 31.

The application of the theory of gaseous jets to the open-hearth furnace is illustrated in Fig. 23, which shows a model of an open-hearth furnace, within the heating chamber of which a current of colored kerosene is circulated. As may be seen in the cut, all of the heating chamber, except that portion occupied by the *bath*, is filled with the kerosene. Here, or more exactly, in all that portion of the heating chamber which is below the sills of the gas ports, water remains, and there is no means by which it can be forced out of the heating chamber.

The presence of the water clearly demonstrates the most important defect of the open-hearth furnace and many other similar furnaces. The hot gases enter and leave the furnace immediately below the roof, and all that portion of the heating chamber below the level of the sill of the ports and above the steel notch or tapping hole, a distance which varies in actual furnaces from 0 m 50 to 1 m 50, is outside of and below the direct current of the gases.

In the model it is clearly shown that the bath of the open-hearth furnace is in a *pocket*, from which the cooled gases cannot flow naturally and where they remain, accordingly, practically immobile. In order to expel these cooled gases from the hearth, as well as to mix the heated gases forming the flame, the gas and the air are directed downward upon the hearth, and as they enter the heating chamber at a high velocity, the phenomenon of the *gaseous jet* is utilized in forcing their current downward to a depth equal to that of the furnace below the port sills, or down to the level of the tapping hole. In European furnaces the entry velocity of the gas and air varies between 12 and 18 meters per second, while the gas velocity in some American furnaces is as high as 50 meters per second.

If the theory of the *gaseous jet* is correct, all open-hearth furnaces which operate satisfactorily should conform to it. This may be determined as follows:

The air enters the heating chamber of the open-hearth furnace preheated to a temperature of 1000° to 1100°, as does the producer gas. Their combination forms a *flaming jet* which, according to the Wanner pyrometer, has a temperature of from 1800° to 1850° instead of the theoretical temperature or *calorific intensity* of a flame in an athermal chamber of 2100° or more.[1]

Knowing the quantity of fuel which is transformed into gas in the producer per second and its composition, it is not difficult to compute the volume per second of the producer gas and the air required for its combustion. Consequently it is not difficult to compute the velocity with which the preheated gas and air enter the heating chamber.

For example, a furnace which has a nominal capacity of 30 tonnes, and is operated to make four melts per day, consumes 0 kg 347 of coal per second, and this at 0° corresponds to 1 m^3 81 of gas and 2 m^3 52 of air per second.[2] Assuming 1000° as the temperature of the gas and the air, the volumes entering the heating chamber each second will be 8 m^3 44 of gas and 11 m^3 77 of air. The gas and the air combine in the heating chamber of the furnace and give a current of flaming gases at a temperature of 1850°.

To what depth will this jet of flame drop within the heating chamber, if the gas and the air are flowing into the chamber with velocities equal to V_{air} and V_{gas}, making angles of α and β with the horizontal?

The operation of an open-hearth furnace is impossible unless the bottom is well sintered in place. If the bottom is not well made, it will be dug up or float up on the molten metal, and in a furnace in which the bottom cannot be thoroughly sintered the metal will not be sufficiently heated.

The depth H of the gaseous jet must absolutely be greater than the distance from the sill of the gas port to the tapping hole.

(1) The method of computing the *Calorific Intensity Curves* of various combustibles, as developed by the translator from the methods of Mallard and Le Chatelier, and as used by Damour, are given in an appendix to this volume.

(2) Onè kilogram of coal gives approximately 5 m^3 22 of producer gas; this requires about 6 m^3 54 of secondary air, this being 1.25 times the theoretical air supply, plus 0 m^3 125 of air for the oxidization of certain elements of the metallic bath.

This height, which has been given by Professor M. A. Pavlow for thirty-six of the best American and European open-hearth furnaces, varies in his table from 500 to 1875 mm.[1]

Returning to the formula upon page 64,

$$H=\frac{v^2 \sin^2 \delta}{2g}\times\frac{273+t_i}{t_m+t_i}$$

it may be seen that the depth to which the gaseous jet descends increases with the increase in t_i, that is to say, with the temperature in the heating chamber of the furnace. It is clear that when the chamber is cold, that is, when $t_i=0°$, H has its minimum value. When $t_i=t_m$, H becomes infinite. This shows that if a certain velocity is impressed upon some of the particles in the midst of a fluid, and if no account is taken of internal friction, the displacement and the duration of the motion of these particles will be infinite.

If the temperature within the chamber of the open-hearth furnace t_i is not equal to 0° nor to the temperature of the hot gases t_m of the jet of flame, the depth H to which the jet descends will become greater and greater as the difference between t_m-t_i becomes less, that is to say, it will be greatest when the gases filling the chamber are lightest. Thus, for example, a jet of kerosene directed downward into water will penetrate it to a greater depth than it will if directed into mercury. In a furnace chamber filled with hot gases the jet of hot gases will descend to a greater distance than it will into a chamber filled with atmospheric air, cold and heavy.

All those who have assisted at the starting up of an open-hearth furnace know that at first the flame in the cold furnace clings to the roof and drops further and further toward the hearth, as the temperature of the furnace gradually increases. In heating up a new furnace, it is necessary to make the bottom by burning it on in place, that is to say, to form it of its different elements, incorporating a small quantity of refractory clay with a silica sand, or a small quantity of basic slag or of dolomite containing from 3 to 7 per cent of magnesium silicate. This bottom cannot be made except at a temperature of from 1600° to 1700°, and it is impossible

(1) *Rev. de la Société russe de Métallurgie*, 1910, pp. 169–183.

to sinter it in place unless the *flame or hot gases lick the hearth of the furnace.*

As the depth H of the gaseous jet is a function of the temperature of the immobile gases filling the heating chamber of the open-hearth furnace, the dimensions have been calculated in the following manner: all of the data from existing furnaces as given in the table by Professor Pavlow have been used to determine the temperature t within the heating chamber at which a jet of flame or hot gases at 1800° would touch the bottom of the furnace; that is to say, at which H exactly designates the distance from the sill of the gas port to the tapping hole.

For the 30-tonne furnaces Nos. 20, 21, 23, 24, 25, for which the inflow of gas and secondary air at 1000° will be 8 m^3 44 of gas and 11 m^3 77 of air per second, the dimensions and the results of the computation are given in the following table:

Number of Furnace	Area of Gas Port	Velocity of Gas at 1000°	Area of Air Port	Velocity of Air at 1000°	Inclination of Gas Port	Inclination of Air Port	Distance from the Bottom of Gas Port Vertically to Tapping Hole
	m^2	m/sec	m^2	m/sec	Degrees	Degrees	Millimeters
20	0.275	29.6	0.54	21.8	15	38	1150
21	0.358	23.5	0.57	20.6	13	40	980
23	0.440	18.9	0.72	16.3	33	33	980
24	0.260	32.4	0.56	21.0	10	38	1020
25	0.320	26.3	0.64	18.4	15	41	1370

Except in Furnace No. 23, the streams of air and gas have different velocities and different inclinations; the masses of the gas and of the air are also different.

The jets of air and gas at different inclinations combine in the flame with the same velocity and inclination. This can be resolved by the parallelogram of velocities, taking into account the different densities of the air and the gases.

The average velocity and inclination of the flame may be obtained graphically by the parallelogram of the velocities of the gas and of the air multiplied by coefficients proportional to their weight. Now the ratio of the weights [1] in this case is equal to

$$\frac{1.81 \times 1.069}{2.525 \times 1.29} = \frac{1.9349}{3.2575} = \frac{37.26 \text{ per cent}}{62.74 \text{ per cent}}$$

Therefore for Furnace No. 20, for example, the result is

For the air: $21.8 \times 0.6274 = 13.6773$
For the gas: $29.6 \times 0.3726 = 11.0290$

The diagonal of the parallelogram for these two values is equal to 24 m 50 per second at an inclination of 27°. The results, which will be obtained for the other furnaces by similar computations, are given in the following table:

Number of Furnace	Average Angle α.	Average Velocity v.
	Degrees	Meters per Second
20	27	24.00
21	29	21.20
23	33	17.30
24	25	24.50
25	29	21.00

Inserting the above values in the formula

$$H = \frac{v^2 \sin^2 \alpha}{2g} \times \frac{273 + t_t}{t_m - t_t}$$

with the values of v and α given in the table and assuming that $t_m = 1850°$ and that H equals the difference in level between the sill of the gas port and the tapping hole, the temperature t_t at which the stream of flame drops to the level of the hearth at the tapping hole may be computed as follows, in order to determine the temperature to which the gases filling the furnace must be raised before it becomes possible to start making the bottom:

[1] It should be remembered that at 0° the volume per second of the gas is 1.81, and its specific weight is 1.069. The volume of the air is 2.525 and its specific weight 1.29.

Furnace No. 20 .... $t_i > 74°$
21 .... $t_i > 58°$
23 .... $t_i > 22°$
24 .... $t_i > 70°$
25 .... $t_i > 158°$

The following example shows the influence of the density of the medium which the jet of gas penetrates. For Furnace No. 20 the depth $H = 1150$ mm at 74°.

Determining the depth of the jet at a temperature $t_i = 0°$, there will be obtained

$$H = \frac{v^2}{2g} \sin^2 \alpha \times \frac{273+0}{1850-0} = 864 \text{ mm.}$$

Therefore the flame or stream of hot gases does not touch the hearth, and the distance by which it falls short of so doing is $1150 - 864 = 286$ mm.

Therefore, in these five furnaces, each of 30 tonnes capacity, selected at random, such a velocity and an inclination have been given to the gas and the air in the heads that the flame settles to the hearth when the other portions of the furnace are at a comparatively low temperature.

It is evident that perfect repairing of the hearth is assured in this manner, but that it is not an advantageous arrangement in the regular working of the furnace. A similar investigation deals with the circulation of the hot gases in the long furnaces used in American plants, where the flues in the heads have a very slight slope and where the velocities are very low for the air and very high for the gas.

This data has been taken from Professor Pavlow's table for two 60-tonne furnaces, Nos. 35 and 36.

Making the same assumptions as in the case of the 30-tonne furnaces, the volume of air at 1000° will be 23 m^3 54 per second and the gas at the same temperature will be 16 m^3 88 per second.

Making the calculations for the temperature t_i at which it becomes possible to make bottom in these furnaces, the results are as follows:

	Furnace No. 35	Furnace No. 36
Area of gas ports	0 m² 7300	0 m² 3935
Velocity of gas at port	23 m 11 per sec	42 m 90 per sec
Area of air ports	2 m² 70	3 m² 75
Velocity of air at port	8 m 72 per sec	6 m 28 per sec
Slope of gas port	6°	12°
Air port	17°	26°
H = depth of jet = difference in level between sill of gas port and the tapping hole	0 m 92	1 m 00
Average velocity of the mixture by parallelogram	11 m 80 per sec	20 m 50 per sec
Average slope of mixture by parallelogram	9° 30′	14° 30′
t_1 at which the mixture commences to lick hearth	1531°	646°

These computations would show that the making of the bottom in Furnace No. 35 would be difficult, and it is hard to believe that the design published is correct.(1)

As regards Furnace No. 36, it will be comparatively easy to make the bottom, and the flame will not be deflected from the surface of the bath. In the operation of this furnace (No. 36) its ports will be burned and worn away, their area will be increased and the velocity of the gas will be accordingly reduced. How much may the port be worn away without interfering to an appreciable extent with the repairing of the bottom?

The formula (F), on page 64, provides the solution of this problem.

The bottom may be made in a satisfactory manner when $t_1 = 1400°$; the corresponding value of v may be deduced:

$$1.00 = \frac{v^2 \sin^2 14.5°}{2 \times 9.81} \times \frac{273+1400}{1850-1400}$$

from which $v = 9$ m 20 per second.

(1) A great deal of empiricism exists in the arrangement of the heads of these furnaces, and in many cases the furnaces as they go into operation differ widely from the drawings. In some plants no two furnaces have identical heads.

The velocity for the original section was 20 m 50 per second; the ratio of these velocities is

$$\frac{20.50}{9.20}=2.2,$$

which shows that the furnace will continue to function with a gas port area 2.2 times its original area.

This analysis is necesssary to enable the designer to plan the lines of the furnace in such a manner that a long campaign will be assured.[1]

XI. METHODS OF COMPUTING FOR FURNACES OF VARIOUS TYPES

In this chapter it is proposed to supply a computation scheme which will be of service to engineers and technicians whose duties make it necessary for them to follow closely the working of reverberatory furnaces. All the data which are available at this time are based upon a small number of furnaces. For this reason it is impossible to confirm the absolute exactness of the coefficients which are presented.

The verification of these numerical constants is necessary in order to fix their limitations. Research should be undertaken by others, and the author will be glad to give the fullest recognition to those co-workers who desire to contribute to the improvement of furnace work and who will aid him in attaining that end.

The following order of procedure is suggested for use in making the computations and sketches for a furnace.

In designing a furnace, the first thing to do is to roughly outline the heating chamber. Within this outline should be indicated the direction of the gaseous currents and, consequently, the system upon which the furnace works.

For example, in the case of a brick kiln, the height will be fixed by the consistency of the unburned bricks, which will limit the height to which they may be set, and the length and width (or diameter) will be fixed by the capacity of the kiln, or the number of bricks to be burned at one setting.

In the case of a furnace for the gradual or continuous heating of ingots, the dimensions of the hearth of the furnace will be

[1] An analysis of this character might show that expensive cooling device were neither desirable or necessary.

determined from data including its production, that is, the number of ingots to be heated per hour, the time required for their reheating and their arrangement upon the hearth of the furnace.

The distance from the hearth to the roof of the furnace will be established by keeping the following facts in mind: sufficient space must be provided to obtain complete combustion of the hot gases, and constant contact of the hot gases with walls or material at a high temperature is an essential condition. A very serious fault of many furnace designs is the premature cooling of the flame or burning gases by directing them in such a manner that they impinge upon cold ingots or other cold material. On the other hand, it would serve no useful purpose to make the chamber too large, but it is desirable to make the roof of sufficient height to give the hot gases of the flame a sufficient time to complete their reaction before they come in contact with bodies sufficiently cold to impede or prevent the completion of the reaction of combustion.

When the lines and working method for the heating chamber have been determined, its volume may be computed.

The same method of procedure is used for tempering furnaces, annealing furnaces, iron-melting furnaces, puddling furnaces, etc.; the very first thing that is necessary is to determine the dimensions of the heating chamber and its *volume*. When the foregoing have been fixed, the composition of the furnace gases, or, more exactly, the volume of air required with reference to that theoretically required to burn the fuel, is assumed.

Calculations show that a good coal from central Russia, burned with the theoretical volume of air required, gives a theoretical calorific intensity of 2082°. With 70 per cent excess air supply this coal will give a calorific intensity of 1400°; and with double the theoretical air supply (100 per cent excess air) the calorific intensity will be 1250°.

Now, it is evident that these temperatures can only be obtained with instantaneous combustion in an athermal chamber, the total amount of heat released being absorbed in raising the temperature of the gases of combustion. In reality the best means of lowering the temperature of the *jet of burning gases* is to operate the furnace so that combustion takes place with only the theoretical air supply. Such combustion requires a *certain length of time;* the combustion of the last traces of combustible gas requires a considerable amount of time in its combination with the small amount of oxygen present, and it will not be effected in the presence of objects which tend to cool the gases.

This flame has a *soft and languishing* character, and an average temperature in the neighborhood of 1100°, and is employed exclusively in chamber brick kilns, tempering furnaces, annealing furnaces and furnaces used in the manufacture of plates and sheets. It is only in the very large chamber brick kilns, where the radiation losses are insignificant, that it is possible to obtain a temperature in the neighborhood of 1400° with this kind of combustion.

If, on the other hand, it is necessary that the reaction of combustion should be completed rapidly, the actual temperature realized will be a little lower than the computed temperature. In this case it will be necessary to have an excess of air and an intimate mixture of the air and gases forming the flame. The greater the excess of air, the greater will be the proportion of the high temperature core in the flame.

A flame with a great excess of air is *sharp and penetrating* and is frequently more detrimental than useful in the uniform heating of material. It can be used in the puddling furnace or in the melting of iron, where the temperature in the furnace, as required by the process, very closely approaches the temperature of the hot flaming gases, or where the operation to be performed somewhat resembles that required of the gas welding or cutting torch.

The slowness and diffusion of the energy in the reaction of theoretical combustion is a phenomenon which is utilized in a number of ways, when it is desired to have a temperature which is not very high, but which is uniform (for tempering, annealing, the reheating of plates, etc.). In these cases, the reaction of combustion is slowed down by using cold air and gas which are not well mixed, in a very large combustion chamber out of contact with cold material. An ideal combustion chamber for a furnace of this character would be a high free space under a roof or dome, from which the currents of hot gases produced by combustion would fall, and in which the hot flaming gases would always be in contact with incandescent brickwork.

In practical work the composition of the furnace gases may be assumed, according to the character of the product. Five cases are presented here.

1st Case.—Small furnaces, within which the hot gases remain only a fraction of a second and are directed *immediately* upon a cold body. Combustion occurs with cold air and solid fuel burned in a *thin* layer upon a simple bar grate. As an example of such a construction, the firebox of a tubular boiler may be taken. Combustion is effected rapidly under very unfavorable conditions —the air supply is usually double that theoretically required.

2d Case.—A simple firebox, with a grate, but with a simple brick arch over the fuel bed. Combustion occurs in a *thick* bed of coal. In the case of the furnace working with chimney or natural draft, it is necessary to have sufficient air pressure to overcome the resistance of the fuel bed to the passage of the air and gases.[(1)] Combustion is rapid, as in the preceding case, but is effected under more favorable conditions. A thick bed of coal is an excellent medium of combustion; the firebox covered with an arch makes a satisfactory combustion chamber. Practice has shown that, under these conditions, when the fire is well operated, the combustion of coal may be effected with an air supply of about 1.50 times that theoretically required.

A furnace fired in this manner in the Lougansk works, using *a blower,* burns 200 kg of coal per hour per square meter of grate surface. When fireboxes of this type are used with natural or chimney draft only, such a high rate of firing cannot be used; the fuel consumption for such cases should not exceed 70 kg per hour. For the ordinary types of furnaces with a firebox of this kind, such as puddling furnaces, reverberatory furnaces for the melting of iron, copper, etc., and for reheating furnaces, the computation should be based on an air supply 1.50 times that theoretically required, and with very good coal 1.70 times.

3d Case.—Producer-gas-fired furnaces. These should be figured as having a secondary air supply 1.50 times that theoretically required.

4th Case.—Regenerative or Siemens furnaces. These should be computed for a secondary air supply of 1.25 to 1.50 times the theoretical requirements.

5th Case.—Annealing furnaces, tempering furnaces, chamber brick kilns, etc., should be computed for the theoretical air supply.

[(1)] This was mentioned in an earlier chapter. This pressure varies with the height between the grate and the hearth of the furnace. The simplest method of providing this pressure is by lowering the grate until it is sufficiently far below the hearth to provide the pressure required to overcome the resistance of the fuel bed.

According to the composition of the gas and the amount of excess air, there will be obtained different quantities of the products of combustion, per kilogram of coal.

By computation it can be shown that one of the best grades of coal mined in the Donietz basin (central Russia)[1] requires for the combustion of 1 kg of coal an air supply of 8 m^3 71, and the volume of the products of combustion is 9 m^3 04, theoretical or neutral combustion.

According to the above, the volume of gases given off by the combustion of 1 kg of this coal will be

Theoretical air supply	9 m^3	04
1.25 times theoretical air supply	11	22
1.50	13	39
1.75	15	59
2.00	17	75

For producer-gas-fired furnaces, 1 kg of coal gives about 5 m^3 22 of producer gas. With a blower combining the air and the steam, this requires 3 m^3 16 of air for the primary supply and 5 m^3 54 for the secondary air supply. The volume of the products of combustion with the theoretical air supply are 9 m^3 71; and with an air supply of 1.50 times the theoretical volume the volume of the products of combustion will be 12 m^3 54.

Knowing the quantity of combustible which will be used in a unit of time and the resulting volume of the products of combustion, there is another factor which must be determined: *the time that the products of combustion or hot gases remain in the heating chamber of the furnace.*

(1) The average composition of this coal is:

	Per cent
C	84.75
H_2	4.80
O_2	4.83
N_2	1.44
S	1.48
Moisture	0.90
Ash	1.80

For larger quantities of moisture and ash, it would be necessary to correct the air supply and the products of combustion volumes accordingly.

According to calculations based upon various furnaces in service, it was found that the time during which the waste gases remained in the furnace varies from a fraction of a second to 3, 5, 7, and up to 10 seconds. The determination of the length of time which the gases must remain in the furnace chamber is based upon the following considerations:

Computations may be made which will give the theoretical calorific intensity obtained by the combustion of a coal with an assumed excess of air—for example, the combustion of the best grade of Donietz basin coal, with an excess of 73.3 per cent of air over that theoretically required will give a calorific intensity of 1400°. If, for the designing of a continuous ingot heating furnace, it is assumed that there will be a temperature of 700° for the gases at the exit port, the total drop of temperature in the heating chamber will be 700°. If the drop in temperature of these gases per second is known, it is comparatively simple to determine the time the gases should remain in the heating chamber of the furnace.

According to those computations which have already been made—computations which at the best are only approximate—the following values have been found for the drop in temperature per second:

Open-hearth furnaces...........	200° per second
Continuous reheating furnaces...	150°–200°
Annealing furnaces.............	100°–150°
Chamber brick kilns...........	80°

It should not be considered that these values are perfectly established. Exact data covering the drop in temperature of the gases per second can only be obtained by a series of observations for each of the varieties of furnaces, using for this purpose furnaces *which are correctly designed and constructed.* This last point presents the principal difficulty which exists at present.

It is evident that the drop in temperature per second of the gases in a furnace heating chamber is a function of:

(*a*) The losses by radiation, which are a function of the exposed surface of the furnace, the conductivity of the walls and the temperature of the hot gases on the inside of the chamber;

that is to say, the losses by radiation are peculiar to each type and size of furnace.[1]

(*b*) The ability of the body being heated to absorb heat from the hot gases and by radiation. This may be generalized in the following statement:

Assuming that one reheating furnace is charged with cold ingots and another with hot ingots, it is very evident that these two furnaces will work in different fashions. The drop in temperature of the hot gases in the first of these furnaces will be much greater than it will be in the second furnace. Accordingly, in the first furnace the time during which the hot gases remain in the heating chamber should be diminished and the volume of the gases per second should be increased. It is therefore necessary that the dimensions of the heating chamber and the size of the ports should be designed and constructed to conform to these conditions.

Similar differences will exist in furnaces designed for the production of different outputs. Consider, for example, a rolling mill and a reheating furnace to serve the mill. It is evident that the operation of the furnace must be regulated to suit the output of the rolling mill, that is, when the mill is working fast and without interruptions of output, it will be necessary to heat fast and, accordingly, to decrease the time during which the hot gases remain in the furnace. On the contrary, if the work of the mill is subject to interruptions, the time during which the hot gases remain in the heating chamber will be increased and the output of the furnace will accordingly be decreased.

The foregoing shows the importance of some of the factors which have to be established, and it is desirable that the value of these factors should be based upon furnaces actually in service, by accurate determinations of the drop in temperature of the gases in the heating chamber per second. These values are of primary importance in the design computations for furnaces.

Therefore, according to the operating conditions of the furnace whose design is to be established, the following order of procedure is observed:

1. The necessary volume of the heating chamber is computed.
2. The volume of gases obtained per kilogram of combustible

[1] *Note by translator.*—The exposure of the furnace to air currents and the convection currents arising from the hot walls must be taken into consideration.

burned is assumed, according to the operating condition of the furnaces to be designed.

3. The theoretical calorific intensity of the combustible, with the assumed excess air supply over that theoretically required, is computed.(1)

4. The following temperatures have been established for the gases leaving the heating chambers of furnaces:

Open-hearth furnaces	1600°
Puddling and reverberatory furnaces	1250°
Tempering or heat treating furnaces	850°
Annealing furnaces, etc.	1000°

5. The difference between the computed temperature, or the theoretical calorific intensity, and the temperature of the gases leaving the heating chamber is divided by the number of degrees of temperature drop of the gases per second. This determines the length of time during which the hot gases remain in the heating chamber.

6. Dividing the volume of the heating chamber by the time during which the hot gases remain in it gives Q_t, the volume of the gas at the temperature t, which passes through the heating chamber each second. This value divided by $1+\alpha t$, gives the volume of the gases at 0°, and, according to the volume of gases required at 0°, the quantity of fuel required per second, per hour or per twenty-four hours will be fixed.

7. Knowing Q_t, according to formulas previously given, the *principal dimensions* of the furnace may be determined. This is done by fixing the velocity of flow of the gases in the different parts of the furnace, and then computing, according to the conditions, the hydrostatic pressure of the gases and the vertical dis-

(1) *Note by translator.*—This may be done by the use of the methods of Mallard and Le Chatelier. The theoretical calorific intensity, however, assumes that combustion occurs instantaneously in an athermal chamber, the total amount of heat released being absorbed in increasing the temperature of the products of combustion. In practice the velocity of combustion is not instantaneous but requires an appreciable time interval; the chamber in which combustion occurs is more or less dithermal and for this reason the practical or actually obtained calorific intensity is less than the theoretical. A further difference is due to the fact that the fuel usually contains more or less moisture, gases being frequently saturated to the dew-point temperature, and in addition the air supply contains some moisture. The design and construction of the furnace affect the result.

tance which must exist between the level of the grate and the hearth of the furnace, if this last operates with natural or chimney draft, or the draft pressure which is necessary if the air is forced in by a blower. After this, the required height of the chimney may be computed.

It is not desired, in this work, to present the complete computations for the design of all of the many different types of furnaces. The following, however, gives an example of the application of these methods of computation, as they have been used by the author.

Problem.—The design required is that of a small furnace for the reheating of small pieces of iron (billets or fagots) to be installed in connection with a waste heat boiler. The height between the hearth and the arch of the furnace cannot be less than 0 m 41; the width of the heating chamber is to be 2 m. The temperature of the gases at the outlet port is to be 1200°.

Computing the volume of the gases,

$$h_{1200} = 0 \text{ m } 41 = 3.50\sqrt[3]{\frac{Q^2{}_{1200}}{B^2 \cdot t}} = 3.50\sqrt[3]{\frac{Q^2{}_{1200}}{2^2 \times 1200}},$$

from which $Q_{1200} = 2 \text{ m}^3\ 77$ per second,

$$Q_0 = \frac{2.77}{1+\frac{1200}{273}} = \frac{2.77}{5.39} = 0 \text{ m}^3\ 514 \text{ per second.}$$

One kilogram of coal burned with an air supply 60 per cent in excess of the theoretical requirements produces

$$9.04 + (8.72 \times 0.60) = 14 \text{ m}^3\ 27 \text{ of gases.}$$

From this the quantity of coal required per second may be deduced

$$x : 1 = 0.514 : 14.27, \qquad x = 0 \text{ kg } 036,$$

which will be 129 kg per hour or 3 tonnes 110 for twenty-four hours.

Fixing the dimensions of the furnace:

(*a*) *Height at Bridge Wall.*—This height will be determined as an inverted weir with the hot gas temperature $t = 1400°$.

$$Q_{1400} = 0.514\left(1+\frac{1400}{273}\right) = 0.514 \times 6.12 = 3 \text{ m}^3\ 15 \text{ per second,}$$

$$h_{1400} = 3.50\sqrt[3]{\frac{Q^2{}_{1400}}{2^2 \times 1400}} = 0 \text{ m } 423.$$

(*b*) *Height of Bridge Wall.*—Considered with reference to the velocity of the gases over the bridge wall, it is theoretically possible to decrease this velocity over the hearth by increasing the thickness of the stream of flowing gases by one-half their thickness over the bridge wall.[1] As it is necessary that the velocity should not be too great over the hearth, the roof will be given a downward slope and the height of the bridge wall will be fixed at not more than one-half the height of the opening over the bridge wall. In the construction of the bridge wall this proportion will be reduced to one-third (140 mm), and the roof will be given a downward slope of the same amount toward the exit port for the gases; a general longitudinal outline of the furnace will, therefore, appear as in Fig. 44.

The working chamber will be supplied with two working doors,

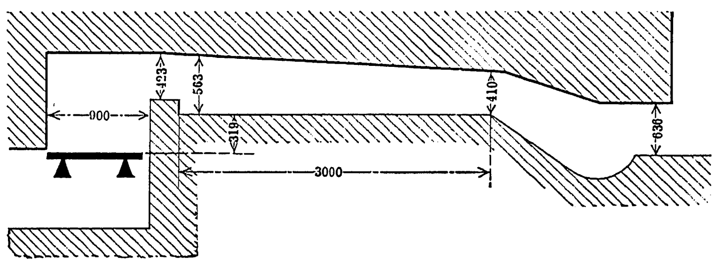


FIG. 44.

each having a clear opening 400 mm in height. The hearth will be given a slight grade or slope toward the gas-exit port, in order to permit the cinder deposited upon the bottom to drain off into the cinder pocket in the flue. The waste gas flue will be dropped below the level of the hearth, giving any cold air or gases which may enter the working chamber a chance to drain out of the chamber. With the usual construction, it is impossible to prevent small amounts of cold air from entering the furnace below the doors.

(*c*) *Dimensions of Grate.*—These will be based upon the assumption that 75 kg of coal can be burned per square meter per hour (p. 75) (chimney draft):

$$\frac{129}{75} = 1 \text{ m}^2\ 72 \text{ or approximately } 2 \text{ m } 0 \times 0 \text{ m } 90.$$

(1) According to Yesmann $h = \frac{2}{3}H$.

(*d*) *Vertical Distance of Grate Bars below the Hearth of the Furnace.*—This distance must be sufficient to impress the desired velocities upon the gases in the different portions of the furnace and firebox.

Starting with the passage of the air through the ash pit, the volume of air required is

$$8.71 \times 1.60 \times 0.036 = 0 \text{ m}^3\ 501 \text{ per second.}$$

In order to secure a velocity of 0 m 50 per second in the ash pit this pit must be given an area of 1 m² 00 = 2 m 0 × 0 m 50.

To impress a velocity $V_1 = 0$ m 50 per second upon the air, a velocity head $h_1 = 0$ m 0127 is required, neglecting friction,[1] from which

$$\delta_1 = 1.29 \times 0.0127 = 0 \text{ mm } 016 \text{ of water.}$$

The total free area between the grate bars is equal to

$$\frac{1 \text{ m}^2\ 80}{3} = 0 \text{ m}^2\ 60.$$

The velocity V_2 of the air passing through the grate will be

$$0.501 : 0.60 = 0 \text{ m } 835 \text{ per second,}$$

from which $h_2 = 0.036$ and

$$\delta_2 = 1.29 \times 0.36 = 0 \text{ mm } 046 \text{ of water column.}$$

Resistance of the Fuel Bed.—It is assumed that the depth of the bed of burning coal upon the grate has a thickness of 150 mm and that it is formed of three layers or tiers of coal in small pieces, each layer having a thickness of 50 mm; therefore in passing through the fuel bed the air will lose its velocity three times.

The velocity head required to impress upon the air the velocity with which it passes through each layer or tier of the fuel will be the resistance offered by that tier to the passage of the air, and the total resistance of the bed of fuel upon the grate will be the sum of the resistances of the number of tiers into which it is assumed that the bed of fuel is divided.

The volume of air which will be contained in the spaces or interstices between the pieces of coal forming the fuel bed may be calculated as follows: the specific weight of coal is about 1.20,

[1] Velocity heads required, refer to Appendix III.

that is, a cubic meter of solid coal will weigh about 1200 kg; a cubic meter of coal in small pieces will weigh about 700 kg. It follows that the volume of the spaces filled with air will be equal to $\frac{1200-700}{1200}=0\ \mathrm{m}^3\ 42$ and that the total cross-sectional area of these interstices will be equal to $0\ \mathrm{m}^2\ 42$ per square meter. As the total area of the grate is $1\ \mathrm{m}^2\ 80$, the area of the air spaces will be

$$0.4\times2\times1.80=0\ \mathrm{m}^2\ 750$$

and the velocity of the air passing through them will be

$$V_3=\frac{0.501}{0.750}=0\ \mathrm{m}\ 68 \text{ per second,}$$

from which $h_3=0\ \mathrm{m}\ 0235$ (1) and $\delta_3=1.29\times0.0235=0\ \mathrm{mm}\ 03$ of water; and as the velocity is lost three times, once for each layer of the fuel bed:

$$\delta_3=0.03\times3=0\ \mathrm{mm}\ 09 \text{ of water column.}$$ (2)

The velocity of the gaseous products of combustion above the fuel bed may be determined in the following manner:

$$Q_{1200}=2\ \mathrm{m}^3\ 77,$$

$$V_4=\frac{2.77}{2\times0.9}=1\ \mathrm{m}\ 54 \text{ per second,}$$

from which

$$h_4=0\ \mathrm{m}\ 115 \text{ and } \delta_4=\frac{0.115\times1.326\ ^{(3)}}{5.39}=0\ \mathrm{mm}\ 0275 \text{ of water.}$$

The velocity of the hot gases over the bridge wall

$$V_5=\frac{3.45}{0.423\times2}=3\ \mathrm{m}\ 75 \text{ per second,}$$

(1) Refer to Appendix III for heads required for various velocities.

(2) *Note by English translator.*—The foregoing is not exactly correct, but it illustrates the method to be followed. The lower layer of the fuel bed will be composed mostly of ashes and a small amount of burning fuel. The second layer will be burning fuel and the third layer will be composed of burning coal and partially burning or fresh fuel. Another factor that has not been considered above is the increase in temperature of the air and its change into the gases of combustion. A considerable volumetric expansion will result in increased resistance to the passage of the air.

(3) This is the weight of 1 cubic meter of the products of combustion.

from which

$$h_5 = 0 \text{ m } 716 \text{ [1] and } \delta_5 = \frac{0.716 \times 1.326}{6.122} = 0 \text{ mm } 155.$$

The total of these pressures and losses will be

1. In the ash pit..................	$\delta_1 = 0$ mm 016
2. In the grate openings............	$\delta_2 = 0$ 046
3. In the fuel bed..................	$\delta_3 = 0$ 090
4. In the upper part of the firebox....	$\delta_4 = 0$ 027
5. Over the bridge wall.............	$\delta_5 = 0$ 155
	$\delta = 0$ mm 334

Therefore, the hydrostatic pressure required at the level of the grate will be, as a minimum, + 0 mm 334 of water column.

At the temperature $t = 1200°$, the weight of 1 cu m of gas in the furnace will be

$$\Delta_{1200} = \frac{1.326}{5.39} = 0 \text{ kg } 246.$$

Therefore a column of this gas 1 m in height will give a hydrostatic pressure of

$$1.29 - 0.246 = 1 \text{ mm } 044 \text{ of water.}$$

Then, using x to express the height in millimeters of the column of gases corresponding to the pressure of a water column of 0 mm 334 in height, the following proportion can be written:

$$\frac{x}{1000} = \frac{0.334}{1.044},$$

from which : $x = 319$ mm, which is the minimum vertical distance of the grate bars below the hearth of the furnace.

(*e*) *Waste Gas or Smoke Flue.*—As the hot gases from the furnace are to be passed under a boiler, there is no necessity for increasing their velocity of flow. Increasing their velocity would necessitate a decrease in the gas pressure below the boiler by means of the draft supplied by the chimney. But such a decrease in the pressure is always accompanied by the sucking in of the cold outside air, which reduces the temperature of the gases and decreases the production of steam; it is therefore desirable to

[1] Refer to Appendix III.

give the waste gas flue sufficient area to reduce its frictional resistance to the passage of the gases to a minimum.

Assuming for the waste gas flue, considered as an inverted weir, a width of 1 m, its height will be determined by Yesmann's formula:

$$h_{1200}=342\sqrt[3]{\frac{2.77^2}{1^2\times1200}}=0 \text{ mm } 636.$$

In order to insure the contact of the hot gases with the hearth of the furnace, the waste gas flue is inclined downward from the hearth level. This provides, at the same time, a pocket into which the cinders from the hearth may drain.

It is now necessary to check the time period of the hot gases in the heating chamber of the furnace and to determine the drop in the temperature of these gases per second.

The average temperature of the hot gases will be

$$\frac{1400+1200}{2}=1300°,$$

from which $$Q_{1300}=2 \text{ m}^3\ 96.$$

The volume of the heating chamber of the furnace is equal, according to the dimensions given, to 2 m³ 80. The time during which the hot gases remain in this chamber is therefore in the neighborhood of one second, and the drop in temperature of the gases is in the neighborhood of 200° per second, which may be considered normal for this type of furnace.

Part II

PRINCIPLES FOR THE RATIONAL CONSTRUCTION OF FURNACES

The problems of furnace construction will be solved when it is possible to regulate the temperature within the enclosure of their heating chambers according to the requirements of the material to be heated. The gas passes from the firebox into the heating chamber without having completed combustion. The first problem to be solved, therefore, is to afford space in the heating chamber within which combustion may be completed. With a short concentrated core of burning gases the highest temperatures are obtained. At other times, according to the material and the method by which it must be heated, it is necessary to prevent the formation of a jet of burning gases and to provide a general combustion of the gases throughout the heating chamber (*a long, soft flame*).

The second problem is in the heating, by means of the hot gases obtained, of those objects which have been placed in the heating chamber of the furnace for this purpose.

For the time being, the first of these problems will be neglected, and this portion of the present work will be devoted exclusively to the solution of the second problem, which may be more definitely stated as follows: In what manner may the hot gases be circulated so that they will, in the most perfect manner, surround the objects being heated and be carried out of the heating chamber, in order that their place may be taken by hotter gases? In what manner may the heating chamber be adapted to obtain such a circulation of the hot gases?

The solution of this second problem is very simple, but in spite of its simplicity it is very poorly understood by practicing furnace designers. Upon the following pages are collected a number of designs of furnaces which have been operated or are still in operation, and which infringe the most simple natural laws, that is to say, a veritable collection of the monstrosities of furnace design.

The Subdivision of a Current of Hot Gas.—Assume that the current of hot flowing gas Q, which is giving off heat or cooling, is to be divided between two vertical ascending channels, q_1 and q_2, this division to be effected in the manner shown in Fig. 45. The veins of gas q_1 and q_2 are at temperatures t_1 and t_2 and $q_1 = q_2$ and $t_1 = t_2$. Assume that during the operation of this system the amount of heat lost from the branch q_1 is a little greater than the amount of heat lost from the branch q_2 and that, accordingly, the temperature t_1 becomes slightly less than t_2. When $t_2 > t_1$, the

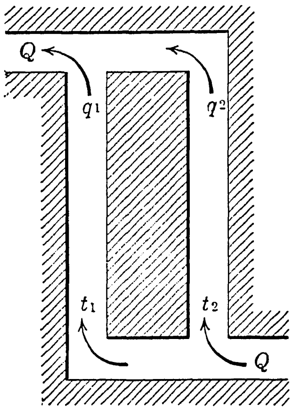


FIG. 45.

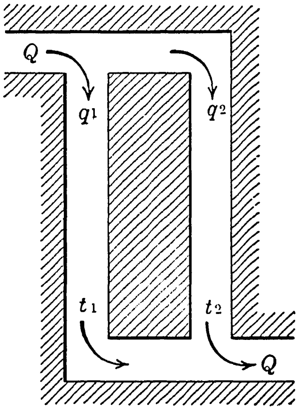


FIG. 46.

weight of the column of gas q_1 becomes greater than that of the column of gas q_2, and accordingly the velocity of the current q_1 will become less than that of the current q_2. The decrease of the velocity of the current of gas q_1 results in a further lowering of its temperature t_1; in other words, the velocity of the current of gas q_1 will continue to diminish, and its temperature t_1 will continue to decrease. During this time the velocity of the current of gas q_2 will, on the contrary, be increased, and its temperature will become higher. The current of gas q_1 will finally cease to flow and the entire volume of the current of gas Q will then pass through the branch q_2. But there will still be a loss of heat from the channel q_1 and after the velocity of this current has decreased to zero it will reverse and travel in the opposite direction, as is indicated in Fig. 47. From the preceding, it may be readily seen that *a current of hot gas which is giving off heat or cooling cannot be subdivided into equal ascending currents.*

When, however, the attempt is made to subdivide the current of gas Q into two equal *descending currents* (Fig. 46), it will be completely successful.

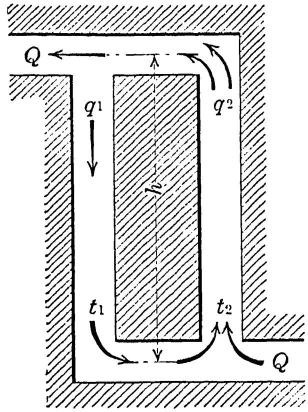


FIG. 47.

Assuming, for example, that the temperature of one of these currents, q_1, should become less than the temperature of the current q_2. In this case the weight of the column of gas q_1 will be increased, and the velocity of its descending motion will be increased. The current of gas q_1 will become stronger than q_2, its temperature will gradually increase and it will finally become equal to q_2. It can be concluded from this that the problem of subdividing a current of gas which is cooling or giving off heat into equal descending channels may be solved in a very simple manner, owing to the fact that there is always a tendency for the temperatures of these descending streams to remain uniform.

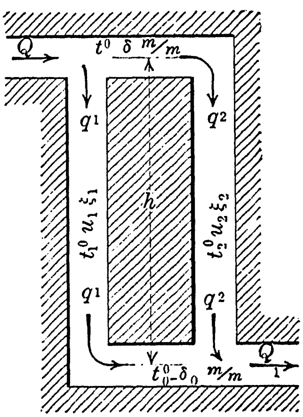


FIG. 48*a*.

Therefore, *if it is desired to subdivide a current of hot gas which is cooling or giving off heat, into equal streams, it is necessary to give these streams a downward or descending direction of flow; or, in other words, a current which is cooling may be subdivided into uniform descending streams.*(1)

(1) Accordingly, it is possible to approximate the frictional resistance in the two channels between which the current is divided, when one branch has a higher resistance or a higher heat loss than the other, because the stream of gas divides itself accordingly.

In Fig. 48*a*, such a case is shown. The stream of gas Q is divided into two descending streams q_1 and q_2, the average temperatures of which are different, being denoted by t_1 and t_2. The average

It is frequently necessary to subdivide a current of cold air or gas which is being heated, as in the hot-blast stove or in furnace regenerators. This problem may be solved as follows: Figs. 48 and 49 show a current of a cold gas circulating through a channel, having walls heated to incandescence. Assume that the stream of cold gas being heated has been equally divided between two

velocities of these two streams are denoted by u_1 and u_2 and the friction in the two branches in millimeters of water is ξ_1 and ξ_2.

The condition necessary for the maintenance of equilibrium, in this case, is that the increase of the hydrostatic pressure in the two branches q_1 and q_2 shall be equal. If there were no loss of hydrostatic pressure in impressing the velocities u_1 and u_2 upon the two branches and in overcoming the frictional resistance ξ_1 and ξ_2 of the two channels to the passage of the gas, the hydrostatic pressure in millimeters of water in the channel q_1 of a height h would be, taking 1.29 kg as the weight of a cubic meter of the furnace gas at 0°

$$1.29h\left[1-\frac{1}{1+\alpha t_1}\right]=1.29\cdot h\cdot\frac{\alpha t_1}{1+\alpha t_1}.$$

For the branch q_2, the hydrostatic pressure would be

$$1.29\cdot h\cdot\frac{\alpha t_2}{1+\alpha t_2}.$$

A part of these increases in the hydrostatic pressure will be expended in overcoming the frictional resistances ξ_1 and ξ_2, and in impressing the velocities u_1 and u_2 upon the columns of gas. These last losses, in millimeters of water, may be expressed in the following manner:

$$\frac{u_1^2}{2g}\times\frac{1.29}{1+\alpha t_1} \quad\text{and}\quad \frac{u_2^2}{2g}\times\frac{1.29}{1+\alpha t_2},$$

and the condition for the equality of the increases in hydrostatic pressure in the two branches is given by the following equation:

$$1.29h\cdot\frac{\alpha t_1}{1+\alpha t_1}-\xi_1-\frac{u_1^2}{2g}\cdot\frac{1.29}{1+\alpha t_1}=1.29h\cdot\frac{\alpha t_2}{1+\alpha t_2}-\xi_2-\frac{u_2^2}{2g}\cdot\frac{1.29}{1+\alpha t_2}.$$

In this equation there are six variables; five of these must be known in order to fix the value of the sixth.

For example, the checker openings around the outside of the checker-work of a Cowper hot-blast stove lose a great deal more heat by radiation and by the cooling effect of the outside of the stove than the central passes. By reason of this they have a much greater cooling effect upon the current of hot gases, and therefore the current of gases flowing downward through these openings is reinforced, since if $t_2<t_1$, $u_2>u_1$. By measuring t_2 and t_1, it is not difficult to find $\frac{u_1}{u_2}$.

descending channels q_1 and q_2 having equal temperatures t_1 and t_2, that one of these streams takes up the heat a little faster than the other and that, for example, t_1 becomes slightly less than t_2. The column of gas q_1 becomes, therefore, slightly heavier than the column q_2, the current q_1 commences to flow with greater energy, and its velocity increases; t_1 commences to become sensibly less than t_2, the current q_1 has a greater cooling effect than the current q_2 which continues, on the contrary, to take up more heat; and, in the end, the entire stream of gases passes through the branch q_1, while in the branch q_2 there will be established at the same time a reverse current which circulates as indicated by the dotted arrows (Fig. 48).

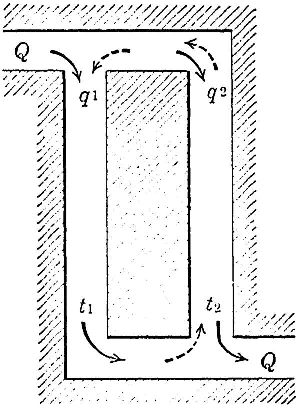


FIG. 48.

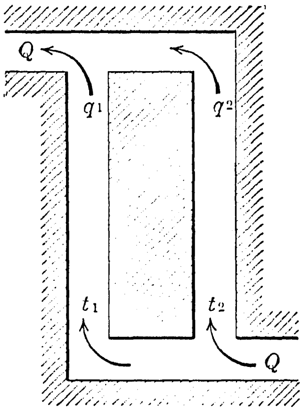


FIG. 49.

Therefore, *a current of cold gases which are being heated cannot be subdivided equally between descending channels.*

When the subdivision of a stream of cold gases is made through ascending channels, the results will be as desired. Assuming that the currents flowing as indicated in Fig. 49 were not equal, $q_1 > q_2$ and, consequently, $t_2 > t_1$. If $t_2 > t_1$, the weight of the column q_2 will be less than the weight of the column q_1, the current q_2 will become stronger; this will cause the temperature t_2 to become lower; at the same time t_1 will increase in this manner and the two temperatures will be equalized; the two currents q_1 and q_2 will therefore be maintained equal.

It therefore follows that *when a current of cold gas is to be heated it should be subdivided into a number of ascending streams.*

These rules regarding the subdivision of gaseous currents, while extremely simple and elementary in character, have received very little attention from metallurgists. The experienced chimney builder, who installs hot-air-heating apparatus in residences, is well acquainted with these laws, which are absolutely ignored by a number of very eminent engineers.

A number of examples of incorrect furnace construction are shown below. These furnaces are designed to work in complete or partial opposition to the foregoing rules; they accordingly work very poorly, and many of them have had to be completely abandoned.

In industrial practice, so many of these defectively designed furnaces are encountered that it is absolutely impossible to enumerate all of them. It is therefore thought that the best method of illustrating these defects will be a systematic description of the various types of furnaces and heating apparatus, with a brief description of the correct and incorrect constructions.

They will be taken up in the following order:

I. Vertical Regenerators;
II. Horizontal Regenerators;
III. Hot-blast Stoves;
IV. Hot-blast Temperature Equalizers;
V. Iron Tube Hot Blast or Air Heaters;
VI. Steam Boilers;
VII. Chamber Furnaces, Brick and Pottery Kilns;
VIII. Cementation Furnaces;
IX. Annealing Furnaces for Malleable Iron;
X. Continuous or Multiple-chamber Kilns;
XI. Muffle Furnaces;
XII. Vertical Furnaces for Tempering, Annealing and Heat Treating;
XIII. Horizontal Tempering Furnaces;
XIV. Annealing and Heating Furnaces for Boiler Plates;
XV. Coal-fired Reverberatory Furnaces;
XVI. Siemens or Regenerative Heating Furnaces;
XVII. Pit Furnaces, Heating and Soaking Pit;
XVIII. Continuous-heating Furnaces;
XIX. Tunnel Furnaces.

I. VERTICAL REGENERATORS

The ordinary vertical regenerator is so constructed that it should work in a rational manner; that is, the high-temperature gases descend regularly and subdivide among the vertical passes according to their temperature, while the air and gas which is being heated subdivides into the ascending passes in the same manner. This is why the idea of Schenwelder, of dividing each of the checker chambers into two compartments, in order to secure uniform working, is fundamentally wrong. He introduced into regenerator construction a superfluous complication, and for this reason his design has been abandoned.

Nevertheless, it should not be concluded from the above that any sort of a vertical regenerator will always work in a regular manner. Vertical regenerators have an inherent tendency to work uniformly, but poor design may cause them to work in a very irregular fashion.

For example, Fig. 50 shows a very common form of regenerator, in which the current of gas from the vertical flues leading to the ports is jetted upon the top of the checkerwork. It is very evident that in this case it will be impossible to secure uniform operation. The checkerwork can only work in a uniform manner when sufficient space is provided below the arch over the chamber for the velocity of the gases to become zero or very close to zero. The eddies, which are formed under too low an arch, disturb the gas distribution and prevent the furnace from working uniformly. In order to secure uniform results it is necessary to provide sufficient space to absorb these eddy currents, in order that the gases may enter the checkerwork with a very low velocity below the arch over the chamber. As an example of a very good type of regenerator we have shown the chamber of a 40-ton open-hearth furnace at the plant of the Pennsylvania Steel Co., Fig. 51.

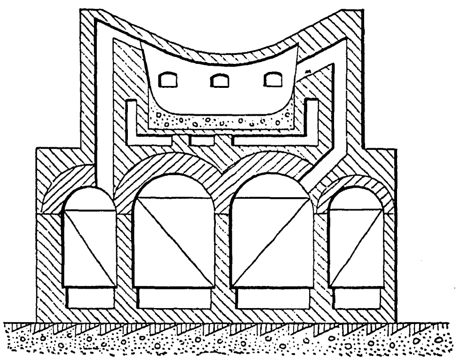


FIG. 50.

From time to time vertical regenerators are designed without regard to the fundamental principles that a current of hot gases in cooling should pass downward, and a current of air or gas which

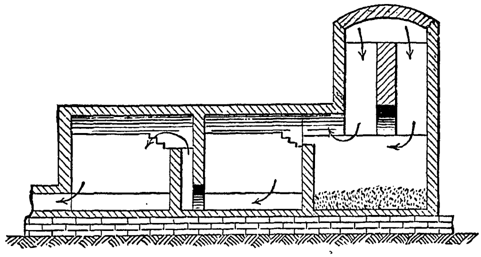


FIG. 51.

is being heated should pass upward. Figs. 52 and 53 show such construction.

In each of these cases the current of gases changes its direction of flow from upward to downward, or *vice versa*. Thus in Fig. 52 the gases which are giving off heat enter at the top and flow to the

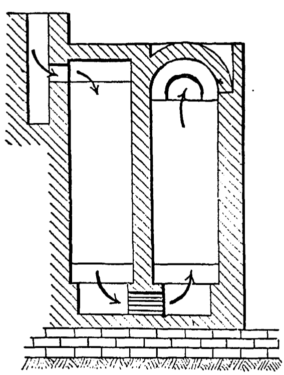


FIG. 52.

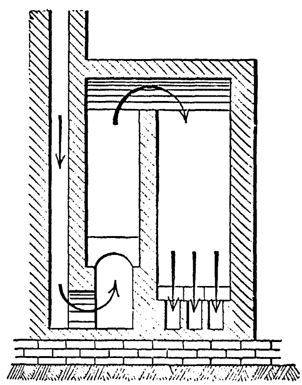


FIG. 53.

bottom of the checkerwork (in the correct manner) and then turn and flow from the bottom to the top of the second chamber (in an incorrect manner); in Fig. 53 the gases which are giving off heat enter at the bottom and rise to the top (in the incorrect way), then they flow downward (in the correct manner).

II. HORIZONTAL REGENERATORS

Horizontal regenerators have a bad reputation. There is nothing surprising in this, because in the design of this type of regenerator there is displayed a gross disregard of the rules governing the subdivision of gaseous currents. Take, for example,

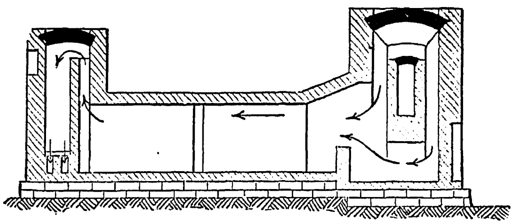


FIG. 54.

the regenerators of the celebrated furnaces designed by H. H. Campbell, which could not be forced to work well (Fig. 54).

In these regenerators the hot gases enter and pass out of the upper part of the chamber. It is evident that in this checker chamber the gases flow in an inverted channel and, if the depth of the flowing stream is less than the distance from the arch to the bottom of the chamber, they will not touch the latter.

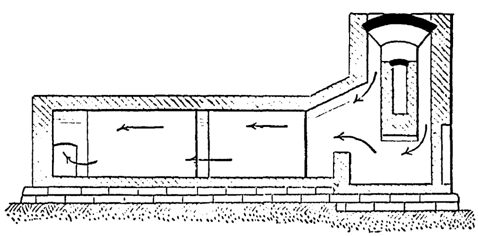


FIG. 55.

Note by English translator.—It may very readily happen that the conductive area of the checkerwork will be sufficient to cause the lower portion of the checker to reach a high temperature; but at the same time the frictional resistance to the flow of the gases and air will usually be so great that this design of checker will not work without forced draft. Horizontal recuperators are subject to this same defect and it is difficult to secure good results unless the passages are very short.

It is comparatively easy to correct these regenerators in a manner which will cause them to work better. Fig. 55 shows the correction or change which should be made. [This change, however, does not affect the frictional resistance of this type of checker work.] The hot gases enter the top part of the chamber and leave at the bottom; the height of the chamber h may be calculated according to the formula of Professor Yesmann in the same manner as the height of the flues leading to the reversing valves; in this way it is possible to design these chambers of such a height that the lower free surface of the current of gases will be at the level of the bottom of the chamber.

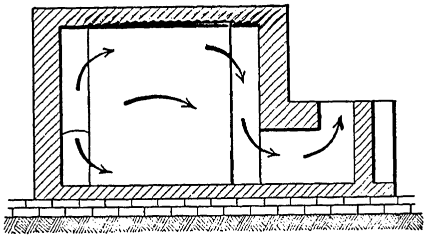


FIG. 56.

The checkers designed by Dalen (a well-known German designer of furnaces) show a construction equally erroneous (Figs. 56 and 57). Here the hot gases are conducted into the checker chamber, not at the top, but below the middle of the chamber. Such a regenerator heats the gas nicely (Fig. 56), but it works very inefficiently, because the heated gases are very light and the hottest gases are trapped in the upper part of the chamber and cannot pass out to the flues, while the upper portion of the checker does not do any work at all and forms a pocket filled with immobile heated gas.

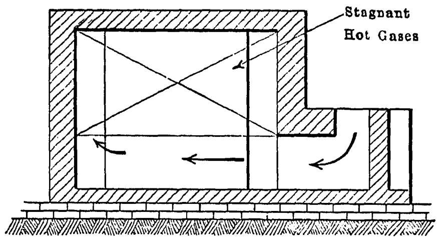


FIG. 57.

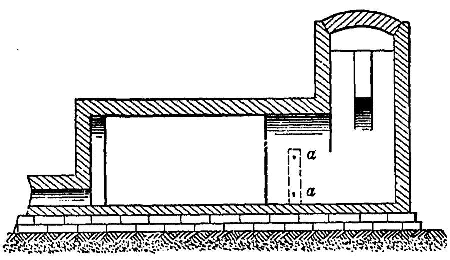


FIG. 58.

Fig. 58 shows a horizontal regenerator constructed in a rational manner. The wall *aa*, indicated by dotted lines, may be removed, as it is useless.

The construction shown by Fig. 51 is very good; in this there are two vertical regenerators placed side by side in place of a horizontal regenerator. The correct direction is given to the flowing gas currents. The main defect of these regenerators is due to the frictional resistance which they offer to the passage of the gases.

Regenerators in which the built-up checkerwork is replaced by vertical channels or cells, constructed in a manner similar to that used in the hot-blast stove designed by Withwell, are considered in the section upon "Hot-blast Stoves."

III. HOT-BLAST STOVES

The regenerative principles of Siemens were applied by Cowper in 1860 to the heating of the air for blowing blast furnaces.[1] At first, however, these hot-blast stoves were not a success. Percy Wedding, in his *Manual of Metallurgy*, shows some of the early designs of hot-blast stoves, one of which is given in Fig. 59. The first stoves were heated with coal fires; the gas from the blast furnace was not utilized until later. The hot products of combustion from the fire flowed directly upward to the top of the stove and passed out of the structure through a primitive chimney valve in the dome. The cold blast entered the apparatus at the top and

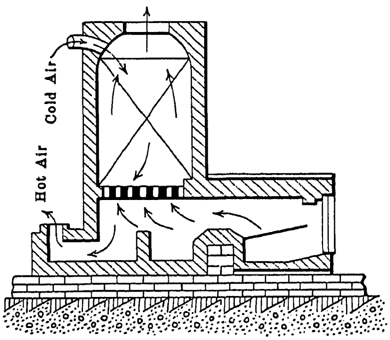


FIG. 59.

[1] *Note by translator.*—The early blast furnaces were simple open-topped shafts with no bell or hopper. The gas was burned at the head of the furnace, and the blast pressure was very low. The charging was very elementary; in many cases the top of the furnace was reached by an inclined plane up which the charge was carted in simple dump-carts which were backed up to the open shaft and dumped the charge into it. Later, when the bell and hopper were utilized to close the top of the furnace the gas became available for heating the blast and making steam. (The name, Hot Blast Stove, is a survival from the early coal-fired stove.)

the hot-blast valve was connected to the lower part of the stove. It may be readily seen that the currents of gas and air in this apparatus were circulated in the wrong direction; that is, in the opposite direction to the natural convection currents.

Cowper afterward perceived the error in the gas circulation in his early design, and in later designs the currents of gas and air were circulated in the correct direction (Fig. 70). Combustion took place in a central or eccentrically placed chamber, the hot gases rising to the dome where they were reversed and subdivided among a number of parallel passes down through the checkerwork. Uniting in a lower chamber, they were then carried away through the chimney valve. The cold blast entered the stove chamber below the checkerwork, through which it passed upward in a number of parallel streams to the dome of the stove. There it changed its direction and passed downward through the combustion chamber to the hot-blast valve. The Cowper hot-blast stove retains this general arrangement to the present day.

It would be difficult to find any apparatus which has passed through as many modifications of design as have these stoves; however, very few of these modifications have come into extended use, because practically all of them have been based upon an entirely false idea in regard to the laws governing the circulation of gases while heating and cooling.

The single weak point of the Cowper hot-blast stove lies in the location of the combustion chamber, which, in the form of a vertical tube or chamber, occupies a very large amount of space and is very poorly adapted for the purpose for which it is employed. The attempts to improve the design of these stoves should be directed toward the elimination of this tube for the combustion chamber. The space located immediately below the dome of the stove might be employed for a combustion chamber, as is suggested below. Moore's patent (Fig. 60) shows a single-pass construction of this character, but is badly worked out. In Moore's stove the hot gases of combustion pass upward, in the opposite direction to their natural convection currents.

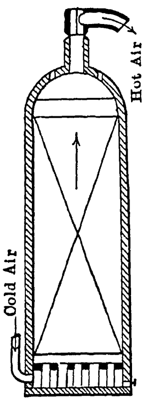


Fig. 60.

The first modification of the Cowper stove was made by

Withwell. The experiment shown in Fig. 61 illustrates the defect of the Withwell stove. Stoves of two designs were photographed, immersed in a glass tank filled with water, Withwell's design on the left and the Cowper design on the right. The period illustrated is that during which they are upon gas, or heating. The uniform level of the lower surface of the colored kerosene forced downward through the checker passes of the Cowper stove illustrates the regularity of the heating of its checkerwork. In the Withwell stove, on the other hand, the chambers are filled with the colored kerosene in a manner which is far from uniform. The chambers

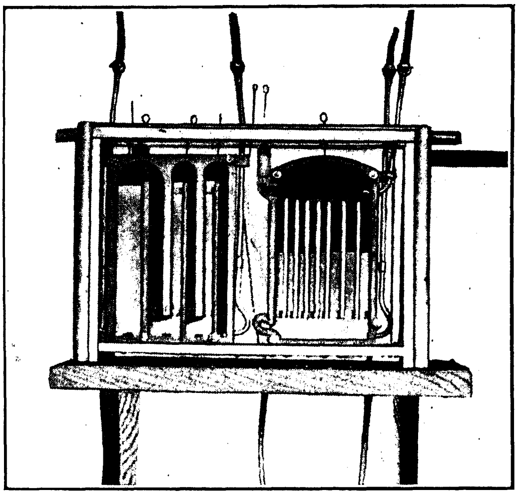


FIG. 61.

through which the kerosene descends are completely filled, while those in which it rises are not filled at all, the kerosene rising in a thread-like stream which comes into contact with a very small portion of the wall area of the chamber. These chambers, therefore, will not be uniformly heated; and for this reason will impart very little heat to the air. If water is introduced at the lower part of the model to represent the cold blast which is being heated, it will be seen that those chambers through which the water rises are completely filled, while in those in which the water passes downward it falls in small streams which have very little contact with the walls of the chamber. A mere glance at these models is sufficient to show clearly the reasons which have led to the complete abandonment of the Withwell stove.(1)

The ideas of Withwell have found their realization in the Massick and Crook stove, which had quite a name at one time, and which was only distinguished from the Withwell stove by the arrangement of the passes. This arrangement was such that the gases traveled alternately up and down through the three passes. The Massick and Crook hot-blast stove has now been superseded for reasons which will be clearly evident to the reader.

Many inventors have applied themselves to the problem of

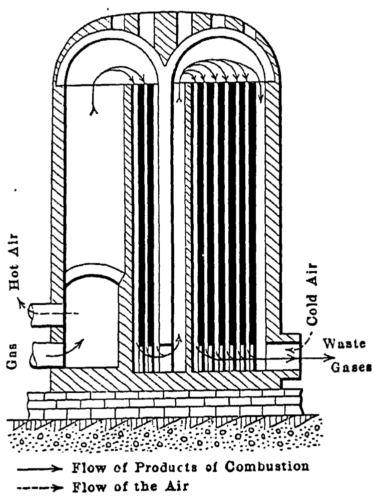


FIG. 62.

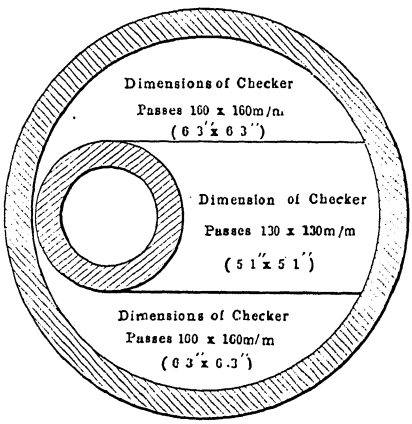


FIG. 63.

securing uniformity in the operation of the checkerwork heating surface and passes, apparently without the slightest suspicion that the regular and uniform descent of the hot gases in cooling is a natural property of the gases. For a time the design patented by Becker (Fig. 63) was much favored. In this design the openings in the checkerwork were given different dimensions, naturally

(1) Withwell himself noted the defects in his design, and in a later design (Fig. 62) he gave the hot gases a circulation in the proper direction, from top to bottom. This design approximates the type of regenerator shown in Fig. 51.

resulting in a less uniform heating than is secured in the ordinary design of the Cowper stove.[1]

In the model (Fig. 61) of the Cowper stove the cross-section was intentionally made with checker openings of several sizes, in order to show, in a graphic manner, that such complications of the checkerwork are useless.

The hot-blast stove patented by J. Z. Stephenson and J. Evans (Fig. 64) has never come into any extended use. In order to insure a uniform distribution of the hot gases through the checkerwork the inventors resorted to a series of complicated walls and dampers in the chamber below the checkers.

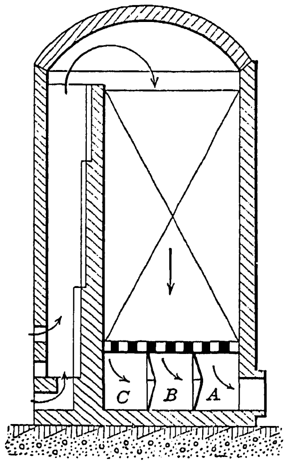


FIG. 64.

The discussion carried on in the technical journals, between Luhrmann and these inventors, is mentioned merely for its historic interest. Luhrmann contended that the largest portion of the waste gases passed through the section above *A* and the smallest portion through the section above *C*. The inventors claimed the contrary. This discussion made it very clearly apparent that the ideas of contemporary constructors, in regard to the circulation of the gases, were confused to a most remarkable extent. The quarrel was finally settled to the satisfaction of both parties.

In the Cowper stove the work of the checkerwork is inherently uniform, and all walls and dampers below the checkers, for the purpose of distributing the gases, are superfluous.[2]

(1) This design will give less frictional resistance in the large openings than in the small; accordingly, the waste gases from the large passes will be hotter than the gases from the smaller passes.

(2) In the *Revue de Métallurgie*, for February, 1913, p. 362, can be found the designs of a Cowper stove built for the furnace at Caen, France, and provided with five chimney valves connecting the chamber below the checkerwork with a bustle pipe outside the stove. The idea was, of course, that this would conduce to the uniform distribution of the gases heating the checkerwork.

This idea, in many ways, resembles that of Stephenson and Evans, mentioned above. The multiplication of the chimney valves is useless. It would be sufficient to replace them by a single valve whose dimensions may be computed by Yesmann's formula.

In a very large number of designs for hot-blast stoves the attempt is made to circulate the hot gases in cooling and the cold gases in heating, in directions contrary to their natural convection currents.

The accompanying sketches of hot-blast stoves—Hartmann (Fig. 65), Hugh Kennedy (Fig. 66), Macco (Fig. 67), Frank Roberts (Fig. 68), Harvey (Fig. 69)—show that the laws governing the subdivision of currents of hot gases while cooling and cold gases while heating are not very well understood at the present time.

Construction of the Combustion Chamber for a Hot Blast Stove.—In the statements which follow, the attempt has been made to

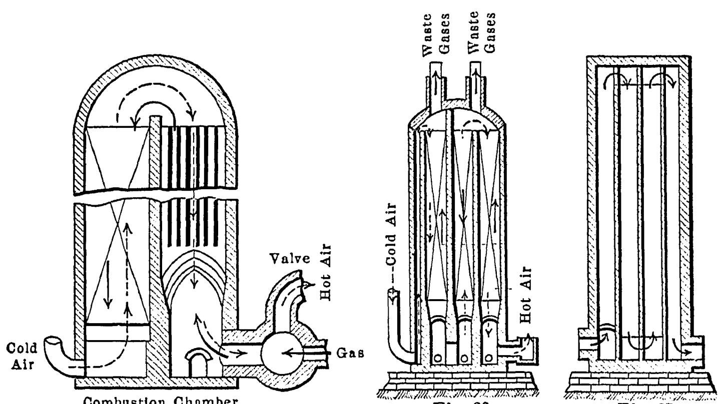


Fig. 65 Fig. 66 Fig. 67

⟶ Flow of Products Combustion
- - -> Flow of Air

show the rational location of the combustion chamber for a hot-blast stove. For this purpose it is necessary to digress slightly at this point, in order to set forth the rational chamber conditions which are necessary for combustion.

If a flame or jet of burning gases in which the reaction of combustion has not been completed is directed into a cold chamber or upon cold objects, combustion will not be completed, even in the presence of an enormous excess of air. *Theoretical combustion,* without an excess of air, can only be obtained when the reactions of combustion are completed in a chamber where the flame is surrounded by incandescent walls and within which it is held for one or two seconds. The construction of the combustion chamber must be such that it is possible to *hold* the flaming gases in the chamber for a predetermined length of time, and that the walls of the chamber will be heated *to the highest possible* temperature.

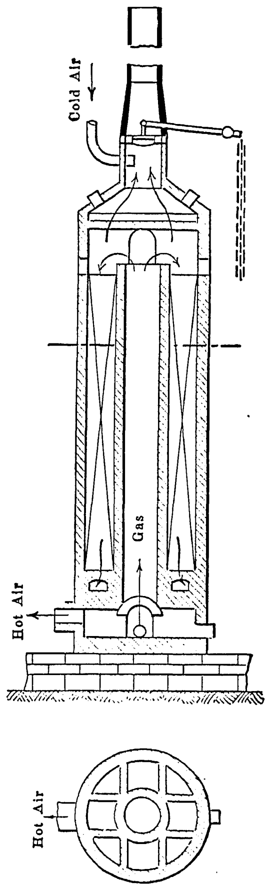


Fig. 68

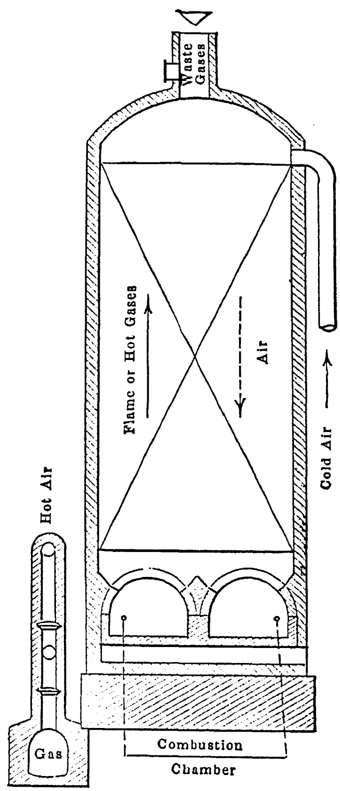


Fig. 69

——→ Flow of Products of Combustion

- - - → Flow of the Air

How should the flame of reacting gases be brought into the combustion chamber?

In the Cowper stove (Fig. 70) the combustion chamber forms a cylindrical or elliptical opening extending the full height of the stove and open to the space below the dome at its upper end. The volume of the combustion chamber is very large.[1] Nevertheless, it does not function in a satisfactory manner, because the streams of flaming gases have a very high temperature and,

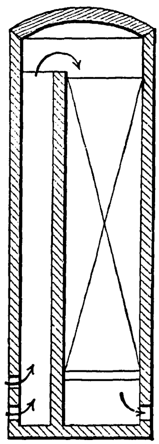


FIG. 70.

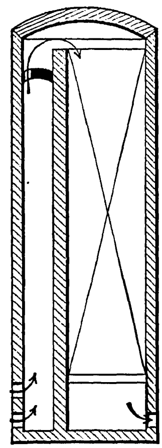


FIG. 71.

accordingly, a very slight density. For these reasons, the stream of flaming gases rises very rapidly, leaves the combustion chamber

[1] Guilow, in an article (*Revue de la Société russe de Métallurgie, 1911, p. 164*) gives the volume of the different portions of a Cowper hot blast stove at the Kouchwa works, and the time the gas remains in these portions, as follows:

	Volume Cubic Meters	Time, Seconds
Combustion chamber	14.53	4.60
Dome of stove	6.16	1.77
Checkerwork openings	42.40	16.96
Chamber below the checkerwork	5.14	5.14
Totals	68.23	28.47

Note by translator to English.—It is rather interesting to compare these volumes with the volumes of hot blast stoves given in a paper by Arthur J. Boynton, National Tube Co., Lorain, Ohio, before the October, 1916, meeting of the American Iron and Steel Institute. The Russian furnace is undoubtedly much smaller than the American furnaces.

without encountering any obstruction to delay it and flows into the space below the dome. Accordingly, the rising column of flaming gases passes upward without filling the combustion chamber and is surrounded by an atmosphere of unburned relatively colder gases, through which it passes. Direct observation of the combustion of blast furnace gas in the Cowper hot-blast stove shows that, as the ratio between the gas and the air supply approaches the theoretical requirements, combustion ceases to be silent and becomes noisy. The flame commences to jet out around the gas burner in bursts, indicating a temporary extinguishment of the flame in the combustion chamber, which is immediately filled with a comparatively cold explosive mixture.

Everything indicates that the correct location for the combustion chamber of the Cowper stove is the dome or space above the checkerwork. The streams of flame which are produced cannot pass directly out of this chamber, but are held under the dome, heating its walls from below. The dome forms a permanent firebox for their combustion. Below the dome of the Cowper stove, therefore, there is a stationary hot zone, which burns those portions of the gases which have not been utilized in the combustion chamber, as soon as they come into contact with the heated chamber. For these reasons the construction of a Cowper stove with the combustion chamber located elsewhere than in the dome is fundamentally wrong.

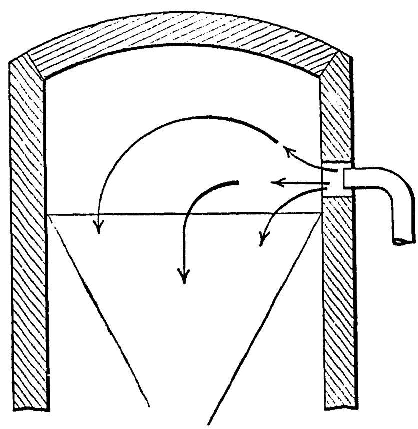


FIG. 72.

However, if the gas and the air are simply introduced above the checkerwork of the stove, as is shown in Fig. 72, there is evidently danger that a portion of the cold gas and air, in place of entering into the reaction of combustion and forming a flame, will drop down to the bottom of the stove, through the checker openings, just as a heavy liquid will sink downward through a light liquid. In order to prevent this occurrence, which is undesirable and wasteful, it is useful to direct the currents of gas and the cold air for their combustion upward toward the dome of the stove and give the combustion chamber an arrangement similar to that shown in Fig. 73. There are many advantages in constructing a hot-blast stove in this manner. In the Cowper hot-blast stove the combustion chamber actually occupies nearly 35 per cent of the space inside of the shell and causes an unnecessary increase of nearly 50 per cent in the cost of the stove.

The analysis of this particular case of combustion-chamber construction permits the following rules to be deduced:

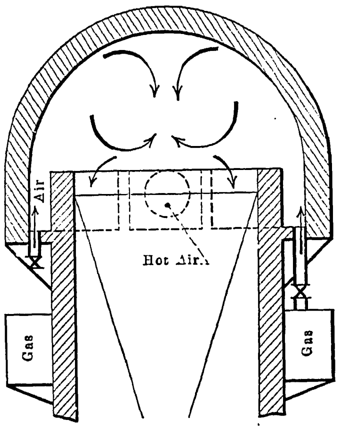


FIG. 73.

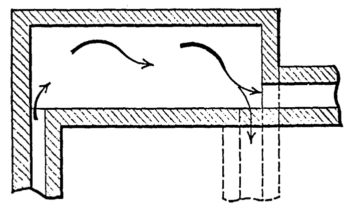


FIG. 74.

1. The jet of flame in the chamber should not be directed horizontally.

2. The streams or jets of gas and air entering the chamber should be, where possible, directed upward toward the dome or roof.

3. The products of combustion should be carried away from the combustion chamber at the hearth level, in either a vertical or a horizontal direction, as shown in Fig. 74.

Nevertheless, it is sometimes necessary that a vertical combustion chamber should be used. In this case, it is not necessary to repeat the error made by Cowper and construct the chamber as an *open pit* (Fig. 70). It can be covered by an arch pierced by one or

Note by English translator.—Although Professor Groume-Grjimailo announced his theories over ten years ago they have not become very widely known. Confusion and lack of knowledge still prevail among the designers of hot-blast stoves. The direction of the natural convection currents of hot gases in cooling and of cold gases in heating are ignored by the constructors of three- and four-pass stoves.

more openings; the total area of these openings may be arrived at by the use of the formula given in an earlier chapter of this work, for furnaces having an orifice in the roof for the escape of the waste gases.

If the area of these orifices and the volume of the combustion chamber are correctly proportioned, the combustion pits of the Cowper stove will be completely filled with the burning gases, which will be held in the combustion chamber by reason of the strangulated outlet, until the reaction of combustion is completed. The free space below the dome of the stove will cease to play the part of a combustion chamber, and it will become possible to carry the checkerwork up higher into the dome of the stove.

IV. HOT-BLAST TEMPERATURE EQUALIZERS

It is not difficult to show that the construction of a rational apparatus for the equalization of hot-blast temperatures is not practicable.

Fig. 75 shows the general arrangement of such hot-blast equalizers as have been constructed. Assuming that the temperature of the blast is higher than that of the brickwork in the equalizer, the branch *A*, in which the hot gases rise, will work *irregularly*, whereas the branch *B*, in which the hot gases pass downward, will be heated *uniformly*. The branch *A* will not heat in a satisfactory manner; the branch *B* will heat in a satisfactory manner.

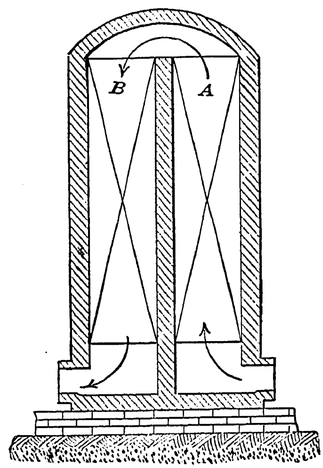


FIG. 75.

Assuming that the temperature of the hot blast drops below the temperature of the brickwork, the branch *A* will commence to cool uniformly and rapidly, whereas the branch *B* will cool irregularly and slowly.

No one has been able to obtain satisfactory results from the use of hot-blast temperature equalizers, and for this reason they are very rarely used.

V. IRON TUBE HOT-BLAST OR AIR HEATERS

This type of apparatus is now practically unknown, except for the description of it which may be found in various books.

In the *Manual of Metallurgy*, by Percy-Wedding, twelve types of iron tube air heaters are described, some dozen or so of pages being devoted to their construction and the causes which led to their being abandoned. Of the many types of this apparatus, the only ones which have survived are those of the *Bessèges* works and the Cleveland type; all of the others have passed out of use.

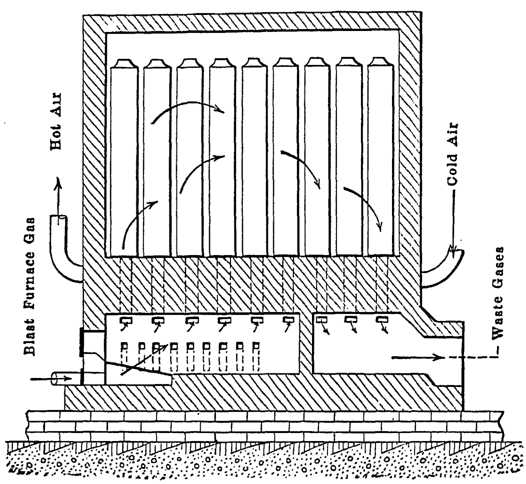


FIG. 76.

The reason for their abandonment is very simple: all of the iron tube air heaters described by Percy-Wedding had the outlet for the waste gases at their highest point. The hottest gases of combustion rose immediately to the top of the chamber, licking the surface of the iron tubes, and heating them irregularly, the top being much hotter than the bottom. The tubes, heated in this manner, burned out, warped and broke.

In the *Bessèges* type of air heater the products of combustion pass from chamber to chamber through ports in the division walls. These ports are located at the level of the hearth. In this apparatus the reacting portion of the flaming gases rise and then drop through zones of uniform heating, where combustion is completed.

These local centers of combustion are due to the excess air supply. In proportion as they cool, the gases drop lower and lower, uniformly heating all of the tubes, which for this reason work in a satisfactory manner.

The Cleveland iron tube air heater, which is not so widely known, likewise works upon the *downdraft principle* (Fig. 76).

The disappearance of the numerous types of iron tube air heaters and the survival of the *Bessèges* and Cleveland designs supplies a very good example of the importance of giving the correct direction to the circulation of the gases in furnaces.

VI. STEAM BOILERS

The constructors of steam boilers very rarely consider the rational distribution of the hot gases. These defects are particularly frequent in the most recently designed types of water-tube boilers, as well as in the older designs. The lack of knowledge of the laws governing the flow of the heated gases explains the

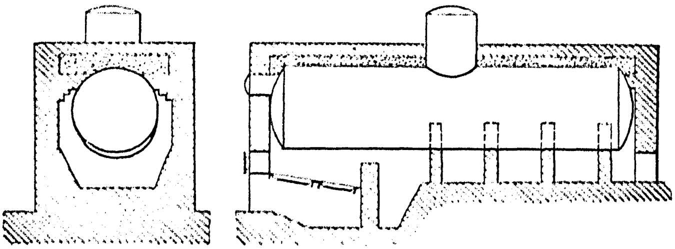


FIG. 77.

numerous and complicated forms of baffling of all kinds, as well as the use of special dampers to force the hot gases to bathe the heating surface of the boiler and its tubes regularly and completely.

(*Note by English translator.*—The absolute neglect and disregard of the most elementary laws of physics is not confined to the circulation of the heated gases, but is grossly violated in the water circulation as well.)

In reality it is not necessary to have any baffling nor walls. The hot gases have a natural tendency to flow in such a way that the entire heating surface of the boiler will be bathed by them in a very uniform and regular manner. Without going into details, some of the useless and erroneous forms of boiler setting and baffling which are given as "good" construction in *Hütte* are shown below. These will very clearly indicate the ideas which the author is endeavoring to set forth.

Fig. 77 shows a single-drum cylindrical boiler as illustrated upon page 865 of the first volume of the French translation of *Hütte* (edition of 1911).

Fig. 78 shows the correct method of setting the same boiler. All of the baffling which was supposed to force the hot gases to travel in a zigzag path has been eliminated. The hot gases, being light, have a tendency to rise and apply themselves to the shell of the boiler. Any constrictions of their path have a detrimental effect, as they tend to increase the velocity of flow. In order to

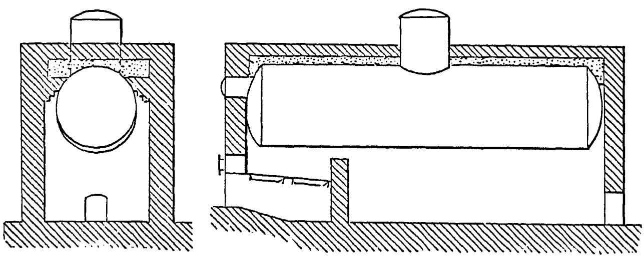


FIG. 78.

increase the time during which the heated gases remain in contact with the boiler after leaving the firebox, the waste gas outlet has been dropped as far as possible by lowering the hearth of the gas chamber behind the bridge wall.

(*Note by English translator.*—This setting would be greatly improved by placing the grate and firebox in a separate setting where the mixing gases would not be chilled below the ignition point.)

The circulation of the gases in a boiler set in this way is effected in the following manner: The boiler will be constantly bathed by the hottest gases which tend to rise to the highest part of the setting. As these gases are cooled they will drop lower and lower by reason of their increase in weight while cooling. They will finally, at their lowest temperature, fall to and pass out through the waste gas outlet.

In order to avoid detrimental eddy currents it is necessary to fix the height of the opening over the bridge wall by the formula for the inverted weir and avoid high gas velocities.

Fig. 79 shows the setting of a four-drum cyclindrical boiler,

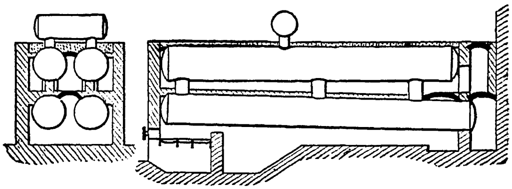


Fig. 79.

having the drums arranged in pairs above each other, according to *Hütte*. The cross-section of this setting shows that the heating chamber is divided into four flues for the hot gases:

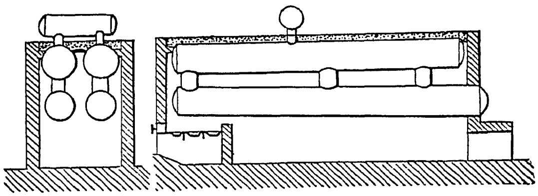


Fig. 80.

1. One passage under the lower drums;
2. Two passages at the sides of the two upper drums;
3. One passage between the two upper drums.

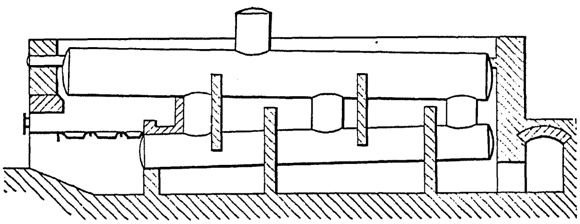


Fig. 81.

A very large proportion of the heating surface of this boiler is covered and insulated by the numerous arches and baffles, making it useless. It is very simple and easy to change this setting by the removal of the walls and arches, as is shown in Fig. 80, at the same time lowering the waste gas outlet to the level of the bottom of the gas chamber where it will remove the coolest gases in the setting.

Fig. 81 shows a two-drum cylindrical boiler (*Hütte*). In this setting, those baffles which are built up from the bottom of the setting are entirely useless and may be removed. This setting will be better if built as shown in Fig. 82.

These three examples will serve to show very clearly the manner in which boiler settings may be greatly simplified. Nevertheless, it is well known that commercial boilers, particularly those designed for use upon ships, such as the Belleville, Niclausse,

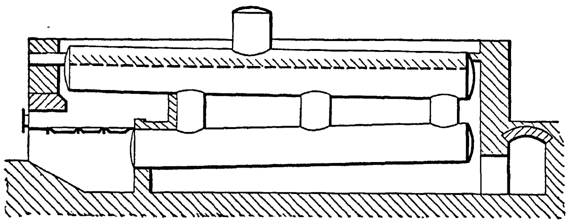


FIG. 82.

Yarrow, etc, are designed to work upon the updraft principle. This is a serious error, as it violates the law of gaseous flow, and as a result there is poor vaporization and a reduction in the efficiency of the application of the heat.

VII. CHAMBER FURNACES, BRICK AND POTTERY KILNS

Barely thirty years ago, direct or updraft kilns were practically the only kind used in the brick and pottery plants. At present their use is decreasing and there is a strong preference for the downdraft kiln.

It may be said that the direct or updraft kilns which are still in service are the last traces of these kilns in this industry. At the same time it is very curious to note that the firm of Ernest Schmatolla,[1] which is engaged almost exclusively in the construction of brick and pottery kilns, in a book published by them, devote almost the entire volume to a description of the old updraft kilns, and make no mention of the downdraft kiln beyond the brief statement that these kilns generally give better results than the updraft kiln.

[1] *Ernest Schmatolla, Die Brennöfen,* 1903.

In a preceding chapter the principles governing the computation for the updraft chamber furnace have been stated. By partially closing the smoke hole in the dome it is possible to force the free lower surface of the hot gases in the kiln chamber down to the hearth level, and thus force the heating of ware placed upon the hearth. Nevertheless such a method of heating is very imperfect and is not uniform. Its mechanics are as follows:

The incandescent and burning gases issuing from the firebox (Fig. 83) rise immediately to the highest point in the chamber; therefore the central portion of the heating chamber does not receive any direct action from these ascending currents, and is accordingly filled by heavier and colder gases than those coming from the fireboxes. These heavy gases gradually drop to the hearth of the chamber as the upper parts of the chamber fill with the hotter gases. Portions of these colder gases in the bottom of the chamber become mixed with the heated gases flowing from the fireboxes and rise with them. Finally a circulation of gases is established within the chamber, so that all portions of the charge are gradually heated.

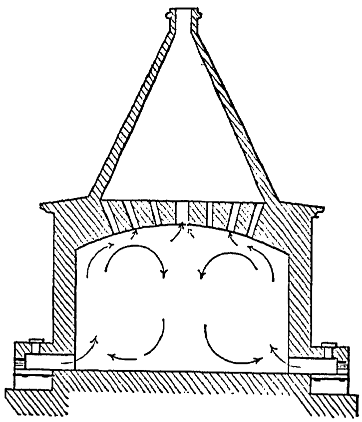


FIG. 83.

Moreover, only a small portion of the heated gases circulate in this manner. The largest portion of the hottest gases escape immediately to the chimney through the smoke holes in the roof of the kiln. For these reasons, all the ware, which is set where it comes into direct contact with the hot gases is hard burned, whereas the burn of the ware which only comes in contact with the colder currents of gases will be less hard and portions of the ware will receive a very slight burn.

While updraft or direct-draft kilns continue to be used in many of the clay products plants, the principles of the downdraft kiln

Note by English translator.—Considerable heat is carried down to the hearth of the kiln by conduction through the charge and the walls.

are so well known in this industry that it will not be necessary to devote much space to them.

Fig. 84 shows the same kiln (Fig. 83) *reconstructed* to work on the downdraft principle. The distribution of the currents of hot gases is very good, as the hottest gases rise to the arch of the furnace and then divide themselves into uniformly descending currents. One of the results of this reconstruction was the reduction of the amount of defective brick turned out from 30 per cent to 1 per cent.

Fig. 85 shows a two-story kiln used in the manufacture of

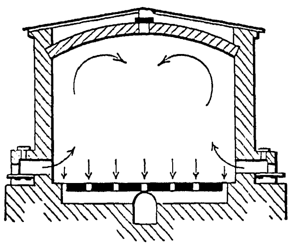


FIG. 84.

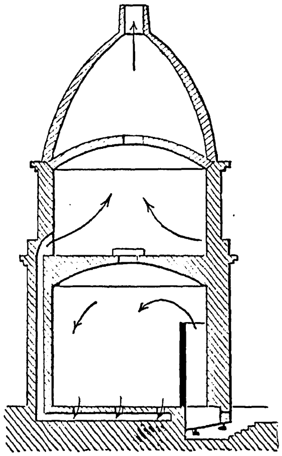


FIG. 85.

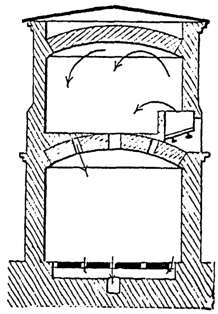


FIG. 86.

porcelain. Such kilns are still actually in use. The upper chamber works in an unsatisfactory manner, the hot gases being divided into ascending currents; it is practically impossible to regulate the distribution of the heat by changing the size of the waste-gas opening. The lower chamber of the kiln, which works on the downdraft principle, has a regular heat distribution.

Fig. 86 shows a kiln of this type correctly constructed, both the upper and the lower chamber working upon the downdraft principle. The firebox has been changed from the lower to the upper chamber.

Note by English translator.—Downdraft "beehive" kilns are widely used in the manufacture of refractory brick. In the silica brick plants it is well known that only one-seventh of the kiln capacity can be used for coke oven shapes, many of which require two burns. The prevailing tendency in building larger and larger kilns is merely to increase the height and diameter. However, there are limits upon the height, due to the crushing of the lower tiers and the difficulty of securing an even burn throughout the kiln. These difficulties do not appear in the small kilns.)

VIII. CEMENTATION FURNACES

Fig. 87 shows an old type of cementation furnace cited in the lectures of Professor Ledebur. The hot gases rising from the

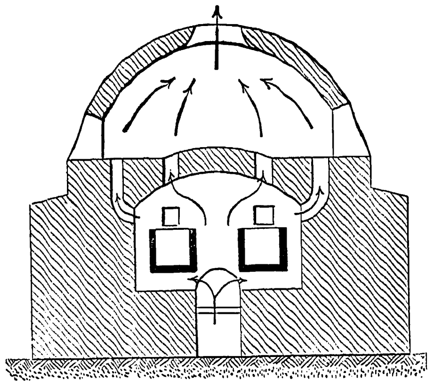


FIG. 87.

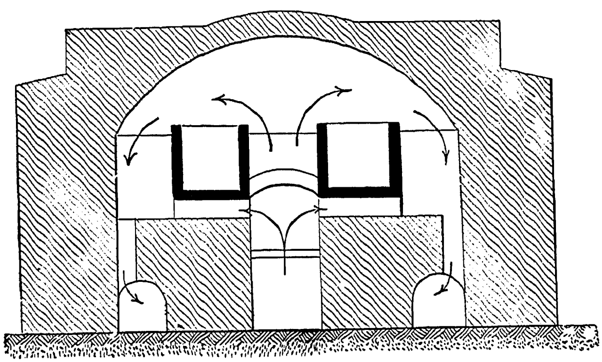


FIG. 88.

firebox are subdivided immediately into ascending currents, which are supposed to envelop and surround the boxes containing the iron bars packed in charcoal. Much better results would be obtained by reconstructing this furnace upon the downdraft principle, as shown in Fig. 88.

IX. FURNACES FOR ANNEALING MALLEABLE IRON CASTINGS

Fig. 89 shows the old style of annealing furnace for malleable-iron castings, according to Professor Ledebur. Fig. 90 is from the patent of P. Schnie in 1898, for the same purpose, but upon the downdraft principle. It is evident that it was practical experience

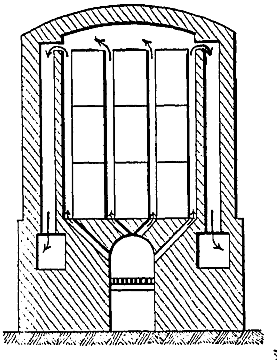


FIG. 89.

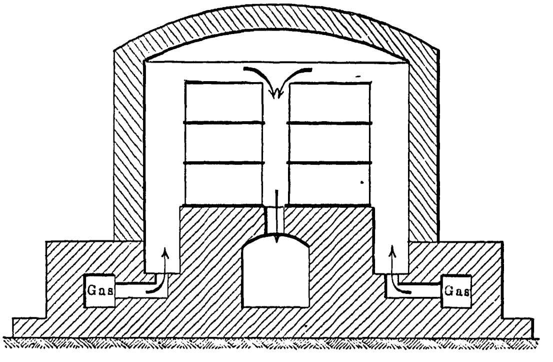


FIG. 90.

and not theory which led the inventor to apply the downdraft principle to these furnaces.

X. CONTINUOUS OR MULTIPLE CHAMBER KILNS RING FURNACES

Ring furnaces or kilns, notwithstanding the apparent perfection of their design, have never worked in a satisfactory manner. Their main disadvantages are the following:

Too rapid cooling of the burned ware;

The resultant cracking and spawling of the ware, which in turn results in a considerable loss of product;

An excess of air during combustion, resulting in a very sharp flame;

The high temperature of the waste gases;

The large amount of fuel required for burning the ware.

All of these defects are due to a disregard of the principles which govern the subdivision of currents of gases when heating or cooling.

This may be clearly shown by the consideration of a simple case (Fig. 91), where A is the portion of the tunnel chamber in which the air is preheated, B the portion of the chamber in which the fuel is introduced and where the ware is burned, C the portion of the chamber in which the ware is preheated by the waste heat, E the flue connection to the chimney and D the portion in which the ware is being set.

The sections may be considered in order, starting with section B of the tunnel, where combustion takes place and the ware is burned. The gas circulation which is to be established in this section must be such that the hot gases flow to all parts of the tunnel chamber being heated. The hot gases, however, imme-

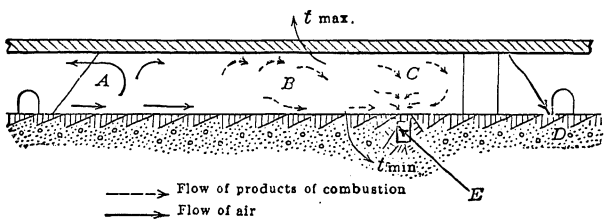


FIG. 91.

diately rise to the roof of the tunnel and then flow to where the cooling gases are drawn down into the waste-gas flue. The placing of the waste-gas flue under the hearth of the tunnel is, in general, favorable to the circulation of the heated gases, as it apparently works upon the downdraft principle. Nevertheless, it is not possible to obtain a uniform burn to the ware at the bottom and the top of the tunnel chamber. The upper part of the tunnel is filled with a stream of light gases, *the superheated products of combustion*, having a high temperature t_{max}, while the hearth of the tunnel is traversed by a *current of waste gases* which are heavier and at a lower temperature t_{min}. In order that the difference between the temperatures t_{max} and t_{min} shall be as small as possible it is necessary to set the ware to be burned in a compact checker which will increase the resistance to the flow of the gases in the upper part of the setting and to decrease the resistance to the flow of the gases in the lower part of the setting by the use of an open checker.

By this artificial means it is possible to pass the greatest portion of the hot gases near the bottom of the setting and in this manner the temperatures t_{max} and t_{min} may be very nearly equalized. Therefore, in the portion C of the tunnel chamber the checkerwork of the ware set to burn should be *set close at the top and open at the bottom.*

The portion A of the tunnel chamber will be considered next. This portion is filled with hot ware cooling, and through it is passed the current of cold air to be heated. This air is a cold and heavy fluid which will flow in the lower portion of the setting without any tendency to rise. The only way in which this stream of air can be forced to flow in the upper portion of the tunnel chamber is to increase the resistance to its flow by the checker work built in setting the ware in the tunnel, which for this purpose should be *set close at the bottom and open at the top,* where less resistance is required. Upon comparing the conditions required in section A with those required in section C of the tunnel chamber, it will be noted that they are diametrically opposed to each other.

It is true that the contraction of the brick or lime, in burning, favors the correct method of operation in these sections. Since, in the section of the tunnel chamber A, an open space will be formed below the roof of the tunnel, due to the skrinkage of the charge, this open space will be quite large in those kilns used in burning lime. But in general, the operation of ring tunnel kilns presents a complicated problem, to which there are two possible compromise solutions:

1. One solution secures an even heating of the crude material, but the incoming air is poorly preheated and the cooling ware is subjected to a sharp drop in temperature which is very liable to result in cracked and spawled ware;

2. The other solution secures a uniform preheat of the incoming air and results in the gradual cooling of the burned ware, but at the same time results in a non-uniform heating of the unburned ware.

The first of these methods of operation is the one ordinarily employed. For example, at the Wachter works at Borovitch, a kiln of this type is used for burning firebrick. The lower part of the tunnel is charged with lumps of fire clay whose quality will not be injured by sudden cooling. An open checker is built on the hearth of the tunnel with these lumps of clay, and on top of this the brick to be burned are set in a close checker in the upper part of the chamber. In this way the current of hot gases, which tends to seek the highest point of the tunnel chamber, is retarded in its flow and forced downward; the gases cool and produce a current of colder waste gases over the hearth of the tunnel. The brick which fill the upper part of the chamber are uniformly

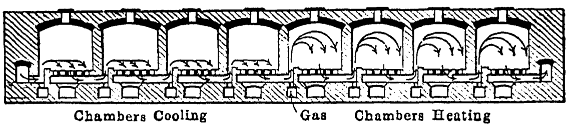


FIG. 92.

heated and burned, but the utilization of their heat for the preheating of the air is very poor and the kiln operates with very little preheat of the air supply.

The construction and design of ring tunnel kilns is therefore defective, because the currents of air being heated and those of gases being cooled have the same direction of flow, whereas it is necessary that the current which is giving off heat should flow downward and that the current which is being heated should flow upward. A ring tunnel furnace is ordinarily unable to satisfy

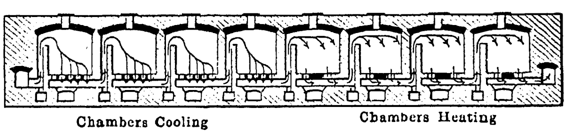


FIG. 93.

this condition, and for this reason its work will always be unsatisfactory.

The existing systems of ring chamber furnaces or kilns of the Mendheim and other similar types are but little more satisfactory than the ordinary types of kilns. Figs. 92 and 93 show two types of the Mendheim ring chamber kiln, the latter of these having a high bridge wall. The outside air enters the system through the chamber from which the burned ware is being removed, and passes in succession through all of the chambers which are cooling.

The course of the air through these chambers is indicated by the arrows; the colder air tends to settle upon the hearth of the chamber. It causes sudden and irregular cooling of that portion of the setting through which it tends to flow, and at the same time is not preheated to a high temperature and assists very little

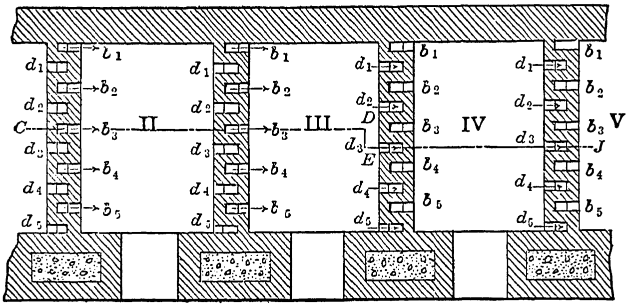


FIG. 94.

in the cooling of the setting. On the other hand the heating of the setting is effected by gases which traverse these chambers in the correct manner.

Due to the influence of the author's work, a Russian engineer, K.-K. Adametzky (Varsovie), has secured a patent for the improvement of the continuous chamber kiln by introducing supplementary channels which connect the lower part of each chamber with the upper part of the adjoining chambers. In this way it is possible to realize the flow principles governing the heating and cooling of gases in the operation of these kilns. However, the construction is slightly complicated, and the author has succeeded in finding a simpler method of accomplishing this result.

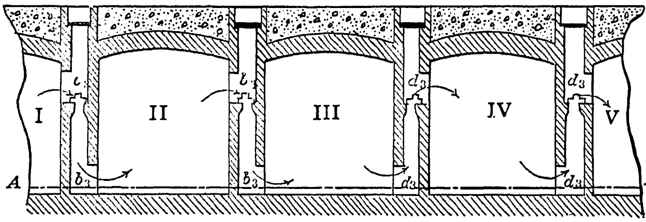


FIG. 95.—Section *CDEJ*, Fig. 94.

Figs. 94 and 95 show sketches of this type of kiln. Combustion occurs in chamber III; the air is being preheated in chambers I and II, and the bricks are being preheated in chambers IV and V. The gases which are cooling have a downward flow, while the air which is being heated flows upward; the circulation of the gases is therefore correct. The change in direction of the flow of the gaseous currents is effected by a system of channels or flues in the walls separating the chambers: channels b_1b_1, b_2b_2, b_3b_3 . . . for the air and d_1d_1, d_2d_2, d_3d_3 . . . for the products of combustion. When the products of combustion are passing into a chamber (for example, IV), the flues *dd* are open and the flues *bb* are closed. The opening and closing of the flues *bb* and *dd* is accomplished in a very simple manner by the damper brick shown in dotted lines.

The kiln which is designed according to this method is very simple in construction, and the author will enter into correspondence with any reader who may be interested in it. From this short description of the ring tunnel or chamber kiln it may be seen that it is not difficult to construct a continuous kiln which will operate in a satisfactory manner through the rational flow of the air and the products of combustion.

XI. MUFFLE FURNACES

For tne annealing and tempering of steel, the annealing of brass and bronze and for other purposes, muffle furnaces possess many advantages. A uniform temperature, the absence of jets of flame, the small amount of oxidization and the protection of the surface of the material being heated (a condition very important in the stamping of metals) make these furnaces a type much favored in these works, despite the fact that the muffles deteriorate quite rapidly. If the currents of hot gases are circulated in a rational manner, muffles of steel castings will give a service of several years. Unfortunately, the great importance which attaches to the correct circulation of the hot gases is not well known.

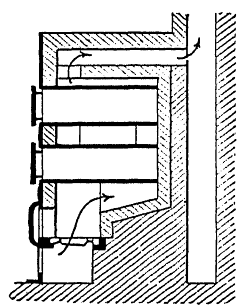


Fig. 96.

Fig. 96 shows a muffle furnace which is widely used. The flame and hot gases are produced in a firebox below the muffles and have an upward direction of flow. The heating of these muffles is therefore not uniform and they wear out very rapidly.

Fig. 97 shows the method of reconstructing this furnace to conform to the rational principles of the circulation of gases. For this purpose the hot gases enter the chamber at the top and pass downward, and the waste-gas flue at the bottom of the chamber in which the muffles are set draws off the coolest gases and passes them to the chimney.

It is by reason of the incorrect direction of the flow of the hot gases that the vertical muffles used for the tempering of shells

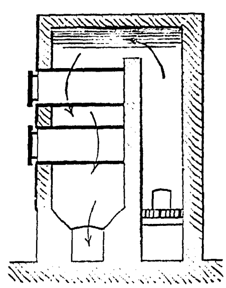


FIG. 97.

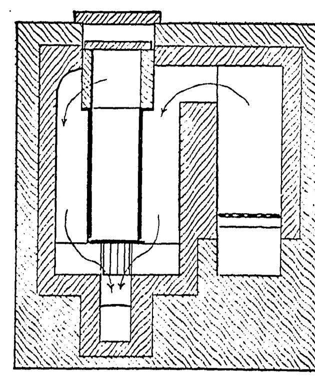


FIG. 98.

for artillery rarely work well. Fig. 98 shows a manner of setting these muffles which has given good results in the author's practice, because it gives the correct circulation of the hot gases. The muffle furnace shown has worked perfectly.

Attention is also called to a horizontal muffle furnace in the Petrograd Arsenal which was at last reconstructed as directed by the author and which worked perfectly thereafter.[1] Originally this muffle furnace, which operated with updraft, had four fire-boxes furnished with dampers; the muffle was of cast iron, 3560 mm long, 1000 mm wide and 100 mm high. It did not operate in a satisfactory manner. By direction of the author it was transformed into a downdraft furnace without any alterations

[1] *Revue de la Société russe de Métallurgie No. 5, 1913; Revue de Métallurgie, XI bis, p. 322, mai 1914; Appendix V.*

to the muffle, and since then has operated in a satisfactory manner, the following points being worthy of note:

1. In spite of the large size of the muffler, its temperature was absolutely uniform, and no differences of temperature could be detected between the different portions of the furnace, by the use of a Le Chatelier pyrometer;

2. By the operation of a damper, this temperature could be regulated very accurately within a wide temperature range (900° to 400°);

3. The quantity of fuel consumed (55 kg 80 per hour) corresponded exactly with the computed fuel consumption.

The only trouble with this furnace resulted from the intermittent operation to which it was subjected. This caused bending and warping of the muffle which failed completely after about two months' service, necessitating replacement. That this warping was entirely due to the intermittent operation of the furnace is borne out by the fact that a vertical muffle for practically the same class of work, installed in a furnace in the Poutiloff works, which is operated continuously, lasts about fourteen months.

In order to eliminate the trouble in the furnace at the Petrograd Arsenal the muffle was removed and the openings in the hearth for the removal of the waste gases were relocated. This furnace works very nicely for tempering and annealing, but it is no longer a muffle furnace. Finally the furnace was reconstructed with a new muffle constructed of special thin bricks, and since that time no operating troubles have developed.

XII. VERTICAL FURNACES FOR TEMPERING, ANNEALING AND HEAT TREATING

The application of the downdraft idea has nowhere given such brilliant results as when applied to vertical furnaces, and there is no other form of furnace which gives so little satisfaction and so much trouble as a vertical tempering furnace. The reason for this is very simple. In every works "the heat-treating department" is the "forbidden ground" of the plant. Secrets, secrets and secrets! As a matter of fact, the men who possess these secrets always know less than those who do not possess them.

Nevertheless, tempering and annealing furnaces only require a working temperature varying from 800° to 1000° and—good or bad—they may be operated. With regard to the heat-treating furnaces it may be said that few of them are good and that the process of heat-treating which is actually in use is less satisfactory than the tempering process.

For heat-treating or temper-drawing furnaces, the temperature must be maintained at less than 700°, that is to say, at a temperature at which the reactions of combustion can scarcely take place, and for this reason the gas must be burned outside the furnace; after this it is necessary to cool it in a special combustion chamber, to the temperature required. It is then passed into the heating chamber of the furnace and heats this last in a uniform manner to 400°, 500°, 600°, according to the requirements of the work.

The furnaces for the tempering of the tubes and jackets for large guns are built as pits with fireboxes at different levels, the waste gases being taken off at the highest part of the pit. The cannon are rotated on their axis during this operation. Reheating these gun parts is done in the same manner in Russia as in the rest of Europe, except that in Russia wood fuel is used, as it is considered better than coal for this purpose. Recently it has been found that these furnaces could be heated with gas, combustion being effected by a large number of burners arranged spirally around the chamber.

Two errors are committed in the design of these furnaces:

1. The high-temperature flaming gases of the burner are produced in the heating chamber;

2. The waste gases are drawn off at the top of the furnace.

With this system the heating of pieces as long as 18 m is a matter of considerable difficulty, and it is evident that in the furnaces for the tempering of such gun tubes, it is necessary to make use of the downdraft principle. Just what must be done to obtain this result is shown more accurately by the drawing (Fig. 99) of a furnace which is in use in one of the large works for tempering heavy field artillery jackets. The flame is developed in a firebox independent of the furnace, from which the hot gases rise into the free spaces located on both sides of the heating chamber, entering the latter through a number of small orifices or ports; then, rising to the top of the chamber, they are carried off by the waste-gas flue. Taking these gases off at the top spoils the furnace. If its designer had located the waste-gas port at the level of the hearth of the heating chamber he would certainly have obtained a much more uniform operating condition.

It would be very simple to change this furnace in the manner shown in Fig. 100. Here the hot gases from the independent firebox are set free in the heating chamber of the furnace and drawn off from the bottom of the chamber. The circulation of the heated gas is indicated by the arrows.

Certainly, this construction is not suitable for heavy artillery

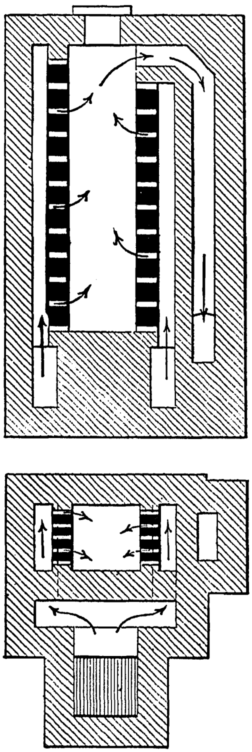


FIG. 99.

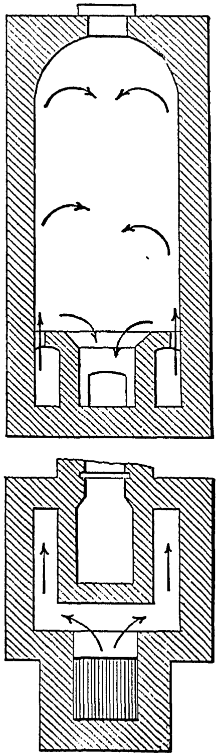


FIG. 100.

tubes and jackets. A pit in the neighborhood of 20 m in depth, as required for the large naval guns, cannot be heated in such a simple manner; but, in general, the problem may be very easily solved if the radiation losses from the heating chamber are avoided.

Another type of vertical annealing furnace is that used for the annealing of iron wire before and after it has been passed through the drawing blocks. Fig. 101 shows the design of a furnace for this purpose according to the Meyl system, as constructed in a works in Russia. This furnace, in spite of its complexity and high cost, was very unsatisfactory in operation. The annealing of the material was very irregular, and ultimately it became necessary to rebuild the furnace. The drawing (Fig. 101) shows clearly the cause of these poor results; the hot gases in the heating chamber rose from the gas ports at the bottom of the chamber to the waste-gas ports at its top and for this reason the iron pot in which the coils of wire were placed was not heated uniformly. Moreover, the small amount of space in the chamber did not afford the possibility for the development of the flame, a portion of the gases being chilled below their ignition temperature by the comparatively cold pot.

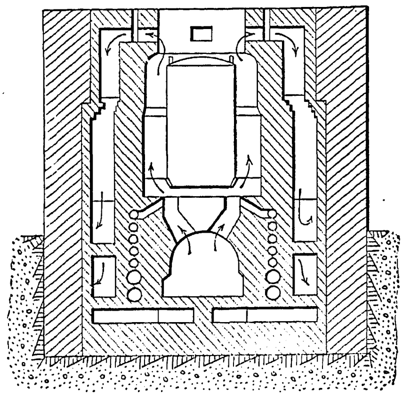


FIG. 101.

It is very evident that a furnace of this kind should be reconstructed to work upon the downdraft principle and in addition should be furnished with a combustion chamber.

XIII. HORIZONTAL TEMPERING FURNACES

Fig. 102 shows a furnace constructed according to the Krupp system for use in the heat-treating or tempering of shells at a plant in Russia. The Krupp Company have adhered to the system of heating the material in the furnace by heat radiated from the roof. It may be seen by this ridiculous construction how far from the true

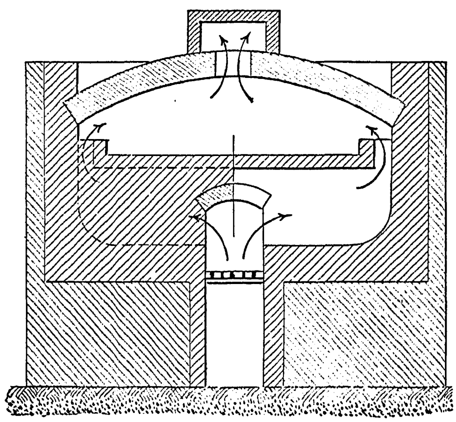


FIG. 102.

principles of heating they have been led by this theory. In the furnace here shown, the purpose of the design was that the shell should be rolled upon the hearth of the furnace, being heated at the same time by the heat reflected or "*reverberated*" from the roof; accordingly the incandescent gases from the fireboxes, in which coke was burned, were passed through the free space under the arch of the furnace, to the working opening at the highest point of the arch. But the shells were not surrounded by the incandescent gases; instead they were placed in a pocket of

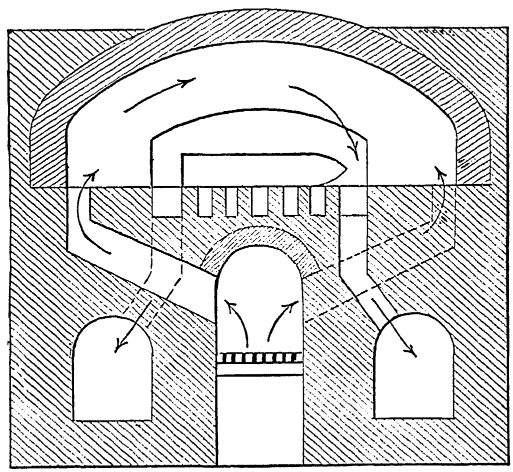


Fig. 103.

colder and stagnant gases which quickly absorbed any heat they might receive by "*reverberation.*"

The work of this furnace was very unsatisfactory. In addition, it was found desirable to substitute wood for the coke, as the latter fuel is costly in Russia. This furnace was reconstructed according to the author's design (Fig. 103). No change was made in the general arrangement of the furnace, but the direction of flow of the hot gases was reversed and the hearth was changed so that the shell rested on a number of parallel ribs of brick, a grid-iron hearth, arranged so that the colder gases were drawn off from the hearth by ports below the level of the shell into waste-gas flues. The furnace then worked perfectly.

XIV. ANNEALING AND HEATING FURNACES FOR BOILER PLATES

Fig. 104 shows an annealing or heating furnace for boiler plates. The downdraft principle is widely used in these furnaces. They work in a very satisfactory manner, but have the great disad-

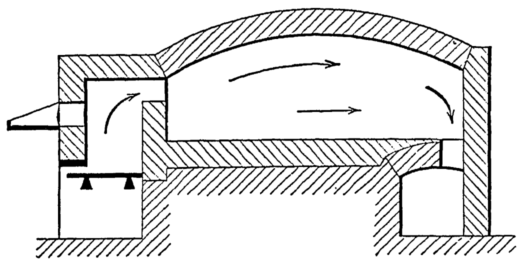


FIG. 104.

vantage that only one plate can be heated at a time, if it covers the entire hearth of the furnace. Accordingly the output of these furnaces is very small and the consumption of fuel is excessively high. If an attempt is made to heat several plates at a time in this furnace by placing one on top of the other in a pile, the heating is irregular and not uniform, as the waste gases cannot be drained

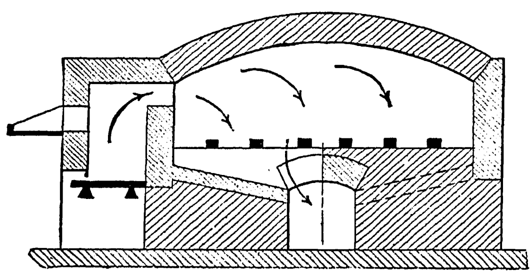


FIG. 105.

away from the heating chamber as they are cooled in giving up their heat to the charge. In order to secure a rapid and uniform heating of several plates at a time, provision must be made for the cooled gases to flow out of the heating chamber as rapidly as they give up their heat. Fig. 105 shows a plate heating furnace constructed for the purpose of heating several plates at a time. These plates are piled upon iron skids. The waste-gas outlet is located at the center of the hearth and the colder gases are drained to it by sloping channels passing in this way *underneath* the plates being heated. As the lower surface of the plates is exposed to the hot gases, their heating is rapid and uniform.

Reiner, an engineer, conceived the plan of annealing or heating plates by placing them in the furnace on their edges, and constructed the furnace shown in Fig. 106. This furnace did not give satisfactory results because the hot gases circulated upon

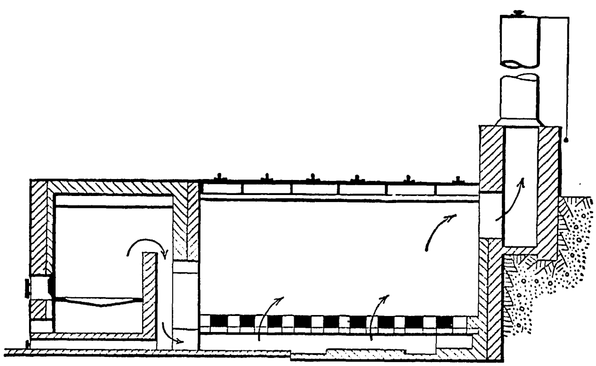


FIG. 106.

the updraft principle, and for this reason the annealing of the plates was not uniform. The idea was therefore abandoned.

The idea of Reiner may be conveniently utilized by reversing the direction of flow of the gases to the downdraft method and building the furnace with a solid roof,[1] placing the working doors in the side wall.

XV. ORDINARY REVERBERATORY FURNACES FOR REHEATING

(With bar grates)

In coal-fired furnaces with the ordinary bar grate a temperature of 1350° may be obtained in the firebox with a regulated air supply. An air supply of 1.50 to 1.75 times the amount theoretically required, according to the quality of the fuel used, will result in such a drop of the temperature of the hot gases that it becomes

[1] *Note by French translator.*—In the plate annealing furnace, as designed by Reiner, the plates were charged into the furnace through its roof and not by doors in the side walls.

impossible to attain the necessary working temperature in the furnace.

A furnace is a very delicate mechanism, which cannot be successfully constructed unless the two following conditions are successfully met:

1. The mixture of the air and the combustible gas must be thorough;

2. The size of the combustion chamber must be such that the jet of flame may form in the fore part of the heating chamber. In addition, as the working temperature of the furnace is very nearly equal to the instantaneous calorific intensity computed for the fuel, the time during which the hot gases remain in the heating chamber is less than one second; the supplying of flaming hot gases to the heating chamber and the withdrawal of the waste gases must be accomplished with uniformity.

From the foregoing it is evident that the art of proportioning these furnaces is, above all, a question of combustion, and that this must be completed before the gases reach the rear of the heating chamber. It will be necessary to take up and analyze this question in order that the direction to be given the current of hot gases may be fixed.

In one of the preceding sections mention was made of the conditions under which combustion may be best effected. In postponing the examination of these conditions the author remarked that, for the better mixture of the combustible gases with the comburent, resource was had to a strangulation of the opening over the bridge wall, just as a kerosene lamp is prevented from smoking by the contraction of the chimney.

The only result of this contraction in the opening over the bridge wall is that of increasing the velocity of the hot gases passing that point. On account of this lowering of the roofs, these gases, after passing into the heating chamber, are slowed down and fill the chamber over the hearth, from which they are drawn off through the waste-gas or chimney port. This shows why it is necessary to place the waste-gas or chimney port at the level of the hearth of the furnace, and exposes the error in the construction of those furnaces which have an updraft and from which the waste gases pass away at the highest point.

Practice completely confirms these deductions; in all reheating furnaces the waste gases should be carried away from the heating chamber at the level of the hearth; notwithstanding this fact, in metallurgical plants for smelting copper, lead and some other metals, a number of old furnaces are still in use which work on the updraft principle.

Fig. 107 shows a reverberatory copper-refining furnace, the waste gases from which are removed at the top. In this case the metal in the bath is in contact with the coolest gases in the heating chamber, and this very certainly lowers the working

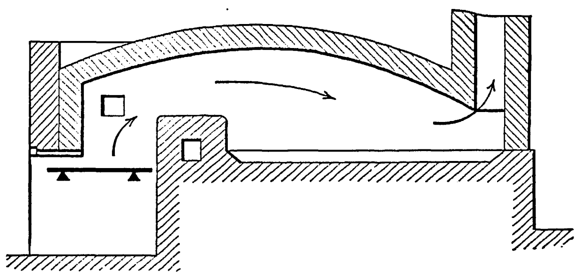


FIG. 107.

temperature of the furnace,(1) or results in the expenditure of a larger amount of fuel than would otherwise be necessary.

XVI. SIEMENS TYPE REHEATING FURNACES

Furnaces of the Siemens type are symmetrical, and the flues and ports through which the gases pass to the heating chamber serve also for the removal of the products of combustion. When reheating furnaces are constructed according to this system, the head construction of the open-hearth melting furnace is frequently copied, a procedure for which there is no necessity, owing to the difference in the purpose for which the heating furnace is used.

This difference affects the operation of the furnace in two important respects:

1. The temperature to which ingots must be reheated for rolling or forging does not exceed 1300°, and, for this reason it is not desirable to direct the jet of flame from the ports directly upon them. For the open-hearth melting furnace, on the con-

(1) In regard to the method of computation for these furnaces, refer to p. 31

trary, it is not only desirable but necessary that the jet of burning gases should be directed against the surface of the bath.

2. The surface of the bath in the open-hearth melting furnace is heated to the boiling point and at the same time rises 500 to 600 mm. The surface of the hearth of the reheating furnace is solid, except that the melted cinder lies there, slightly wetting the hearth. For this reason, in the open-hearth melting furnace it is necessary to elevate the sills of the ports so that they are at a higher level than the surface of the bath during the boil. But there is no reason for elevating the sill of the ports for a reheating furnace above the hearth level.

In actual practice, if the port sills of a reheating furnace are raised above the hearth to a height equal to the thickness of the ingots to be heated, as is ordinarily done, this will make it impos-

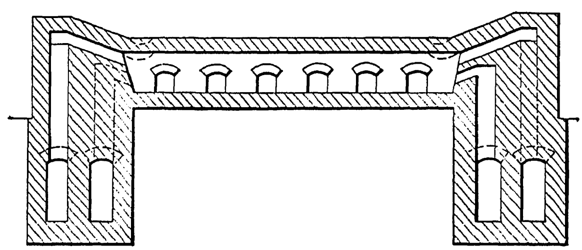


FIG. 108.

sible to get rid of the colder gases which rest on the hearth, and the ingots to be heated will be surrounded by these chilled gases.

It is evident that the rapid heating of the ingots is not possible unless these colder gases can be drained off from the hearth of the furnace. The port sills for Siemens type reheating furnaces, should, therefore, coincide with the hearth level.

Fig. 108 shows a curious design for a Siemens type reheating furnace. The hearth is 8020 mm long, with a width of 2000 mm. The roof is 1000 mm above the hearth and the heads are of the type used on melting furnaces.

Originally, the roof was straight from one end to the other; later it was dropped, just beyond the heads, as shown by the dotted lines, to a height of 680 mm from the hearth. It is evident that with a hearth length of 8020 mm, the jet of flaming gases did not touch the hearth, and did not heat the ingots which were placed near ports through which the waste gases were passing. The same phenomenon occurs in the open-hearth melting furnace when, by reason of the wear of the ports, the velocity of the air and gas is reduced, and the jet of flame has a tendency to seek the roof of the furnace, its thickness being normally that of the gaseous stream, as computed according to Yesmann's formula. At the corners of the hearth, close to the ports, a pocket of colder gases is formed, and the bath freezes. In the Goujon works, at Moscow, a case of this kind occurred with a furnace having a hearth 14 m long. In order to correct this condition two burners for naphtha were installed, one behind the other.

In the case under consideration a different procedure was followed. The roof of the furnace was dropped close to the heads, to the height required for the inverted weir, thus forcing the

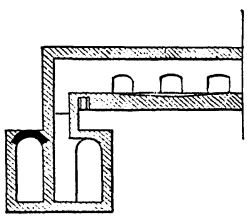


FIG. 109.

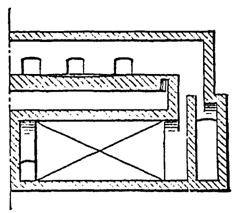


FIG. 110.

flame of burning gases to lick the hearth, not only where it issued from the ports, but also farther on. While estimating at its true value this method of reconstruction for the type of heads frequently used upon these furnaces, the author believes much better results can be attained in another way.

Figs. 109 and 110 are two sketches of constructions which may be employed in such a case. They are based upon the following considerations:

(*a*) The chilled gases are drawn off at the hearth level of the reheating furnace; this method of removing the inactive gases results in the uniform action of the hot gases upon the ingots and favors their regular and uniform heating.

(*b*) The producer gas and the preheated air enter the heating chamber of the furnace at a very low velocity, with the result that combustion takes place throughout the entire chamber, and the points of *sharp* combustion, close to the ports in the jet of flame, are eliminated. When the air and gas are heated to the temperature of the jet of flame, which may have a temperature of 1500° to 1600°, the ingots being heated are liable to be burned if exposed to the impinging action of the gases.

This assumes that the producer gas and the air are mixed outside of the heating chamber of the furnace.

(*c*) In order to prevent the melted cinder from flowing into the ports of the furnace the hearth is given a slight drainage slope from both ends to the center of the hearth. The port sills are cooled by inverted troughs immediately below the bridge wall, through

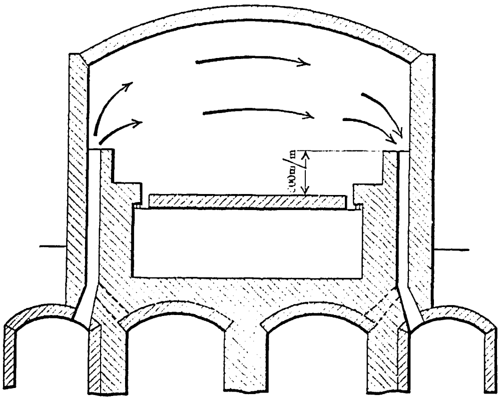


FIG. 111.

which a forced circulation of cold air is obtained by chimney suction.

A very interesting example of the complete neglect of all the fundamental principles of rational furnace construction is found in a Siemens type furnace for carbonizing and tempering armor plate, according to the Krupp[1] system. This furnace is a backward step in design and construction.

It may be seen in Fig. 111 that the vertical heads of the Siemens furnace are elevated 800 mm above the hearth of the furnace. The armor plate to be heated was placed upon a car supported above its deck by short brick columns, and then run into the furnace. It was found, nevertheless, to be within a pocket of chilled gases, which could not be drained away from the

[1] This refers to the Russian works using the Krupp system.

armor plate. Moreover, this furnace was intended to work with a *negative* pressure in the heating chamber, with the result that the reheating of the armor plate had to be accomplished while it was surrounded by a current of colder air drawn in through the door of the furnace. The effects of such a method of operation are obvious.

It had been planned to carbonize and heat two armor plates at a time, placing one above the other and separating them by a layer of charcoal. It may be stated that the lower plate was carbonized much more slowly than the upper; and it was necessary to give up the idea of carbonizing two plates at a time. Actually, the carbonizing was done one plate at a time, double the number of furnaces were required with twice the expense for installation.

Moreover, the men in charge of the Russian works were so thoroughly impressed with the prestige of the Krupp Company that they would not consent to the cutting down of these heads to the level of the hearth of the furnace.

XVII. PIT FURNACES

In pit-reheating furnaces the defect which is commonly encountered consists in the incorrect location of the waste-gas

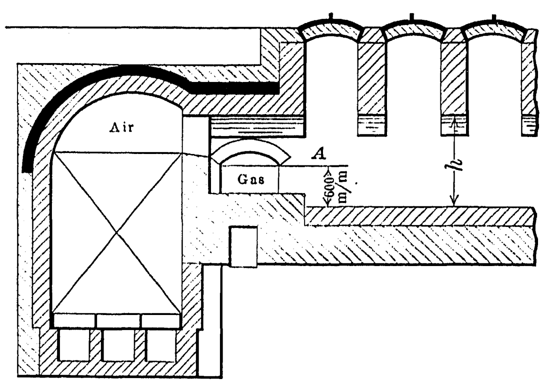


Fig. 112.

ports. In these furnaces, by reason of the considerable height of the heating chamber, it is very necessary that attention should be paid to the locating of the waste-gas port at the level of the hearth of the pits. Unless this is done, the lower parts of the ingots will be plunged into a pocket of cold and stagnant gas and therefore will not be heated uniformly, the top of the ingots being hotter than the bottom.

A number of designs for pit furnaces have been seen by the author, but none of them were perfectly satisfactory, and in many of them there were gross violations of the physical laws in regard to the carrying off of the waste gases. Fig. 112 shows a pit furnace of the "new Siemens system" of a very costly design. These pits are only supplied with regenerators for the air supply, regenerators for the gas being omitted. Therefore, the products of combustion cannot pass out of the heating chamber without passing through the air regenerators, which have a port sill or bridge wall extending to a height of 600 mm above the hearth of the pit (refer to the dotted line A). It is very clear that the hearth of these pits forms a pocket which will be filled with colder

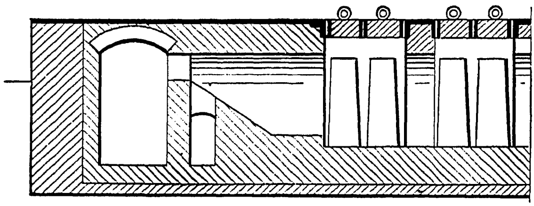


FIG. 113.

gases. Nothing can be done to save this furnace and make it heat properly. The walls were lowered between the pits until the height h was equal to the thickness of the stream under the inverted weir, as calculated by Yesmann's formula. After the furnace was reconstructed in this manner the pits worked, but the velocity of the gas in the heating chamber was equal to the velocity under a normal inverted weir, that is to say, one under which the height h equals 1 m, the velocity v being 6 m per second (refer to page 52). A velocity as high as this is unsatisfactory. The hot gases did not remain in the heating chamber a sufficient length of time to give up their heat to the ingots. For this reason the fuel consumption of these pits was very large. The best means of reconstructing this furnace consists in removing the bridge wall in front of the regenerator for air, so that the bottom of this port will be as nearly as possible at the level of the hearth of the pits.

Fig. 113 shows Siemens type pit furnace at the Salda (Oural) works, constructed according to the designs of Dalen. The gases are carried away near the top of these pits. For this reason, only a small portion of the top of the ingots is exposed to the hot gases. In the pits the lower portion of the ingots was heated much less than their top. This made it necessary to obstruct the ends of the furnace by charging ingots against the walls (thick end down). In order to improve the work of the furnace it was necessary to cut out the sloping portion of the bottom of the port, lowering it to the level of the hearth of the pit.

As the Siemens fired type of soaking pit is the most desirable furnace for the heating of ingots and is widely used, the principles governing its rational construction are given:

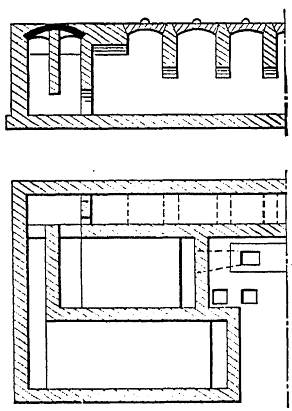


FIG. 114.

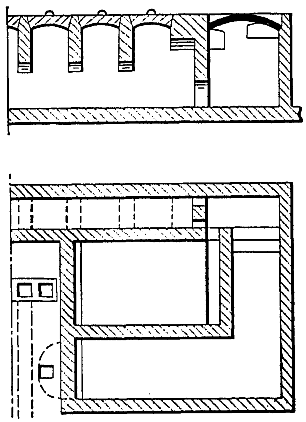


FIG. 115.

1. It is desirable that the heating chamber of these pits should be operated with a slightly negative pressure, in order that the operatives will not be burned when the covers are taken off;

2. The arches over the regenerators should be at the same level as the covers, so that the entire furnace can be covered with cast-iron plates forming a working platform;

3. It is more convenient to make the regenerators horizontal and construct them in such a manner that the gases enter at their top and pass out at their lowest portion to enter the flues leading to the reversing valves;

4. The port sills should be placed as close as possible to the level of the hearth of the pits. If this construction is employed

it will not be necessary to continue the walls between the pits close to hearth, and the hot gases will have a low velocity;

5. The dimensions of the ports should be computed by the formula for the inverted weir. This will make it possible to obtain a slow velocity of flow for the hot gases and the air in these ports, and a slow gas current through the heating chamber of the pits;

6. In order to effect the mixture of the gas and the air, the heads shown in Figs. 114 and 115 may be used. The differences between these two constructions are clearly shown by the figures.

XVIII. CONTINUOUS HEATING FURNACES

In a previous chapter (page 49) it was stated that the only continuous heating furnace which was correctly constructed was the Morgan design. All the other designs for this type of furnace contain many defects, which will be analyzed in the present chapter.

The hearth of the furnace may be flat. Of late, however, the name of continuous furnace has been limited to that type of furnace in which the material to be heated is pushed or carried through the furnace upon water-cooled skids supported above the masonry hearth of the furnace, or in lower temperature work by conveyor chains. This method is logical. Like all other mechanisms in which work is placed and from which it is withdrawn, a furnace should be:

Correctly fed by the hot gases coming from the firebox. At the same time the products of combustion must be carried away.

The stagnation of partially cooled gases within the furnace will result in considerable damage; the hot gases from the firebox cannot get into the heating chamber and this reduces the amount of contact between the hot gases and the material being heated, and in addition reduces the temperature differential between the gases and the material.

When the ingots or billets are carried upon tubes there is *below them* a canal or flue of a sufficient height to carry the chilled gas dropping below the material being heated, and to carry this cool gas to the chimney port *in its flow along the hearth.* As it passes off from the heating chamber its place is taken by hotter gases. This arrangement results in a uniform circulation of the gases. The flame of the reacting gases is in the highest and hottest portion of the chamber where complete combustion can take place before the gases cool below their ignition temperature.

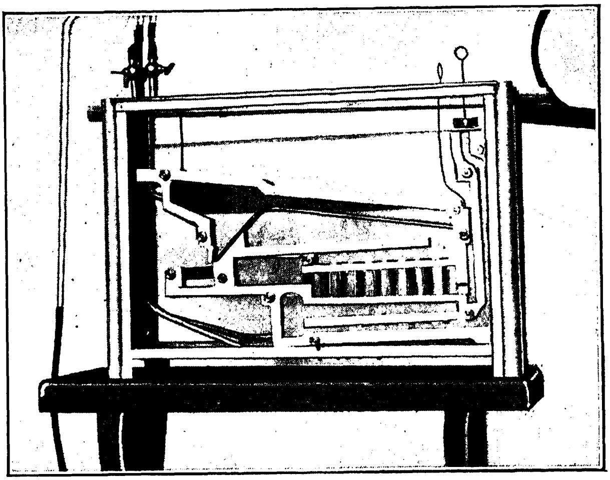


Fig. 116.

The hot gases drop down toward the ingots or billets and lose their heat by coming in contact with them, and the colder gases then descend further to the lower level from which they flow to the chimney port.

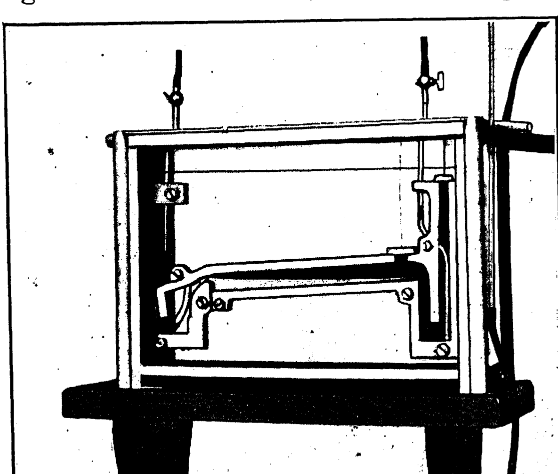


Fig. 117.

Compare this type of construction with the working system of many continuous heating furnaces of widely known and used designs. Fig. 116 is a photographic demonstration of the circulation of the hot gases (colored kerosene) in a Morgan continuous heating furnace. Fig. 117 is a similar demonstration of the pocket of chilled gas (water) which forms in those continuous heating furnaces having an ascending roof and hearth. It is readily seen that in this last case the forcing out of the cold gases by the hot gases flowing into the heating chamber is obstructed to the highest degree.

The performance of these furnaces, when constructed with a horizontal hearth, is slightly better. This type of hearth does not form a pocket for the cold gases, but the flow of the colder gases to the waste-gas port is very nearly cut off by the ingots which are being heated.

In computing the width of the hearth of the furnace it is necessary to give thorough consideration to the fact that the cold gases must be removed from the heating chamber. In practice the hearth is sometimes given a width equal to twice the length of the ingot to

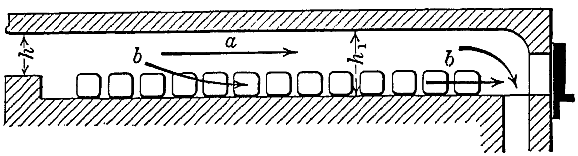


FIG. 118.

be heated, so that at each end of the ingots a wide channel will be formed for the cold gases.

What reason can be assigned for making the hearth so wide? Would it not be better to proportion the width of the furnace more closely to the length of the ingots or billets to be heated? The author has been unable to find any other reply to the foregoing questions than the following: In order that the furnace may operate well and heat uniformly, it is absolutely necessary that the burned gases be rapidly and completely carried away from the heating chamber. The side channels, at the two ends of the file or tier of ingots which are being reheated, form the only path by which these gases may pass from the furnace. If a low temperature at the charging end of the furnace is desired, it is necessary that these channels should have sufficient width.

In the case shown (Fig. 118), the continuous furnace has the roof and hearth horizontal, and the height of the gas port above the bridge wall is h; the height of the working doors is less than this dimension; the waste gases are carried away from the heating chamber by a port in the hearth of the furnace. The velocity of the current of gases in the chamber is retarded because the chamber is formed as an inverted weir having a reservoir (the height of the roof above the hearth being $h_1 > h$); the ingots are carried upon the hearth. Two currents of gases are possible: *aa* immediately below the roof and *bb* over the hearth. In order to reach the waste-gas outlet the hot gases *aa* must descend a distance h_1 and expend in doing this a certain hydrostatic pressure, which will be designated as δ mm of water. The current *bb* has to overcome the friction of the ingots. It is evident that the hot gases cannot escape immediately through the waste-gas opening, except in a case where the resistance to the passage of the current of cooler gases *bb*, equal to δ_1 mm of water, is greater than δ, that is to say, in a case where δ_1 is greater than δ.

The greater the width of the furnace in proportion to the

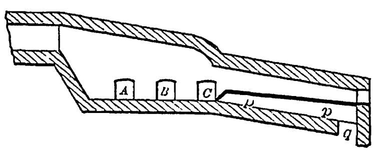


FIG. 119.

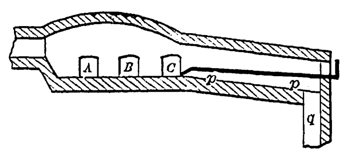


FIG. 120.

length of the ingots, the less will be the value of δ_1; and therefore there will be no reason to fear that the hot gases will pass immediately to the waste-gas flue.

When the ingots are placed upon pipe skids (Figs. 119 and 120) a channel *pp* will be formed below them through which the cooler waste gases will flow to the waste-gas port *q*. The heating chamber having a descending roof, a free space *mm*, a well-defined combustion chamber, will be formed under this roof, providing a rationally constructed heating chamber for continuous ingot reheating furnaces.(1)

Figs. 119 and 120 show two methods by which the flow of the reacting gases may be slowed down sufficiently to permit combustion to be completed in the front portion of the heating chamber. In Fig. 119 this is done by a sharp drop in the roof, dividing the heating chamber into two sections, one much higher than the

(1) This arrangement, moreover, facilitates the operation of the furnace by making it easier to roll the ingots over while they are heating.

other. This drop in the roof acts as a dam to arrest the flow of the gases for a sufficient length of time for the completion of combustion. Fig. 120 shows the Morgan type of roof. The drop in the roof simply checks the flow of the hot reacting gases; the port through which the gases enter the heating chamber is located some distance above the hearth. The author believes that these

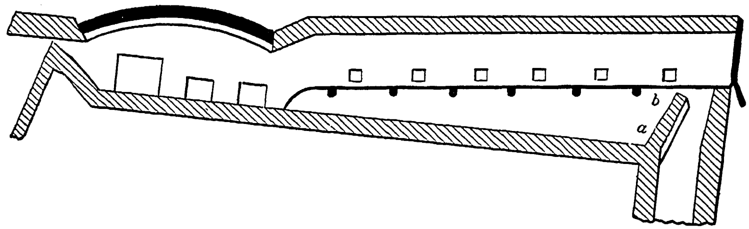


FIG. 121.

two methods of retarding the flow of the combining gases and promoting combustion are equally good.

However, it is here necessary to call attention to an error often made in the design of these furnaces. The working doors, *A*, *B*, and *C* must not be located at different heights. If these doors are placed at different heights, *the highest door acts as a chimney* in conjunction with the lowest door. Cold air will be drawn into the heating chamber at the lowest opening while a jet of flame

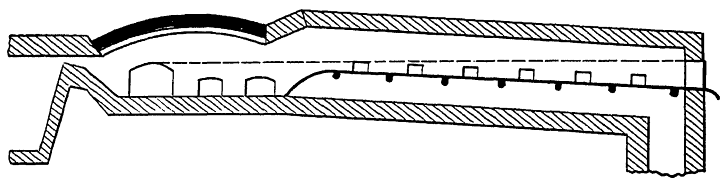


FIG. 122.

and smoke will escape from the highest opening. The sills for all of the working openings should be placed at the same level.

The last three illustrations in this chapter show the reconstruction, according to the direction of the author, of two very costly furnaces of a Swedish type. These furnaces were installed in a plant located in the Oural district.

Fig. 121 shows the longitudinal section of the furnace as originally built. During its operation it became very apparent that the wall *ab* resulted in the formation of a pocket of stagnant gases. As a remedy for this condition, the author suggested the removal of the wall and the reconstruction of the furnace according to the sketch (Fig. 122).

NOTE.—Furnaces with descending skids give a great deal of trouble with billets sagging due to improper heating conditions.

Another small furnace, of a similar design, is shown in Fig. 123, in which the reconstruction recommended is shown in dotted lines. These changes were made, and the output of the furnace was thereby increased from 11.2 tonnes to 14.7 tonnes a day, that is to say, an increase in output of 30 per cent, with the same coal consumption.

As to the profile which should be given to the roof of a con-

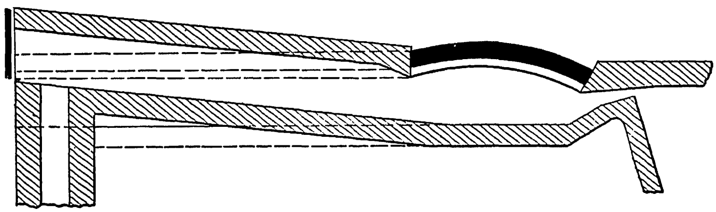


FIG. 123.

tinuous heating furnace, the following suggestions may be of service:

If the mixture of the air and the combustible gases is not sufficiently intimate in the firebox, it will be necessary to use some method of restricting and retarding the flow of the reacting gases within the port. The device frequently used is a strangulation or restriction of the port or passage through which the reacting gases enter the heating chamber. A serious disadvantage of this method is that it gives the jet of flame an exaggerated velocity. In order to absorb and reduce this velocity, two special forms of roof construction are used, which are shown in Figs. 122 and 123. If the mixing in the firebox is well accomplished, or if the gases flow into a large free space under the roof of the heating chamber at a low velocity, these spaces will act as a combustion chamber, and there will be no necessity for the strangling of the flow through the port.

If the rear end of the roof ascends, recourse is frequently had to strangulations or dropping of ridges in the roof of the furnace. The free height below such a strangulation should be

$$h_t = A\sqrt[3]{\frac{Q_t^2}{B^2 \cdot t}}.$$

For an ascending roof, when a currrent of cold gases spreads over the hearth, all of these strangulations are useless.

As a general conclusion, it must be remembered that a descending roof results in a concentration of the hot gases in the front part of the heating chamber. An ascending roof causes the hot gases to flow to the rear of the furnace, and for this reason continuous reheating furnaces are built with descending roofs. Furnaces for the uniform heating of long pieces of material [1] should be given ascending roofs.

XIX. TUNNEL FURNACES OR KILNS

Notwithstanding the fact that tunnel furnaces or kilns present attractive possibilities, they are used comparatively little, because,

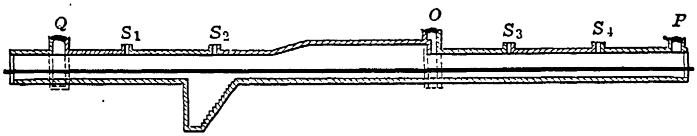


FIG. 124.

owing to the irrational construction of those already in use, they rarely work in a satisfactory manner.

Figure 124 is a longitudinal section of the Groendal furnace for curing briquets. The central portion of this kiln has a high roof which forms a combustion chamber within which the combustion of the producer gas occurs. This gas is brought to the chamber by the gas flue O across the roof of the kiln. The gas producer is operated with a blower. Air is supplied by a fan blower and flows to the kiln through the flue P, entering the kiln through ports in the roof, in the same manner as the gas. The hot gases in this kiln flow directly under the roof. In order

[1] For example, tube- or pipe-welding furnaces.

to force these gases down to the hearth of the kiln the dampers s_1 and s_2 are opened more or less.[1]

By forcing the hot gases to circulate through piles of briquets, the frictional resistance to the flow of the gases is increased; it becomes necessary, accordingly, to increase the pressure of the blower, forcing the primary air in below the grate of the gas producer and the gas into the furnace, and to increase the height of the chimney. At the same time, the clay-filled joints, which are not air-tight, tend to prevent any such increase in the pressure from the blower.

In order to overcome this difficulty, Groendal forces a stream of air under the cars in the kiln for the purpose of cooling them. It is probable that supplementary air from this source enters the chamber of the kiln and has a tendency to turn the hot gases back to the roof. As a consequence of this, these kilns must be made very long (from 50 to 70 m).

In his design of a tunnel kiln, Groendal commits the obvious error of carrying away the waste gases by ports in the roof of the tunnel. By reason of this location of the waste-gas outlet, the train of cars carrying the briquets is plunged into a pocket of cold gases. The second fault which he commits consists of seeking to force the hot gases to pass through the pile of briquets for the entire length of the tunnel. This is very difficult and in order to accomplish it the inventor limits the charge of brick upon the cars to two tiers.

The author believes that, in the design of the tunnel kiln, it is necessary to place one feature of the problem above all the others. It is not possible to force the stream of hot gases to pass through the piles of briquets, which are 50 m long. It is more advantageous to heat the briquets by a descending current of hot gases.

The difference between these two methods—that of Groendal and that proposed by the author—may be studied by comparing the cross-sections of the tunnel kiln designed by Groendal (Fig. 125) and the cross-section as corrected by the author (Fig. 126). Groendal placed a flat roof at a distance of 150 mm above the

[1] It is claimed that the manipulation of the dampers s_3 and s_4 improves the preheating of the air by forcing it down on the hearth of the kiln. These dampers are evidently useless for this purpose, as the cold air will naturally seek the lowest portion of the chamber.

briquets, which are loaded on the cars in two tiers; in squeezing the hot gases against the briquets, and forcing them to traverse

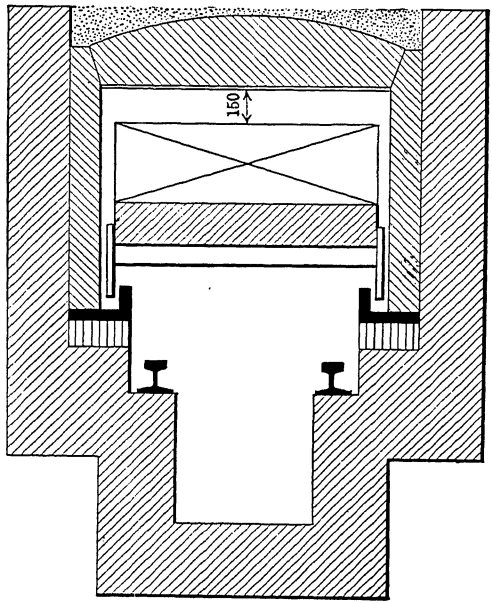


Fig. 125.

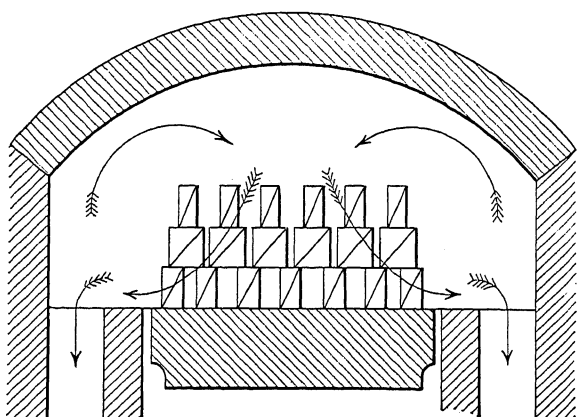


Fig. 126.

their checkerwork, he forces a part of the hot gases through the orifices in the roof of the kiln.

The author thinks that there should be charged upon the cars as many tiers of briquets as it is convenient to place, according to the conditions of manufacture. He has shown three tiers in Fig. 126. The tunnel is likewise made considerably larger than the cars. The roof is raised so that there will be a space above the top tier of briquets, sufficient for the current of hot gases to follow the roof. By reason of the increase in width of the tunnel, forming a free space on each side between the wall and the pile

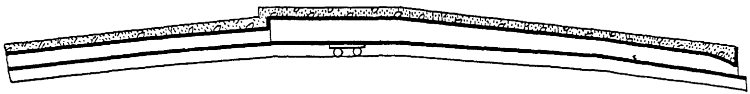


Fig. 127.

of briquets, two currents of gases will be formed, which will follow the hearth of the tunnel and flow into the waste-gas ports. The descending movement of the gases which causes them to pass through the tiers of briquets is indicated by the arrows.

Moreover, in order to obtain, through the length of the kiln, the descending circulation of the hot gases giving up their heat to the briquets and the ascending circulation of the air being heated, the author, instead of using the straight tunnel of Groendal, would form the tunnel on the arc of a vertical circle, raising the central

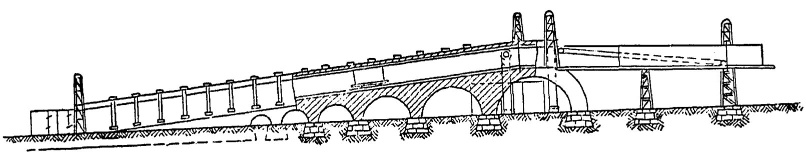


Fig. 128.

portion so that it will be higher than the ends, as is shown in Fig. 127.

A tunnel kiln constructed upon this principle would not require blowers, because the raising of the central portion (the combustion chamber) above the end at which the cold air enters the tunnel, would produce a hydrostatic pressure sufficient to overcome the resistance to the air current offered by the tunnel through which the cars pass, carrying the cooling briquets. On the other hand, the hot gases which are produced in the combustion chamber descend toward the low end of the tunnel, imparting their heat to the incoming briquets, and acted upon by the draft from the chimney, since the resistance offered by the tunnel, which has been increased in size, is not very large.

The author believes that a tunnel kiln corrected as he has indicated will work in a satisfactory manner.(1)

(1) While the author was writing these lines he did not know that, in Sweden, the principle of the tunnel kiln built on the arc of a vertical circle, had been applied with considerable success in the Aminoff furnace (Fig. 128) for the continuous carbonization of wood (*Revue de la Société russe de Métallurgie*, No. 1, pp. 48–64, 1912; extract, *Revue de Métallurgie, décembre, 1913, pp. 678–9*). These results completely confirm the author's view.

CONCLUSION

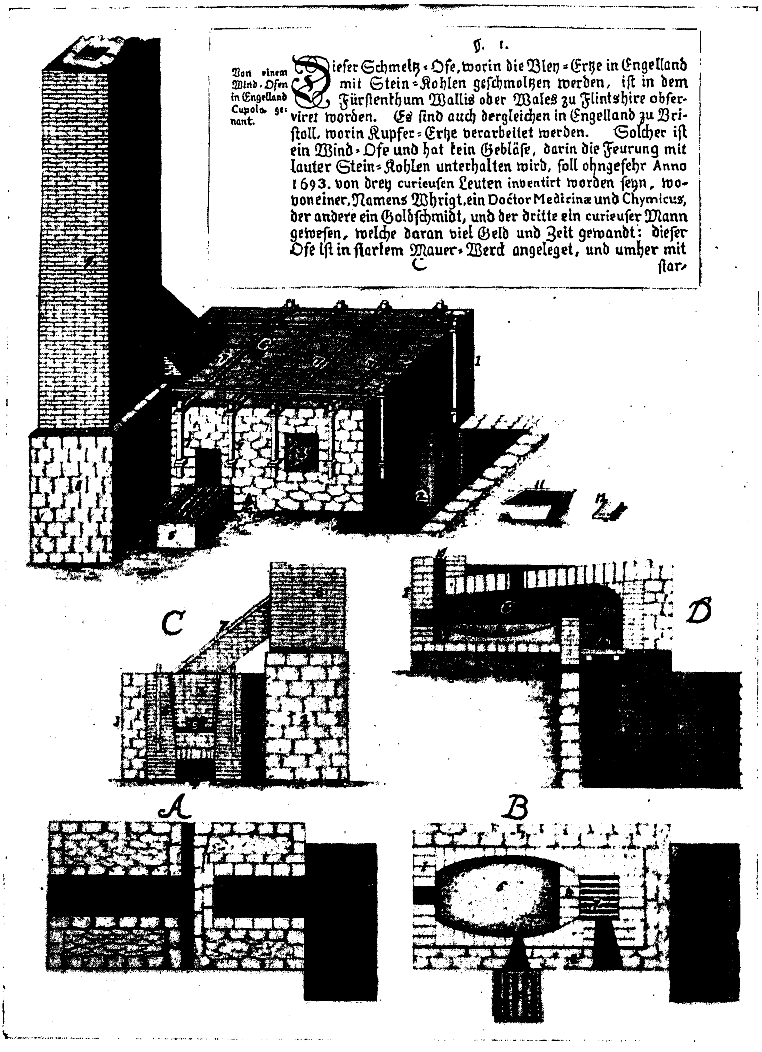
It is clearly shown, by the large number of examples which have been given, that only those furnaces in which the circulation of the gases corresponds to the natural laws will work in a satisfactory manner; that is, the hot gases which are giving off heat and cooling should flow downward; the cold gas which is being heated should flow upward.

This simple truth was demonstrated at a very early date. The first form of reverberatory furnace, working with natural draft and furnished with a chimney, the so-called cupola, was invented in 1698. In Fig. 129 are reproduced some designs and a portion of the description taken from the *Manual of Metallurgy* of Schlüter, edition of 1738. This furnace, from which the waste gases pass off through the roof of the chamber, is the *ancestor* of the updraft furnace.

In the same work there is also described another German furnace, the inventor of which is unknown, but which was probably built about 1730, for the purification of copper from lead (Fig. 130). The disks of copper-lead are set on edge on a hearth shaped with a gutter or flue which slopes downward and drains into the waste-gas flue.

The flame or hot gas comes from a fire upon a grate of bricks. In filling the heating chamber, these gases surround the disks of metal, melting the lead, and descend between the disks and flow toward the port and the chimney. Into the same flue the melted lead flows, and accumulates there, until it is drawn off, by means of the sloping of the bottom of the flue, into a small pit in front of the furnace (a fore hearth).

This shows that downdraft furnaces were invented a great many years ago. Many thousands of furnaces have been constructed since that time; nevertheless it is only in this late day that any clear conception has been arrived at concerning the mechanics of the circulation of the hot gases within the furnace.



§. 1.

Von einem Wind-Ofen in Engelland Cupola genant.

Dieser Schmeltz-Ofe, worin die Bley-Ertze in Engelland mit Stein-Kohlen geschmoltzen werden, ist in dem Fürstenthum Wallis oder Wales zu Flintshire observiret worden. Es sind auch dergleichen in Engelland zu Bristoll, worin Kupfer-Ertze verarbeitet werden. Solcher ist ein Wind-Ofe und hat kein Gebläse, darin die Feurung mit lauter Stein-Kohlen unterhalten wird, soll ohngefehr Anno 1693. von drey curieusen Leuten inventirt worden seyn, wovon einer, Namens Whrigt, ein Doctor Medicinæ und Chymicus, der andere ein Goldschmidt, und der dritte ein curieuser Mann gewesen, welche daran viel Geld und Zeit gewandt: dieser Ofe ist in starkem Mauer-Werck angeleget, und umher mit star-

C

FIG. 129.

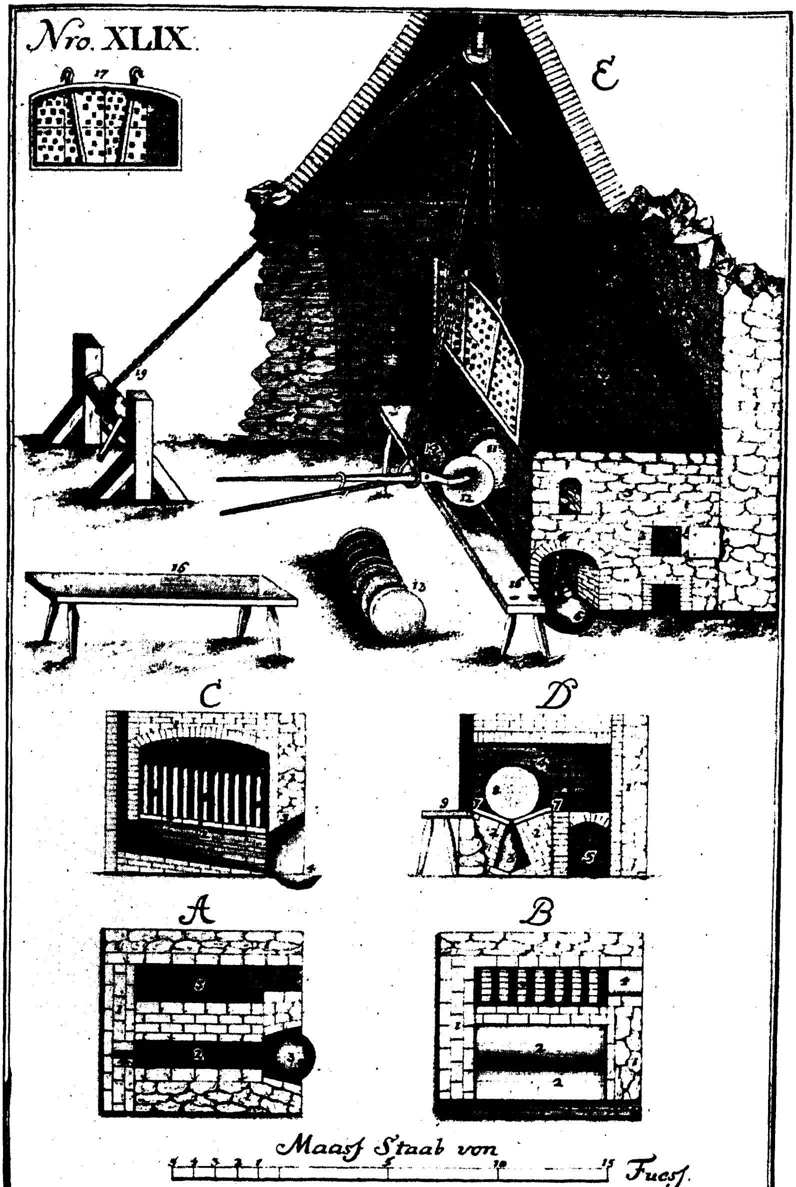


FIG. 130.

APPENDIX I

FORMULA FOR THE INVERTED WEIR ACCORDING TO THE COMPUTATION OF PROF. J.-C. YESMANN [1]

A CURRENT of hot gas of indefinite width is moving downward at a very low velocity v (Fig. 25). After it has reached the level of a horizontal sill DD having a length B (this dimension being taken at right angles to the section shown) which is fixed, the current commences to flow in a stream of a certain depth below the sill, which confines it upon its upper surface. The dimensions and arrangement of the orifice through which the gas flows are assumed to permit it to flow freely without any increase in pressure. The flow of the gas under these conditions is analogous to the flow of a stream of water over the crest of a weir or dam.

As the gas reaches the level of the sill it changes its direction of flow; the very slight vertical velocity becomes horizontal and is increased by reason of differences in pressure and level. The layer of gas, as it commences to flow horizontally, will be very thin. If the width of the inverted weir or sill from D to D is sufficient, as the flow is established, it will in time traverse some given section as a current composed of a number of parallel stream lines.

Take two cross-sections of the flowing stream, ab, where the gas has not yet acquired the horizontal velocity component, and cd, where it may be considered that the motion takes place in parallel streams with a velocity V. The current at this point being established, the small vertical velocity v, which is very slight as compared with the horizontal velocity V, may be neglected. It may therefore be concluded that the horizontal flow is effected by reason of a difference in head or level, this last being composed of the sum of the piezo and isometric heads.

[1] Extract from an article appearing in 1910 in the *Annales de l'Institut Polytechnique de Petrograd.*

This difference in head may be determined for the two extreme flow lines, *bd*, the flow line following the horizontal surface of the crest of the weir, and *ac*, which flows as the free surface of the stream.

Assume that p_i designates the hydrostatic pressure of the gas, which is at rest at the point *a*, and p_a is the hydrostatic pressure of the gas which is in motion at the same point.[1]

These two pressures are evidently equal:

$$p_a = p_i. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (a)$$

The pressure at the point *b* which is located at a distance H above the point *a* will be

$$p_b = p_i - \Delta_m H.^{(2)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (b)$$

Δ_m being the specific weight of the gas which is in motion.

[1] It should be carefully noted that there are two gaseous mediums present at this point, the one in a state of motion, of which the flow is being studied and the other assumed to be at rest, constituting an atmosphere within which the flowing gas moves.

[2] Knowing the pressure p_0 at the point *b* of a gas, the pressure at any other point *a* at any vertical distance designated as *h* (Fig. 131) may be deduced very easily. The differential equation for the hydrostatic head in the case of all heavy liquids is as follows:

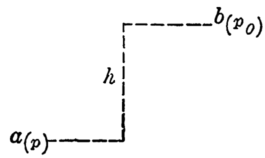


FIG. 131.

$$dp = \rho g dz,$$

ρ designating the density of the liquid.

Now, according to Mariotte's law, the relation for gas will be

$$\frac{p}{\rho} = \text{constant} = k = \frac{p_0}{\rho_0},$$

from which $\dfrac{dp}{\rho} = \dfrac{g}{k} dz$, and by integrating, h being the difference in level and Δ_0 the specific weight of the gas at the point *b*,

$$L\frac{p}{p_0} = \frac{g}{k}(z - z_0) = \frac{gh}{k} = \frac{\rho_0 g}{p_0} h = \frac{\Delta_0}{p_0} h.$$

Passing to the exponential function and taking only the first two terms of the developed series, which give an approximation sufficiently exact for the purpose, by reason of the small value of h with regard to the piezometric height $\dfrac{p_0}{\Delta_0}$, the following expression is obtained,

$$p = p_0 + \Delta_0 h.$$

In the same manner it will be found that

$$p_c = p_t - \Delta_t(H-h), \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (c)$$

$$p_d = p_t - \Delta_t(H-h) - \Delta_m h, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (d)$$

Δ_t being the specific weight of the gas which is at rest and h being the thickness of the layer of gas flowing through the section cd.

The difference in head or pressure which acts to cause the flow of the gaseous vein bd is

$$\frac{p_b}{\Delta_m} - \frac{p_d}{\Delta_m} = \frac{1}{\Delta} \quad [\Delta_t(H-h) + \Delta_m h - \Delta_m H] = (H-h)\frac{\Delta_t - \Delta_m}{\Delta_m}.$$

The difference in pressure or head for the vein ac which rises, as it flows a to c, the distance $cc_1 = H-h$, will be

$$\frac{p_a}{\Delta_m} = \frac{p_c}{\Delta_m} - (H-h) = \frac{\Delta_t}{\Delta_m}(H-h) - (H-h) = (H-h)\frac{\Delta_t - \Delta_m}{\Delta_m}.$$

This difference in pressure or head being the same for the two extreme veins or flow lines bd and cd, it is not difficult to see that it will be the same for all of the intermediate veins or lines of flow.

Therefore, the velocity of all of the elementary veins or flow lines may be determined, as they are based upon the difference in level and they will be the same; that is, the velocities of all the particles flowing through the section cd are equal.

It is possible to determine the theoretical velocity of the gas in motion by the equation

$$\frac{v2}{2g} = (H-h)\frac{\Delta_t - \Delta_m}{\Delta_m},$$

from which

$$v = \sqrt{2g(H-h)\frac{\Delta_t - \Delta_m}{\Delta_m}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

The theoretical volume of gas Q_t which will flow under the sill or weir in a layer whose thickness is h and whose width is B will be equal to

$$Q_t = Bh\sqrt{2g(H-h)\frac{\Delta_t - \Delta_m}{\Delta_m}} \quad . \quad . \quad . \quad . \quad (2)$$

If it is possible, in the case of flowing gases which is being analyzed, to adopt the hypothesis assumed by Boussinesq, for the flow of water over a weir, the head at the crest h is established in such a fashion that q_t attains its maximum value for any given value of H. It is evident that the following expression will then be obtained:

$$\frac{dQ_t}{dh}=0,$$

from which, according to Eq. (2),

$$\sqrt{H-h}-\frac{h}{2\sqrt{H-h}}=0,$$

from which

$$h=\tfrac{2}{3}H, \quad \ldots \ldots \ldots \quad (3)$$

and

$$H=\tfrac{3}{2}h. \quad \ldots \ldots \ldots \quad (4)$$

In taking into account this last relationship, Eq. (2) will take the form

$$Q_t=\frac{2}{3}\sqrt{\frac{1}{3}}BH\sqrt{2gH\frac{\Delta_t-\Delta_m}{\Delta_m}},$$

and

$$Q_t=\sqrt{\frac{1}{3}}Bh\sqrt{2gh\frac{\Delta_t-\Delta_m}{\Delta_m}}.$$

The actual quantity of gas Q flowing under the inverted weir will be less than the theoretical volume by reason of the frictional and other resistance; therefore these equations will become:

$$Q=\frac{2}{3}\sqrt{\frac{1}{3}}\mu BH\sqrt{2gH\frac{\Delta_t-\Delta_m}{\Delta_m}}, \quad \ldots \ldots \quad (5)$$

$$Q=\sqrt{\frac{1}{2}}\mu_1 Bh\sqrt{2gh\frac{\Delta_t-\Delta_m}{\Delta_m}}, \quad \ldots \ldots \quad (6)$$

in which the coefficients μ and μ_1 may not be equal, as, in passing from equations (3) and (4) to the actual flow of the gas, it will be found that the relation between H and h is different from that given in these equations.

According to the researches of Bazin, the coefficient μ for the flow of water is given by the formula

$$\frac{2}{3}\sqrt{\frac{1}{3}}\mu=\left[0.70+0.185\frac{H}{E}\right]\left[0.405+\frac{0.003}{H}\right]\left[1+0.55\frac{H^2}{p^2}\right], \quad (7)$$

in which p is the depth of the channel and E[1] must not be less than $\tfrac{2}{3}H$.

(1) E being the distance DD (Fig. 25).

In order to determine approximately the limits of variation of the coefficients which enter into the preceding formula they have been computed for two particular cases.

Assuming 0 m 30 as the depth of the layer of gas flowing under the crest of the weir, this will correspond approximately to $H=0$ m 45, for which the minimum length of the sill or width of crest E is 0 m 30 ($\frac{2}{3}H$) and the maximum width of crest will be in the neighborhood of 1 m (about $2H$); from which, applying formula (7), for the first case $\mu=1.045$ and for the second case $\mu=0.838$. It may be assumed that these values will be the same for a gas flowing under an inverted weir.

Actually, whatever the value of this friction for a gas, it will be less than the friction of water, but it can be assumed, as a first approximation, that this difference will be compensated for by the greater friction of the free surface of the gas against the medium which is at rest.

Therefore, the value of H being very nearly 0 m 45 and that of E being 1 m, the following expression is obtained:

$$Q=0.322BH\sqrt{2gH\frac{\Delta_i-\Delta_m}{\Delta_m}}. \quad . \quad . \quad . \quad . \quad . \quad (8)$$

Then, assuming that $\mu=\mu_1$, for the case in which h is 0 m 30 and E is 1 meter, the expression will be

$$Q=\sqrt{\frac{1}{2}}\cdot 0.838Bh\sqrt{2gh\frac{\Delta_i-\Delta_m}{\Delta_m}}=0.593Bh\sqrt{2gh\frac{\Delta_i-\Delta_m}{\Delta_m}}. \quad . \quad . \quad (9)$$

If the current of gas which is flowing under the weir and the gas medium through which it flows have the same chemical composition it can be admitted that

$$\frac{\Delta_i}{\Delta_m}=\frac{T_i}{T_m}=\frac{273+t_i}{273+t_m},$$

t representing the temperature.

By taking $\Delta_0=1$ kg 29, the weight of a cubic meter of air when it is at the temperature of 0°, that is to say when $T_i=273°$, it will be found by Formula (9) that

$$Q=0.593B\sqrt{\frac{2g}{273}}\sqrt{h^3t}=0.156\sqrt{h^3t},$$

from which

$$h=3.42\sqrt[3]{\frac{Q^2}{B^2t}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

If the thickness of the crest of the inverted weir exceeds 1 m, the resistance to the flow of the gas will be increased and it will be necessary to increase the value of h. On the contrary, if the thickness of the crest and the roughness of the wall surface are diminished, the coefficient in the last formula will become less, and its minimum value corresponding to $\mu=1$ will, as it is easy to see, be equal to or very close to 3.05.

The preceding formulas have been established upon the supposition that the gas arriving at the bridge wall has a horizontal velocity equal to zero. But it follows from these considerations that, in the case where the horizontal velocity is not a negligible factor (Fig. 132) the form of the formula does not change.

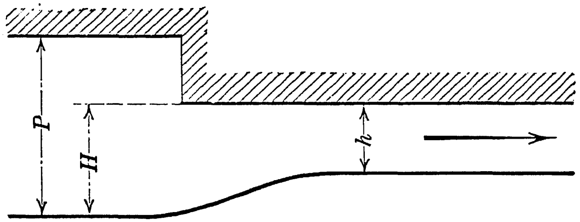


FIG. 132.

In these cases it is evidently necessary to increase the value of the coefficient H to $1+0.55\frac{H^2}{p^2}$, conforming to Bazin's formula, p being the depth or thickness of the flowing current of gas before it reaches the sill or weir, and being equal to the height of the stream over the weir plus the value of H. When this last value is known, it becomes possible to introduce into the formula a correction for each particular case, as well as in the preceding computations.

It will be seen in the numerical examples examined that the relation between the thickness or depth of the gaseous stream which flows through the opening over the bridge or its depth below the inverted weir, the volume of gas per second Q and the length of the crest of the weir at right angles to the direction of flow B, is finally given by the equation:

$$h=A\sqrt[2]{\frac{Q^2}{B^2t}}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

A being a coefficient determined experimentally and depending upon the depth of the gaseous stream, h, the thickness of the crest of the weir in the direction of flow, E, the velocity with which the gases arrive at the weir or sill, the temperature and specific weight of the gas composing the stream, and the roughness of the walls at the sides and top of the flowing stream.

APPENDIX II

VOLUME OF GASES [1]

Values of $1+0.00367t$

The quantity $1+0.00367t$ gives for a gas the volume at $t°$ when the pressure is kept constant, or the pressure at $t°$ when the volume is kept constant, in terms of the volume or the pressure at 0°.

(*a*) This part of the table gives the values of $1+0.00367t$ for values of t between 0° and 10° C. by tenths of a degree.

(*b*) This part gives the values of $1+0.00367t$ for values of t between $-90°$ and $+1990°$ C. by 10° steps.

These two parts serve to give any intermediate value to one tenth of a degree by a simple computation as follows: In the (*b*) table find the number corresponding to the nearest lower temperature, and to this number add the decimal part of the number in the (*a*) table which corresponds to the difference between the nearest temperature in the (*b*) table and the actual temperature. For example, let the temperature be 682°.2:

We have for 680 in table (*b*) the number........	3.49560
And for 2.2 in table (*a*) the decimal.............	0.00807
Hence the number for 682.2 is...............	3.50367

(*a*) Values of $1+0.00367t$ for Values of t between 0° and 10° C. by Tenths of a Degree

t	0.0	0.1	0.2	0.3	0.4
0	1.00000	1.00037	1.00073	1.00110	1.00147
1	.00367	.00404	.00440	.00477	.00514
2	.00734	.00771	.00807	.00844	.00881
3	.01101	.01138	.01174	.01211	.01248
4	.01468	.01505	.01541	.01578	.01615
5	1.01835	1.01872	1.01908	1.01945	1.01982
6	.02202	.02239	.02275	.02312	.02349
7	.02569	.02606	.02642	.02679	.02716
8	.02936	.02973	.03009	.03046	.03083
9	.03303	.03340	.03376	.03413	.03450

t	0.5	0.6	0.7	0.8	0.9
0	1.00184	1.00220	1.00257	1.00294	1.00330
1	.00550	.00587	.00624	.00661	.00697
2	.00918	.00954	.00991	.01028	.01064
3	.01284	.01321	.01358	.01395	.01431
4	.01652	.01688	.01725	.01762	.01798
5	1.02018	1.02055	1.02092	1.02129	1.02165
6	.02386	.02422	.02459	.02496	.02532
7	.02752	.02789	.02826	.02863	.02899
8	.03120	.03156	.03193	.03290	.03266
9	.03486	.03523	.03560	.03597	.03633

[1] *Smithsonian Tables.*

VOLUME OF GASES—*Continued*

(*b*) Values of $1+0.00367t$ for Values of t between $-90°$ and $+1990°$ C. by 10° Steps

t	00	10	20	30	40
−000	1.00000	0.96330	0.92660	0.88990	0.85320
+000	1.00000	1.03670	1.07340	1.11010	1.14680
100	1.36700	1.40370	1.44040	1.47710	1.51380
200	1.73400	1.77070	1.80740	1.84410	1.88080
300	2.10100	2.13770	2.17440	2.21110	2.24780
400	2.46800	2.50470	2.54140	2.57810	2.61480
500	2.83500	2.87170	2.90840	2.94510	2.98180
600	3.20200	3.23870	3.27540	3.31210	3.34880
700	3.56900	3.60570	3.64240	3.67910	3.71580
800	3.93600	3.97270	4.00940	4.04610	4.08280
900	4.30300	4.33970	4.37640	4.41310	4.44980
1000	4.67000	4.70670	4.74340	4.78010	4.81680
1100	5.03700	5.07370	5.11040	5.14710	5.18380
1200	5.40400	5.44070	5.47740	5.51410	5.55080
1300	5.77100	5.80770	5.84440	5.88110	5.91780
1400	6.13800	6.17470	6.21140	6.24810	6.28480
1500	6.50500	6.54170	6.57840	6.61510	6.65180
1600	6.87200	6.90870	6.94540	6.98210	7.01880
1700	7.23900	7.27570	7.31240	7.34910	7.38580
1800	7.60600	7.64270	7.67940	7.71610	7.75280
1900	7.97300	8.00970	8.04640	8.08310	8.11980
2000	8.34000	8.37670	8.41340	8.45010	8.48680

t	50	60	70	80	90
−000	0.81650	0.77980	0.74310	0.70640	0.66970
+000	1.18350	1.22020	1.25690	1.29360	1.33030
100	1.55050	1.58720	1.62390	1.66060	1.69730
200	1.91750	1.95420	1.99090	2.02760	2.06430
300	2.28450	2.32120	2.35790	2.39460	2.43130
400	2.65150	2.68820	2.72490	2.76160	2.79830
500	3.01850	3.05520	3.09190	3.12860	3.16530
600	3.38550	3.42220	3.45890	3.49560	3.53230
700	3.75250	3.78920	3.82590	3.86260	3.89930
800	4.11950	4.15620	4.19290	4.22960	4.26630
900	4.48650	4.52320	4.55990	4.59660	4.63330
1000	4.85350	4.89020	4.92690	4.96360	5.00030
1100	5.22050	5.25720	5.29390	5.33060	5.36730
1200	5.58750	5.62420	5.66090	5.69760	5.73430
1300	5.95450	5.99120	6.027[illegible]0	6.06460	6.10130
1400	6.32150	6.35820	6.39490	6.43160	6.46830
1500	6.68850	6.72520	6.76190	6.79860	6.83530
1600	7.05550	7.09220	7.12890	7.16560	7.20230
1700	7.42250	7.45920	7.49590	7.53260	7.56930
1800	7.78950	7.82620	7.86290	7.89960	7.93630
1900	8.15650	8.19320	8.22990	8.26660	8.30330
2000	8.52350	8.56020	8.59690	8.63760	8.67030

TABLE C

Weight of Gases

Values of $1 \div (1+0.00367t)$ for values of t between $-90°$ and $+2990°$ by $10°$ steps.

t	00	10	20	30	40
−000	1.0000	1.0381	1.0792	1.1237	1.1721
+000	1.0000	0.9643	0.9319	0.9009	0.8718
100	0.7315	0.7122	0.6944	0.6770	0.6605
200	0.5767	0.5646	0.5534	0.5423	0.5316
300	0.4760	0.4677	0.4600	0.4523	0.4450
400	0.4052	0.3992	0.3935	0.3879	0.3824
500	0.3527	0.3483	0.3439	0.3396	0.3353
600	0.3123	0.3087	0.3053	0.3019	0.2986
700	0.2802	0.2773	0.2746	0.2718	0.2691
800	0.2541	0.2517	0.2494	0.2472	0.2449
900	0.2324	0.2304	0.2285	0.2266	0.2247
1000	0.2141	0.2124	0.2108	0.2092	0.2076
1100	0.1985	0.1971	0.1957	0.1943	0.1929
1200	0.1850	0.1838	0.1826	0.1814	0.1801
1300	0.1733	0.1721	0.1711	0.1700	0.1689
1400	0.1629	0.1619	0.1610	0.1600	0.1591
1500	0.1537	0.1528	0.1520	0.1511	0.1503
1600	0.1455	0.1447	0.1439	0.1432	0.1425
1700	0.1381	0.1374	0.1367	0.1360	0.1354
1800	0.1315	0.1308	0.1302	0.1296	0.1290
1900	0.1254	0.1248	0.1243	0.1237	0.1231
2000	0.1199	0.1194	0.1188	0.1183	0.1178

t	50	60	70	80	90
−000	1.2247	1.2824	1.3457	1.4156	1.4932
+000	0.8453	0.8196	0.7955	0.7727	0.7518
100	0.6447	0.6301	0.6157	0.6020	0.5892
200	0.5213	0.5091	0.5023	0.4931	0.4844
300	0.4378	0.4308	0.4241	0.4175	0.4113
400	0.3771	0.3720	0.3669	0.3620	0.3574
500	0.3312	0.3273	0.3234	0.3196	0.3159
600	0.2953	0.2922	0.2891	0.2860	0.2831
700	0.2665	0.2639	0.2613	0.2589	0.2564
800	0.2427	0.2406	0.2385	0.2364	0.2344
900	0.2228	0.2210	0.2192	0.2175	0.2158
1000	0.2060	0.2045	0.2030	0.2014	0.2000
1100	0.1915	0.1902	0.1888	0.1873	0.1863
1200	0.1789	0.1778	0.1766	0.1755	0.1743
1300	0.1679	0.1669	0.1658	0.1649	0.1639
1400	0.1581	0.1572	0.1563	0.1554	0.1546
1500	0.1494	0.1486	0.1478	0.1470	0.1463
1600	0.1417	0.1410	0.1402	0.1395	0.1388
1700	0.1347	0.1340	0.1334	0.1327	0.1321
1800	0.1283	0.1277	0.1271	0.1265	0.1260
1900	0.1226	0.1220	0.1215	0.1209	0.1204
2000	0.1173	0.1168	0.1163	0.1158	0.1153

APPENDIX III

TABLE 1

TABLE OF THE VELOCITY HEADS OR PRESSURES REQUIRED TO IMPRESS VELOCITIES RANGING FROM 0 TO 30 M 90 PER SECOND

	0.0	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
0	0.00	0.00	0.00	0.00	0.01	0.01	0.02	0.03	0.03	0.04
1	0.05	0.06	0.07	0.09	0.10	0.11	0.13	0.15	0.17	0.18
2	0.20	0.22	0.25	0.27	0.29	0.32	0.34	0.37	0.40	0.43
3	0.46	0.49	0.52	0.56	0.59	0.62	0.66	0.70	0.74	0.78
4	0.82	0.86	0.90	0.94	0.99	1.03	1.08	1.13	1.17	1.22
5	1.27	1.33	1.38	1.43	1.49	1.54	1.60	1.66	1.71	1.77
6	1.83	1.90	1.96	2.02	2.09	2.15	2.22	2.29	2.36	2.43
7	2.50	2.57	2.64	2.72	2.79	2.87	2.94	3.02	3.10	3.18
8	3.26	3.34	3.43	3.51	3.60	3.68	3.77	3.86	3.95	4.04
9	4.13	4.22	4.31	4.41	4.50	4.60	4.70	4.80	4.90	5.00
10	5.10	5.20	5.30	5.41	5.51	5.62	5.73	5.84	5.94	6.06
11	6.17	6.30	6.38	6.50	6.62	6.74	6.86	6.98	7.10	7.22
12	7.34	7.46	7.59	7.71	7.84	7.96	8.09	8.22	8.35	8.48
13	8.61	8.75	8.88	9.02	9.15	9.29	9.43	9.57	9.71	9.85
14	9.99	10.15	10.28	10.43	10.57	10.72	10.87	11.02	11.17	11.32
15	11.47	11.63	11.78	11.94	12.09	12.25	12.41	12.57	12.73	12.89
16	13.05	13.22	13.38	13.55	13.71	13.88	14.05	14.22	14.39	14.56
17	14.73	14.91	15.08	15.26	15.46	15.61	15.79	15.97	16.15	16.33
18	16.51	16.70	16.88	17.07	17.25	17.44	17.63	17.82	18.02	18.21
19	18.40	18.60	18.79	18.99	19.18	19.38	19.58	19.78	19.99	20.19
20	20.39	20.60	20.80	21.01	21.21	21.42	21.63	21.84	22.06	22.27
21	22.48	22.70	22.91	23.13	23.34	23.56	23.78	24.00	24.23	24.49
22	24.67	24.88	25.10	25.31	25.53	25.74	25.98	26.23	26.47	26.72
23	26.96	27.20	27.44	27.67	27.91	28.15	28.39	28.63	28.88	29.12
24	29.36	29.61	29.85	30.10	30.34	30.59	30.84	31.10	31.35	31.61
25	31.86	32.12	32.37	32.63	32.88	33.14	33.49	33.66	33.93	34.19
26	34.45	34.72	34.99	35.25	35.52	35.79	36.06	36.34	36.61	36.89
27	37.16	37.44	37.71	37.99	38.26	38.54	38.82	39.11	39.39	39.68
28	39.96	40.25	40.54	40.82	41.11	41.40	41.69	41.98	42.28	42.57
29	42.86	43.16	43.46	43.76	44.06	44.36	44.66	44.96	45.27	45.57
30	45.87	46.18	46.49	46.79	47.10	47.41	47.72	48.04	48.35	48.67

TABLE 2

TABLE OF THEORETICAL VELOCITY $v = \sqrt{2gh}$ CORRESPONDING TO DIFFERENT VELOCITY HEADS h

Velocity Head or Fall, h Meters	Velocity, Meters per Second	Velocity Head or Fall, h Meters	Velocity, Meters per Second	Velocity Head or Fall, h Meters	Velocity, Meters per Second	Velocity Head or Fall, h Meters	Velocity, Meters per Second	Velocity Head or Fall, h Meters	Velocity, Meters per Second
0.001	0.140	0.25	2.215	0.58	3.373	0.91	4.225	1.24	4.933
0.002	0.198	0.26	2.259	0.59	3.402	0.92	4.248	1.25	4.953
0.003	0.243	0.27	2.301	0.60	3.431	0.93	4.271	1.26	4.972
0.004	0.280	0.28	2.344	0.61	3.459	0.94	4.294	1.27	4.991
0.005	0.313	0.29	2.385	0.62	3.488	0.95	4.317	1.28	5.011
0.006	0.313	0.30	2.426	0.63	3.516	0.96	4.340	1.29	5.031
0.007	0.370	0.31	2.466	0.64	3.543	0.97	4.362	1.30	5.050
0.008	0.395	0.32	2.506	0.65	3.571	0.98	4.384	1.31	5.069
0.009	0.420	0.33	2.542	0.66	3.598	0.99	4.407	1.32	5.089
0.01	0.443	0.34	2.584	0.67	3.625	1.00	4.429	1.33	5.108
0.02	0.626	0.35	2.620	0.68	3.652	1.01	4.451	1.34	5.127
0.03	0.767	0.36	2.658	0.69	3.679	1.02	4.473	1.35	5.146
0.04	0.886	0.37	2.694	0.70	3.706	1.03	4.495	1.36	5.165
0.05	0.990	0.38	2.730	0.71	3.732	1.04	4.517	1.37	5.184
0.06	1.085	0.39	2.766	0.72	3.758	1.05	4.539	1.38	5.203
0.07	1.172	0.40	2.801	0.73	3.784	1.06	4.560	1.39	5.222
0.08	1.253	0.41	2.836	0.74	3.810	1.07	4.582	1.40	5.241
0.09	1.329	0.42	2.870	0.75	3.836	1.08	4.603	1.41	5.259
0.10	1.401	0.43	2.904	0.76	3.861	1.09	4.624	1.42	5.278
0.11	1.468	0.44	2.938	0.77	3.886	1.10	4.645	1.43	5.297
0.12	1.334	0.45	2.971	0.78	3.911	1.11	4.666	1.44	5.315
0.13	1.597	0.46	3.004	0.79	3.936	1.12	4.687	1.45	5.333
0.14	1.657	0.47	3.037	0.80	3.961	1.13	4.708	1.46	5.351
0.15	1.715	0.48	3.069	0.81	3.986	1.14	4.729	1.47	5.370
0.16	1.772	0.49	3.100	0.82	4.011	1.15	4.750	1.48	5.388
0.17	1.826	0.50	3.132	0.83	4.035	1.16	4.770	1.49	5.406
0.18	1.879	0.51	3.163	0.84	4.059	1.17	4.790	1.50	5.425
0.19	1.931	0.52	3.194	0.85	4.083	1.18	4.811	1.51	5.443
0.20	1.981	0.53	3.224	0.86	4.107	1.19	4.831	1.52	5.461
0.21	2.030	0.54	3.253	0.87	4.131	1.20	4.852	1.53	5.479
0.22	2.078	0.55	3.285	0.88	4.155	1.21	4.872	1.54	5.496
0.23	2.124	0.56	3.314	0.89	4.178	1.22	4.892	1.55	5.514
0.24	2.170	0.57	3.344	0.90	4.202	1.23	4.913	1.56	5.532

TABLE 2—*Continued*

TABLE OF THEORETICAL VELOCITY $v = \sqrt{2gh}$ CORRESPONDING TO DIFFERENT VELOCITY HEADS h

Velocity Head or Fall, h Meters	Velocity, Meters per Second	Velocity Head or Fall, h Meters	Velocity, Meters per Second	Velocity Head or Fall, h Meters	Velocity, Meters per Second	Velocity Head or Fall, h Meters	Velocity, Meters per Second	Velocity Head or Fall, h Meters	Velocity, Meters per Second
1.57	5.550	1.90	6.105	2.23	6.614	2.56	7.087	2.89	7.530
1.58	5.567	1.91	6.122	2.24	6.629	2.57	7.101	2.90	7.543
1.59	5.585	1.92	6.138	2.25	6.644	2.58	7.114	2.91	7.556
1.60	5.603	1.93	6.154	2.26	6.658	2.59	7.128	2.92	7.569
1.61	5.620	1.94	6.170	2.27	6.673	2.60	7.142	2.93	7.582
1.62	5.637	1.95	6.186	2.28	6.688	2.61	7.156	2.94	7.594
1.63	5.655	1.96	6.202	2.29	6.703	2.62	7.169	2.95	7.607
1.64	5.672	1.97	6.217	2.30	6.717	2.63	7.183	2.96	7.620
1.65	5.690	1.98	6.232	2.31	6.732	2.64	7.197	2.97	7.633
1.66	5.707	1.99	6.248	2.32	6.746	2.65	7.210	2.98	7.646
1.67	5.724	2.00	6.264	2.33	6.761	2.66	7.224	2.99	7.659
1.68	5.741	2.01	6.279	2.34	6.775	2.67	7.237	3.00	7.672
1.69	5.758	2.02	6.295	2.35	6.790	2.68	7.251	3.01	7.684
1.70	5.775	2.03	6.311	2.36	6.804	2.69	7.265	3.02	7.697
1.71	5.792	2.04	6.326	2.37	6.819	2.70	7.278	3.03	7.710
1.72	5.809	2.05	6.341	2.38	6.833	2.71	7.291	3.04	7.722
1.73	5.826	2.06	6.357	2.39	6.847	2.72	7.305	3.05	7.735
1.74	5.812	2.07	6.372	2.40	6.862	2.73	7.318	3.06	7.748
1.75	5.859	2.08	6.388	2.41	6.876	2.74	7.332	3.07	7.760
1.76	5.876	2.09	6.403	2.42	6.890	2.75	7.315	3.08	7.773
1.77	5.893	2.10	6.418	2.43	6.904	2.76	7.358	3.09	7.786
1.78	5.909	2.11	6.434	2.44	6.919	2.77	7.372	3.10	7.798
1.79	5.926	2.12	6.449	2.45	6.933	2.78	7.385	3.11	7.811
1.80	5.942	2.13	6.464	2.46	6.947	2.79	7.398	3.12	7.823
1.81	5.959	2.14	6.479	2.47	6.961	2.80	7.411	3.13	7.836
1.82	5.975	2.15	6.494	2.48	6.975	2.81	7.425	3.14	7.849
1.83	5.992	2.16	6.510	2.49	6.989	2.82	7.437	3.15	7.861
1.84	6.008	2.17	6.525	2.50	7.003	2.83	7.451	3.16	7.873
1.85	6.024	2.18	6.541	2.51	7.017	2.84	7.464	3.17	7.886
1.86	6.041	2.19	6.555	2.52	7.031	2.85	7.477	3.18	7.898
1.87	6.057	2.20	6.570	2.53	7.045	2.86	7.490	3.19	7.911
1.88	6.073	2.21	6.584	2.54	7.059	2.87	7.503	3.20	7.923
1.89	6.089	2.22	6.599	2.55	7.073	2.88	7.517	3.21	7.936

TABLE 2—*Continued*

TABLE OF THEORETICAL VELOCITY $v=\sqrt{2gh}$ CORRESPONDING TO DIFFERENT VELOCITY HEADS h

Velocity Head or Fall, h Meters	Velocity, Meters per Second	Velocity Head or Fall, h Meters	Velocity, Meters per Second	Velocity Head or Fall, h Meters	Velocity, Meters per Second	Velocity Head or Fall, h Meters	Velocity, Meters per Second	Velocity Head or Fall, h Meters	Velocity, Meters per Second
3.22	7.948	3.55	8.345	3.88	8.725	4.21	9.088	4.54	9.437
3.23	7.960	3.56	8.357	3.89	8.736	4.22	9.099	4.55	9.448
3.24	7.973	3.57	8.369	3.90	8.747	4.23	9.109	4.56	9.458
3.25	7.985	3.58	8.380	3.91	8.758	4.24	9.120	4.57	9.468
3.26	7.997	3.59	8.392	3.92	8.769	4.25	9.131	4.58	9.479
3.27	8.009	3.60	8.404	3.93	8.780	4.26	9.142	4.59	9.489
3.28	8.022	3.61	8.415	3.94	8.792	4.27	9.152	4.60	9.500
3.29	8.034	3.62	8.427	3.95	8.803	4.28	9.163	4.61	9.510
3.30	8.046	3.63	8.439	3.96	8.814	4.29	9.174	4.62	9.520
3.31	8.085	3.64	8.450	3.97	8.825	4.30	9.185	4.63	9.530
3.32	8.070	3.65	8.462	3.98	8.836	4.31	9.195	4.64	9.541
3.33	8.082	3.66	8.474	3.99	8.847	4.32	9.206	4.65	9.551
3.34	8.095	3.67	8.485	4.00	8.858	4.33	9.217	4.66	9.561
3.35	8.107	3.68	8.497	4.01	8.869	4.34	9.227	4.67	9.572
3.36	8.119	3.69	8.508	4.02	8.880	4.35	9.238	4.68	9.582
3.37	8.131	3.70	8.520	4.03	8.892	4.36	9.248	4.69	9.592
3.38	8.143	3.71	8.531	4.04	8.903	4.37	9.259	4.70	9.602
3.39	8.155	3.72	8.543	4.05	8.914	4.38	9.270	4.71	9.612
3.40	8.167	3.73	8.554	4.06	8.925	4.39	9.280	4.72	9.623
3.41	8.179	3.74	8.566	4.07	8.936	4.40	9.291	4.73	9.633
3.42	8.191	3.75	8.577	4.08	8.946	4.41	9.301	4.74	9.643
3.43	8.203	3.76	8.588	4.09	8.957	4.42	9.312	4.75	9.653
3.44	8.215	3.77	8.600	4.10	8.968	4.43	9.322	4.76	9.663
3.45	8.227	3.78	8.611	4.11	8.979	4.44	9.333	4.77	9.673
3.46	8.239	3.79	8.622	4.12	8.990	4.45	9.343	4.78	9.684
3.47	8.251	3.80	8.634	4.13	9.001	4.46	9.354	4.79	9.694
3.48	8.263	3.81	8.645	4.14	9.012	4.47	9.364	4.80	9.704
3.49	8.274	3.82	8.657	4.15	9.023	4.48	9.375	4.81	9.714
3.50	8.286	3.83	8.668	4.16	9.034	4.49	9.385	4.82	9.724
3.51	8.298	3.84	8.679	4.17	9.045	4.50	9.396	4.83	9.734
3.52	8.310	3.85	8.691	4.18	9.055	4.51	9.406	4.84	9.744
3.53	8.322	3.86	8.702	4.19	9.066	4.52	9.417	4.85	9.754
3.54	8.333	3.87	8.713	4.20	9.077	4.53	9.427	4.86	9.764

TABLE 2—*Continued*

Table of Theoretical Velocity $v=\sqrt{2gh}$ Corresponding to Different Velocity Heads h

Velocity Head or Fall, h Meters	Velocity, Meters per Second	Velocity Head or Fall, h Meters	Velocity, Meters per Second	Velocity Head or Fall, h Meters	Velocity, Meters per Second	Velocity Head or Fall, h Meters	Velocity, Meters per Second	Velocity Head or Fall, h Meters	Velocity, Meters per Second
4.87	9.774	10	14.006	43	29.044	76	38.613	145	53.334
4.88	9.784	11	14.690	44	29.380	77	38.866	150	54.246
4.89	9.794	12	15.343	45	29.712	78	39.117	155	55.143
4.90	9.804	13	15.970	46	30.040	79	39.367	160	56.025
4.91	9.814	14	16.572	47	30.365	80	39.616	165	56.894
4.92	9.824	15	17.154	48	30.686	81	39.863	170	57.749
4.93	9.834	16	17.718	49	31.004	82	40.182	175	58.592
4.94	9.844	17	18.263	50	31.329	83	40.352	180	59.424
4.95	9.854	18	18.791	51	31.631	84	40.594	185	60.243
4.96	9.864	19	19.306	52	31.939	85	40.835	190	61.052
4.97	9.874	20	19.808	53	32.245	86	41.074	195	61.850
4.98	9.884	21	20.297	54	32.548	87	41.313	200	62.638
4.99	9.894	22	20.775	55	32.848	88	41.549	205	63.416
5.00	9.904	23	21.242	56	33.145	89	41.785	210	64.185
5.25	10.489	24	21.698	57	33.440	90	42.019	215	64.944
5.50	10.387	25	22.146	58	33.732	91	42.252	220	65.695
5.75	10.621	26	22.584	59	34.021	92	42.483	225	66.438
6.00	10.849	27	23.015	60	34.408	93	42.713	230	67.171
6.25	11.073	28	23.437	61	34.593	94	42.942	235	67.898
6.50	11.292	29	23.852	62	34.875	95	43.170	240	68.616
6.75	11.507	30	24.260	63	35.155	96	43.397	245	69.328
7.00	11.718	31	24.661	64	35.433	97	43.622	250	70.031
7.25	11.926	32	25.055	65	35.709	98	43.847	255	70.728
7.50	12.130	33	25.444	66	35.983	99	44.070	260	71.418
7.75	12.330	34	25.826	67	36.254	100	44.299	265	72.102
8.00	12.528	35	26.203	68	36.524	105	45.386	270	72.780
8.25	12.722	36	26.575	69	36.791	110	46.454	275	73.450
8.50	12.913	37	26.942	70	37.057	115	47.498	280	74.114
8.75	13.102	38	27.303	71	37.321	120	48.519	285	74.773
9.00	13.288	39	27.660	72	37.583	125	49.520	290	75.446
9.25	13.471	40	28.013	73	37.843	130	50.500	295	76.074
9.50	13.652	41	28.361	74	38.101	135	51.462	300	76.726
9.75	13.830	42	28.804	75	38.385	140	52.407		

APPENDIX IV

FRICTION OF GASES FLOWING THROUGH BRICK FLUES OR MAINS (1)

BY W. A. MOJAROW

IN the flow of gases beneath an inverted weir, or, as it may be designated, inverted flow, the stream of gas may or may not fill the flue or furnace to its full height; practically, it may be said that in those cases where the stream of gases does not touch the hearth or bottom of the flue, the depth or vertical thickness of the inverted stream will be given by the formula developed by M. J. Yesmann, Professor of Hydraulics at the Polytechnic Institute of Petrograd:

$$h_t = A\sqrt[3]{\frac{Q_t^2}{B^2 t}},$$

in which Q_t = the volume of gas flowing at the temperature $t°$;

B = the length of crest of the inverted weir, that is, the width of the furnace or flue;

A = a coefficient which is not constant and varies from 2.97 to 3.62 according to h_t and B.

In order to arrive at the actual movement of the gases it is necessary to take account of the drop in pressure and, likewise, of the decrease in velocity which occurs at the different parts of the furnace or flue.

These losses are caused by:

1. Changes in the direction of flow;
2. Changes in the cross-sectional area of the channel;
3. Friction against the walls of the channel.

Information regarding the first two losses and changes of

(1) *Revue de la Société russe de Métallurgie, I, pp. 335–371 (1913). Extrait de Revue de Métallurgie, XI bis, p. 320, mai 1914.*

velocity may be found in the theory of ventilation and these may be applied; but at the present time there is no series of observations which will permit the computation of the frictional losses.

Calculations by the author show that these frictional losses may be evaluated by the following expression:

$$\gamma = m \cdot \frac{SL}{\omega} \cdot v_t \cdot p_t = m \cdot \frac{SL}{\omega^2} q_0 p_0 = m \frac{B}{\omega^2} q_0 p_0, \quad . \quad . \quad . \quad (A)$$

in which γ = millimeters of water column;

m = the coefficient of friction of the gas against the brick;

S = the perimeter of the channel in meters, or that portion of the perimeter " wet " by the gas;

ω = the cross-sectional area of the channel or stream of gas in square meters;

L = the length in meters of the portion of the channel considered;

$B = SL$ = the superficial wall area of the portion of the channel considered;

v_t = the velocity of flow of the stream of gas at the temperature $t°$ in meters per second;

p_t = the weight in kilograms per cubic meter of the gas at the average temperature $t°$;

$t°$ = the average temperature of the gas in the part of the channel considered;

q_0 = the volume of the gas at 0° and 760 mm pressure;

p_0 = the specific weight of the gas at 0° and 760 mm.

According to Professor Groume-Grjimailo, under the usual working conditions of metallurgical furnaces, the gases passing through the internal channel of these furnaces should always be considered with reference to the air which forms the external atmosphere, and the difference between the density of the hot gases and the air supplies the motive force for the flow of these gases; therefore, taking as the basis of comparison air at 0° and 760 mm which has a specific weight $p_0 = 1.29$ kg per cubic meter, it follows that if $V_0 = 1.0$ m per second and $L = 1.0$ m, the above formula becomes:

$$\gamma = m \cdot \frac{S \cdot 1}{\omega} 1.293,$$

in which $\gamma = m$ if $\frac{\omega}{S} = 1.293$. In other words, *the coefficient m is* *only the head, expressed in millimeters of water, corresponding to the friction loss per second of air at* 0° *and* 760 *mm passing through a brick-lined flue with a velocity of 1 m per second, the sectional area of the flue corresponding to the equation* $\frac{\omega}{S}=1.293$.

According to the computations of the author, 0.016 may be taken as the value of the coefficient m with sufficient precision for the computations met with in metallurgy. The verification of this coefficient was made by using it in computations upon Cowper and Massick and Crook hot-blast stoves.

The formula (A) may be used for all cases in which a current of gases is flowing through mains or flues.

1. If the current of flowing gas is subdivided into a number of channels or streams of equal area, the friction loss in each of these secondary channels is

$$\gamma_1 = m \cdot \frac{s}{\omega_n^2} \cdot \frac{Q}{n} p_0,$$

in which

$$s = \frac{S}{n} \quad \text{and} \quad \omega_n = \frac{\omega}{n};$$

n being the number of secondary channels or flues.

2. If the cross-sectional areas of these secondary channels or flues are not equal (as, for example, in the gas and air flues of an open-hearth furnace), the values $B=SL$ and ω in the formula (A), for the parts located between the point where the gaseous currents separate and the point where the secondary flues join comprise, respectively, the entire surface touched by the gases in passing and the sum of the average areas of all the channels or flues through which the gases pass.

3. If the gas in motion does not fill the furnace to its full height, the depth of the gaseous stream is computed by the formula for the inverted weir. The value given to the perimeter of the channel should be based upon the depth of the stream of gases. As the lower stream of the gas in motion slides on top of the immobile layer of the same gas in the bottom portion of the flue, it is necessary to take account of the frictional resistance by considering the entire perimeter of the flowing stream of gases, including the " free lower surface " of the stream, and the cross-sectional area of the stream as determined by the depth of the stream in the flue.

APPENDIX V

MUFFLE FURNACE FOR TEMPERING AND ANNEALING STEEL AT THE PETROGRAD ARSENAL [1]

THIS furnace supplies an interesting example of the application of hydraulic laws to the computation of furnaces. An old furnace of English construction with a cast-iron muffle and four dampers, working with an updraft, was in operation at the Petrograd Arsenal. It did not heat well and the heat was poorly distributed. This furnace was replaced by a very costly one, which was constructed according to a German patent and also worked with an updraft. The muffle was of brick. This furnace, notwithstanding its complex design, could never be made to heat uniformly.

Upon the request of the superintendent of the forge, Professor Groume-Grjimailo studied the conditions and recommended the reconstruction of the old English furnace, in such a manner that it would work on *the downdraft principle*. This was done according to Figs. 133 to 136. The binding of the furnace was not altered.

After reconstruction the furnace worked perfectly. It was practically impossible to detect any difference in temperature between the different portions of the large muffle (3600×1000×100 mm) by the use of two Le Chatelier pyrometers. The temperature of the muffle could be changed as desired, and it could be made to vary between large limits (900° to 400°) by changing the setting of the damper in the waste-gas flue.

A short calculation was made later concerning this furnace. The computed consumption of fuel was 55 kg 80 per hour, about 640 kg per day. The actual consumption per day was 690 kg. This estimate was based entirely upon the analysis of the gases at different points within the furnace.

A cast-iron muffle was chosen for this furnace. A similar

[1] *Revue de la Société russe de Métallurgie, I, pp. 423–426 (1913). Revue de Métallurgie, mai 1914 (p. 115, MS.).*

Plan

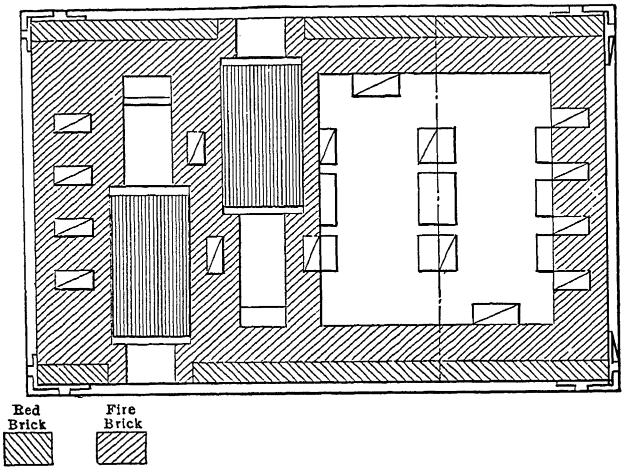


FIG. 133.—Horizontal Sections.

Section a, b, c, g, e, m

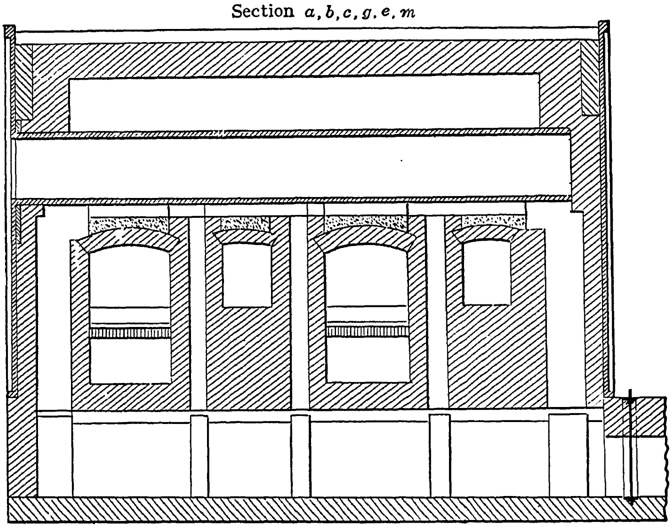


FIG. 134.—Longitudinal Section.

vertical muffle for continuous service, constructed according to the designs of the author, was in use at the Poutiloff works (Petrograd). In the arsenal the daily interruption of the work resulted in the rapid wearing out of the muffle, and after a run of two months it was decided to remove the cast-iron muffle and operate the furnace with a perforated brick hearth through which waste gases could pass off. The furnace was reconstructed in this manner, without a muffle, and upon being placed in operation worked perfectly, heating very uniformly and fulfilling perfectly all the demands for annealing and tempering which were made upon it.

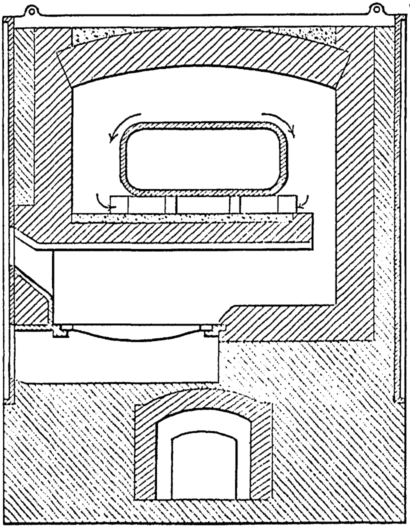


FIG. 135.—Cross-section through Firebox.

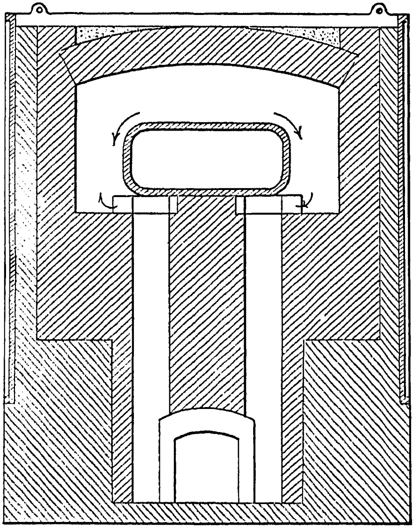


FIG. 136.—Cross-section through Waste Gas Ports.

The following shows the computations which were made:

The volume of the heating chamber, according to the design, was 3 m³ 30; the temperature of the gas was to be 1000°, the combustion being neutral, that is to say, without excess air. The instantaneous calorific intensity was computed as 2000°, giving a drop in temperature of the gases of 200° per second, the time which the gases remain in the heating chamber being

$$\frac{2000-1000}{200}=5 \text{ seconds.}$$

The volume of gas per second is found as follows:

$$\theta_{1000} = 3.3 : 5 = 0 \text{ m}^3\ 66$$

$$\theta_1 = \frac{\theta_{1000}}{1+\frac{1000}{273}} = 0 \text{ m}^3\ 14$$

The consumption of coal per second is arrived at as follows: 1 kg of coal gives about 9 m^3 00 of gas, and, accordingly:

$$x = \frac{0.14}{9.00} = 0 \text{ kg } 0155 \text{ per second} = \text{about } 55 \text{ kg } 80 \text{ per hour.}$$

The dimensions of the fireboxes were the same as before,

$$\omega = 4 \times 1 \times 0.56 = 2 \text{ m}^2\ 24,$$

and corresponded with a fuel consumption of 25 kg of coal per square meter of grate per hour. The rate of combustion was rather slow, a condition which is particularly favorable for realizing theoretical combustion.

The area of the hot gas ports from the firebox:

$$\omega_1 = 4 \times 0.16 \times 0.32 = 0 \text{ m}^2\ 20.$$

The velocity in these parts is low:

$$v_{1000} = 0.66 : 0.20 = 3 \text{ m } 20 \text{ per second.}$$

The area of the waste gas ports is

$$\omega_2 = 16 \times 0.24 \times 0.12 = 0 \text{ m}^2\ 46.$$

The velocity in these ports is

$$v_{1000} = 0.66 : 0.46 = 1 \text{ m } 43 \text{ per second.}$$

These low velocities make it certain that the waste-gas flue will receive the coolest gases in the chamber and are also very favorable to the obtaining of a uniform temperature over the whole hearth of the muffle.

The height of the waste-gas flue according to Yesmann's formula is 0 m 38.

The computation for the chimney was not made, as the existing chimney was amply sufficient and gave a large margin of safety, the velocity of the gases at its outlet being very nearly zero.

These computations have once more confirmed the theory of naces developed by Professor Groume-Grjimailo, based upon draulic laws.

In the author's opinion the computation for a furnace has ome a problem so simple and clear, that, *when a furnace has n correctly computed it cannot fail to work properly*; it has ome possible to guarantee the correct working of a furnace hout any more risk than is involved in the guarantee of the formance of a turbine or a steam engine. It only remains to ain the best types of furnaces for each particular purpose, and s is impossible without the cooperation of the works and those o direct them.

APPENDIX VI

ARCH BRICK WORK

FORMULAS AND TABLES FOR FANTAIL OR TAPER ARCHES

BY A. D. WILLIAMS

TABLES giving the number of shape and straight brick required to make a complete circle are found in many of the firebrick and facebrick catalogues. These are very convenient, as far as they extend, in regard to the standard shapes, but are of comparatively little assistance in working out the shapes required for "fantail" or taper arches, which may be built either of standard or special shapes as desired, or for conical or skew arches.

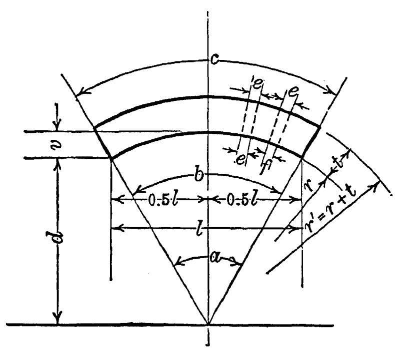


FIG. 137.

The accompanying tables were computed to facilitate the working out of arch shapes, and it is believed that the accompanying diagram supplies sufficient information to facilitate their use. They have been carefully computed and checked, but the writer will be grateful to anyone who will call his attention to any errors which may have been made in the tables.

In the diagram the letters and following formulas correspond:

$$r=\frac{(0.5l)^2+v^2}{2v}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

$$\sin\frac{a}{2}=\frac{0.5l}{r}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

$$b=\frac{6.2832r\times a(\text{deg.})}{360} \quad . \quad . \quad . \quad . \quad . \quad (3)$$

or

$$=0.01754r\times a(\text{deg.}).$$

$$d=r-v. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

$$r'=r+t. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

$$c=0.01754r'\times a(\text{deg.}). \quad . \quad . \quad . \quad . \quad (6)$$

$$k=c-b. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

$$g=e-f. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

$$\frac{c}{e}=n=\text{total number of brick in span.} \quad . \quad . \quad . \quad (9)$$

$$\frac{k}{g}=n'=\text{number of taper brick in span.} \quad . \quad . \quad (10)$$

$$n-n'=n''=\text{number of straight brick in span.} \quad . \quad (11)$$

Thrust, uniform loading $=T_u=\dfrac{3P_u l}{2v}$.

Thrust, concentrated load $=T_c=\dfrac{3P_c l}{v}$.

P_u = uniform load, pounds per square foot.

P_c = concentrated load in pounds at center of span.

When several concentrated loads are to be considered, reduce to an equivalent concentrated center load.

TABLE 1

FORMULAS AND TABLES FOR ARCH BRICK

Firebrick Arches; Rise, Length, Radius, for Chord of Unity

No. 1	No. 2	No. 3	No. 4	No. 5	No. 6	No. 7	No. 8
Rise or Middle Ordinate Decimal of Span or Chord	Logarithm or Rise or Middle Ordinate	Rise or Middle Ordinate Inches per 1 Ft Span or Chord	Arc	Length of Arc	Logarithm of Length of Arc	Radius of Arc	Logarithm of Radius of Arc
v	log v	v	a, Degrees, Minutes and Seconds	b or c	log b log c	r or r'	log r log r'
0.0300	8.47712	......	13–44–04	1.00240	0.00104	4.1816	0.62135
0.0313	8.49485	$\frac{3}{8}$	14–19–42	1.00261	0.00113	4.0092	0.60306
0.0328	8.51550	......	15–00–00	1.00285	0.00124	3.8307	0.58327
0.0339	8.52961	$\frac{13}{32}$	15–30–52	1.00305	0.00132	3.7043	0.58671
0.0350	8.54407	......	16–01–00	1.00327	0.00142	3.5889	0.55496
0.0365	8.56180	$\frac{7}{16}$	16–19–05	1.00355	0.00154	3.4429	0.53693
0.0391	8.59176	$\frac{15}{32}$	17–53–09	1.00407	0.00176	3.2313	0.50738
0.0400	8.60206	......	18–17–44	1.00426	0.00185	3.1450	0.49762
0.0417	8.61979	$\frac{1}{2}$	19–04–11	1.00462	0.00192	3.0254	0.47978
0.0443	8.64612	$\frac{17}{32}$	20–11–00	1.00522	0.00226	2.8535	0.45538
0.0450	8.65321	......	20–34–18	1.00539	0.00233	2.8002	0.44719
0.0469	8.67094	$\frac{9}{16}$	21–26–23	1.00585	0.00253	2.6887	0.42954
0.0493	8.69237	......	22–30–00	1.00645	0.00279	2.5629	0.40873
0.0495	8.69442	$\frac{19}{32}$	22–36–56	1.00652	0.00282	2.5500	0.40654
0.0500	8.69897	......	22–50–32	1.00665	0.00288	2.5250	0.40226
0.0521	8.71670	$\frac{5}{8}$	23–47–44	1.00723	0.00313	2.4254	0.38476
0.0547	8.73789	$\frac{21}{32}$	24–58–24	1.00796	0.00344	2.3125	0.36409
0.0550	8.74036	......	25–06–34	1.00805	0.00348	2.3002	0.36177
0.0573	8.75809	$\frac{11}{16}$	26–09–01	1.00873	0.00378	2.2102	0.34442
0.0599	8.77740	$\frac{23}{32}$	27–19–34	1.00954	0.00412	2.1168	0.32567
0.0600	8.77815	......	27–22–16	1.00957	0.00414	2.1133	0.32497
0.0625	8.79588	$\frac{3}{4}$	28–30–00	1.01088	0.00470	2.0314	0.30776
0.0650	8.81291	......	29–37–42	1.01123	0.00485	1.9555	0.29126
0.0651	8.81361	$\frac{25}{32}$	29–40–22	1.01127	0.00487	1.9527	0.29063
0.0658	8.81840	......	30–00–00	1.01153	0.00498	1.9319	0.28597
0.0677	8.83064	$\frac{13}{16}$	30–50–42	1.01217	0.00525	1.8802	0.27421
0.0700	8.84510	......	31–52–43	1.01302	0.00562	1.8207	0.26024

FORMULAS AND TABLES FOR ARCH BRICK—*Continued*

Firebrick Arches; Rise, Length, Radius, for Chord of Unity

No. 1	No. 2	No. 3	No. 4	No. 5	No. 6	No. 7	No. 8
Rise or Middle Ordinate Decimal of Span or Chord	Logarithm of Rise or Middle Ordinate	Rise or Middle Ordinate Inches per 1 Ft Span or Chord	Arc	Length of Arc	Logarithm of Length of Arc	Radius of Arc	Logarithm of Radius of Arc
v	log v	v	a, Degrees, Minutes and Seconds	b or c	log b log c	r or r'	log r log r'
0.0703	8.84703	$\frac{27}{32}$	32–00–48	1.01313	0.00567	1.8132	0.25846
0.0729	8.86283	$\frac{7}{8}$	33–06–09	1.01410	0.00608	1.7552	0.24432
0.0750	8.88081	......	34–07–23	1.01493	0.00644	1.7042	0.23151
0.0755	8.87807	$\frac{29}{32}$	34–20–50	1.01513	0.00652	1.6934	0.22875
0.0781	8.89279	$\frac{15}{16}$	35–30–42	1.01618	0.00697	1.6396	0.21473
0.0800	8.90309	......	36–21–42	1.01698	0.00731	1.6025	0.20479
0.0807	8.90703	$\frac{31}{32}$	36–40–23	1.01728	0.00728	1.5893	0.20120
0.0826	8.91672	......	37–30–00	1.01807	0.00778	1.5555	0.19187
0.0833	8.92082	1	37–50–04	1.01841	0.00742	1.5422	0.18815
0.0850	8.92942	......	38–35–32	1.01916	0.00824	1.5131	0.17986
0.0859	8.93418	$1\frac{1}{32}$	38–59–32	1.01956	0.00841	1.4982	0.17656
0.0885	8.94714	$1\frac{1}{16}$	40–08–58	1.02075	0.00892	1.4567	0.16336
0.0900	8.95424	......	40–49–03	1.02146	0.00922	1.4339	0.15652
0.0911	8.95974	$1\frac{3}{32}$	41–18–14	1.02197	0.00944	1.4177	0.15158
0.0938	8.97197	$1\frac{1}{8}$	42–30–03	1.02329	0.00999	1.3795	0.13973
0.0950	8.97772	......	43–08–26	1.02389	0.01025	1.3663	0.13355
0.0964	8.98387	$1\frac{5}{32}$	43–39–42	1.02460	0.01055	1.3446	0.12858
0.0990	8.99545	$1\frac{3}{16}$	44–47–58	1.02593	0.01112	1.3121	0.11797
0.0995	8.99763	......	45–00–00	1.02617	0.01122	1.3066	0.11613
0.1000	9.00000	......	45–14–23	1.02646	0.01134	1.3000	0.11394
0.1016	9.00673	$1\frac{7}{32}$	45–56–36	1.02731	0.01170	1.2811	0.10760
0.1042	9.01773	$1\frac{1}{4}$	47–04–35	1.02871	0.01229	1.2520	0.09761
0.1050	9.02119	......	47–26–20	1.02914	0.01247	1.2430	0.09446
0.1068	9.02845	$1\frac{9}{32}$	48–13–44	1.03015	0.01290	1.2238	0.08771
0.1094	9.03892	$1\frac{5}{16}$	49–16–16	1.03162	0.01352	1.1995	0.07900
0.1100	9.04139	......	49–37–47	1.03196	0.01366	1.1914	0.07604
0.1120	9.04914	$1\frac{11}{32}$	50–30–12	1.03312	0.01415	1.1721	0.06895

FORMULAS AND TABLES FOR ARCH BRICK—*Continued*

Firebrick Arches; Rise, Length, Radius, for Chord of Unity

No. 1	No. 2	No. 3	No. 4	No. 5	No. 6	No. 7	No. 8
Rise or Middle Ordinate Decimal of Span or Chord	Logarithm of Rise or Middle Ordinate	Rise or Middle Ordinate Inches per 1 Ft Span or Chord	Arc	Length of Arc	Logarithm of Length of Arc	Radius of Arc	Logarithm of Radius of Arc
v	log v	v	a, Degrees, Minutes and Seconds	b or c	log b log c	r or r'	log r log r'
0.1146	9.09512	$1\frac{3}{8}$	51–38–13	1.03466	0.01476	1.1481	0.05996
0.1150	9.06070	......	51–48–39	1.03490	0.01490	1.1445	0.05860
0.1167	9.06664	......	52–30–00	1.03593	0.01533	1.1305	0.05326
0.1172	9.06888	$1\frac{13}{32}$	52–46–04	1.03623	0.01546	1.1252	0.05121
0.1198	9.07843	$1\frac{7}{16}$	53–53–47	1.03784	0.01613	1.1033	0.04270
0.1200	9.07918	......	53–58–58	1.03797	0.01619	1.1017	0.04205
0.1224	9.08777	$1\frac{15}{32}$	55–01–11	1.03949	0.01682	1.0825	0.03442
0.1250	9.09691	$1\frac{1}{2}$	56–08–42	1.04116	0.01752	1.0625	0.02633
0.1276	9.10586	$1\frac{17}{32}$	57-15–54	1.04287	0.01823	1.0434	0.01846
0.1300	9.11394	......	58–17–48	1.04447	0.01890	1.0265	0.01138
0.1302	9.11464	$1\frac{9}{16}$	58–22–57	1.04461	0.01895	1.0252	0.01079
0.1328	9.12324	$1\frac{19}{32}$	59–29–50	1.04638	0.01969	1.0077	0.00332
0.1340	9.12702	......	60–00–00	1.04715	0.02002	1.0000	9.00000
0.1350	9.13033	......	60–26–18	1.04792	0.02033	0.9934	9.99714
0.1354	9.13167	$1\frac{5}{8}$	60–37–28	1.04820	0.02040	0.9907	9.99593
0.1400	9.14613	......	62–43–28	1.05147	0.02180	0.9607	9.98259
0.1406	9.14806	$1\frac{11}{16}$	62–49–26	1.05190	0.02197	0.9594	9.98198
0.1450	9.16137	......	64–41–19	1.05516	0.02332	0.9346	9.97061
0.1458	9.16386	$1\frac{3}{4}$	65–01–39	1.05576	0.02357	0.9302	9.96859
0.1500	9.17609	......	66–47–30	1.05869	0.02488	0.9085	9.95825
0.1510	9.17910	$1\frac{13}{16}$	67–13–02	1.05973	0.02520	0.9033	9.95584
0.1517	9.18091	......	67–30–00	1.06034	0.02545	0.9000	9.95423
0.1550	9.19033	......	68–53–37	1.06288	0.02649	0.8840	9.94643
0.1563	9.19382	$1\frac{7}{8}$	69–26–13	1.06392	0.02691	0.8779	9.94344
0.1600	9.20412	......	70–58–44	1.06693	0.02814	0.8613	9.93513
0.1615	9.20806	$1\frac{15}{16}$	71–36–06	1.06816	0.02864	0.8547	9.93184
0.1650	9.21748	......	73–03–06	1.07109	0.02983	0.8401	9.92432

FORMULAS AND TABLES FOR ARCH BRICK—*Continued*

Firebrick Arches; Rise, Length, Radius, for Chord of Unity

No. 1	No. 2	No. 3	No. 4	No. 5	No. 6	No. 7	No. 8
Rise or Middle Ordinate Decimal of Span or Chord	Logarithm of Rise or Middle Ordinate	Rise or Middle Ordinate Inches per 1 Ft Span or Chord	Arc	Length of Arc	Logarithm of Length of Arc	Radius of Arc	Logarithm of Radius of Arc
v	log v	v	a, Degrees, Minutes and Seconds	b or c	log b log c	r or r'	log r log r'
0.1667	9.22185	2	73–44–08	1.07254	0.03042	0.8334	9.92084
0.1697	9.22975	......	75–00–00	1.07511	0.03145	0.8213	9.91452
0.1700	9.23045	......	75–00–44	1.07537	0.03156	0.8203	9.91397
0.1719	9.23521	$2\frac{1}{16}$	75–53–30	1.07702	0.03222	0.8131	9.91015
0.1750	9.24551	......	77–09–36	1.07977	0.03333	0.8018	9.90406
0.1771	9.24818	$2\frac{1}{8}$	78–01–12	1.08165	0.03409	0.7943	9.90001
0.1800	9.25527	......	79–11–44	1.08428	0.03514	0.7844	9.89456
0.1823	9.26077	$2\frac{3}{16}$	80–07–39	1.08639	0.03599	0.7768	9.89033
0.1850	9.26717	......	81–13–04	1.08890	0.03699	0.7682	9.88546
0.1875	9.27300	$2\frac{1}{4}$	82–13–27	1.09126	0.03793	0.7604	9.88105
0.1882	9.27459	......	82–30–00	1.09193	0.03819	0.7583	9.87986
0.1900	9.27875	......	83–13–38	1.09365	0.03888	0.7529	9.87673
0.1927	9.28490	$2\frac{5}{16}$	84–13–17	1.09625	0.03991	0.7456	9.87253
0.1950	9.29003	......	85–13–23	1.09850	0.04080	0.7385	9.86837
0.1979	9.29648	$2\frac{3}{8}$	86–22–29	1.10137	0.04233	0.7306	9.86367
0.2000	9.30103	......	87–12–20	1.10347	0.04276	0.7250	9.86034
0.2031	9.30776	$2\frac{7}{16}$	88–27–14	1.10660	0.04399	0.7169	9.85543
0.2050	9.31175	......	89–26–10	1.10855	0.04476	0.7106	9.85164
0.2071	9.31619	......	90–00–00	1.11072	0.04560	0.7072	9.84949
0.2083	9.31876	$2\frac{1}{2}$	90–29–34	1.11196	0.04605	0.7041	9.84762
0.2100	9.32222	......	91–41–32	1.11374	0.04680	0.6969	9.84315
0.2135	9.32948	$2\frac{9}{16}$	92–29–24	1.11743	0.04822	0.6922	9.84025
0.2188	9.33995	$2\frac{5}{8}$	94–32–12	1.12312	0.05043	0.6807	9.83295
0.2200	9.34242	......	94–59–53	1.12444	0.05094	0.6782	9.83135
0.2240	9.35017	$2\frac{11}{16}$	96–29–13	1.12885	0.05264	0.6703	9.82624
0.2265	9.35516	......	97–30–00	1.13168	0.05372	0.6650	9.82284
0.2292	9.36015	$2\frac{3}{4}$	98–30–05	1.13466	0.05487	0.6600	9.81953
0.2300	9.36173	......	98–48–35	1.13557	0.05521	0.6585	9.81854

FORMULAS AND TABLES FOR ARCH BRICK—*Continued*

Firebrick Arches; Rise, Length, Radius, for Chord of Unity

No. 1	No. 2	No. 3	No. 4	No. 5	No. 6	No. 7	No. 8
Rise or Middle Ordinate Decimal of Span or Chord	Logarithm of Rise or Middle Ordinate	Rise or Middle Ordinate Inches per 1 Ft Span or Chord	Arc	Length of Arc	Logarithm of Length of Arc	Radius of Arc	Logarithm of Radius of Arc
v	log v	v	a, Degrees, Minutes and Seconds	b or c	log b log c	r or r'	log r log r'
0.2344	9.36991	$2\frac{13}{16}$	100–28–08	1.14061	0.05714	0.6505	9.81323
0.2396	9.37946	$2\frac{7}{8}$	102–24–54	1.14667	0.05944	0.6415	9.80720
0.2400	9.38021	......	102–33–50	1.14714	0.05962	0.6408	9.80675
0.2448	9.38880	$2\frac{15}{16}$	104–20–44	1.15284	0.06177	0.6330	9.80142
0.2465	9.39195	......	105–00–00	1.15621	0.06304	0.6302	9.79950
0.2500	9.39794	3	106–15–36	1.15912	0.06413	0.6250	9.79588
0.2600	9.41497	......	109–53–51	1.17150	0.06874	0.6108	9.78588
0.2604	9.41567	$3\frac{1}{8}$	110–04–25	1.17200	0.06893	0.6101	9.78541
0.2673	9.42693	......	112–30–00	1.18072	0.07215	0.6013	9.77912
0.2700	9.43136	......	113–28–34	1.18429	0.07346	0.5980	9.77667
0.2708	9.43270	$3\frac{1}{4}$	113–45–36	1.18533	0.07384	0.5970	9.77597
0.2800	9.44716	......	117–25–40	1.19746	0.07826	0.5851	9.76722
0.2813	9.44909	$3\frac{3}{8}$	117–26–54	1.19920	0.07889	0.5850	9.76717
0.2821	9.45041	......	120–00–00	1.20027	0.07928	0.5774	9.76144
0.2900	9.46240	......	120–30–04	1.21102	0.08315	0.5759	9.76035
0.2917	9.46489	$3\frac{1}{2}$	121–02–15	1.21336	0.08399	0.5744	9.75919
0.3000	9.47712	......	123–51–18	1.22495	0.08812	0.5667	9.75333
0.3100	9.49136	......	127–08–33	1.23926	0.09316	0.5584	9.74691
0.3109	9.49265	......	127–30–00	1.24059	0.09363	0.5575	9.74624
0.3125	9.49485	$3\frac{3}{4}$	128–01–18	1.24289	0.09443	0.5563	9.74527
0.3200	9.50515	......	130–28–37	1.25391	0.09827	0.5506	9.74086
0.3300	9.51851	......	134–19–16	1.26892	0.10343	0.5425	9.73443
0.3333	9.52288	4	134–45–44	1.27395	0.10515	0.5417	9.73373
0.3341	9.52386	......	135–00–00	1.27517	0.10557	0.5412	9.73335
0.3400	9.53148	......	136–51–46	1.28428	0.10866	0.5377	9.73050
0.3500	9.54407	......	139–58–05	1.29997	0.11393	0.5321	9.72603
0.3541	9.54921	$4\frac{1}{4}$	141–13–56	1.30650	0.11611	0.5301	9.72431
0.3600	9.55630	......	142–14–08	1.31599	0.11925	0.5284	9.72299

MISCELLANEOUS TABLES

TABLE 2

Operation	Chamber Volume	
	Cubic Meters per 1000 Kg. of Coal Burned per Hour	Cubic Feet per 1000 Pounds of Coal Burned per Hour
Kilns, Ordinary brick	1000	16,018
Porcelain	120	1,923
Refractory brick	35	560
Glass, bottles	35	560
Puddling furnaces	11	177
Crucible steel furnaces	3	50
Reverberatory furnaces, Iron melting	2	35
Steel melting	1	17

	Ratio between Grate Area and Hearth Area
Roasting of ores, etc	15
Copper melting	5
Iron melting	3
Puddling	2
Steel melting	1

From Le Chatelier, Le Chauffage Industriel.

TABLE 3

BESSEMER CONVERTERS

Blast pressure about 25 pounds per square inch = 1 kg 76 per square centimeter.

Blast volume about 900 to 1100 cubic feet of free air per ton per minute = 25 m^3 50 to 31 m^3 20 per tonne.

Blast pressure must be about seven times the pressure due to the static head of the molten metal.

Molten metal weighs about 430 pounds per cubic foot = 6888 kg per cubic meter.

Depth of metal in vessel about 12 inches = 300 mm.

Static head of metal about 3 pounds per square inch = 0 kg 21 per square centimeter.

Tuyere blocks are about 24 inches = 600 mm in depth.

Tuyere holes are about 0.5 inch = 12 to 15 mm diameter. Five to 7 holes per block.

TABLE 4

CHIMNEY DRAFT (Lange)

$$F_0 = \frac{BG(1+at_0)}{\gamma\delta\, 3600 v_n},$$

$$d_0 = \sqrt{\frac{4F_0}{\eta}}, \qquad F_0 = \frac{P_n Q(1+\alpha t)}{\delta v_n},$$

$$H_r = (15d_0 + 2.5v_n + al - 160tgi)\frac{700 - t_m}{700 + t_m},$$

$$z = H_r(\gamma - \gamma_1), \qquad d_u = d_0 + 0.016H_r,$$

$$tgi = \frac{d_u - d_0}{2h} = 0.008 \text{ to } 0.010,$$

in which F_0 = area of bore of chimney at top in square meters;
B = kilograms of coal burned per hour;
G = kilograms of flue gases per kilogram of coal;
$\alpha = 1 \div 273 = 0.00367$;
t_0 = temperature of gases at base of chimney;
$\gamma = 1.293$ = weight of a cubic meter of air at 0° and 760 mm;
δ = ratio between the weight of flue gases and air;
d_0 = diameter of bore of chimney at top;
$\eta = 3.1416$ for a circle;
3.3137 for an octagon;
4.00 for a square;
l = length of flue in meters;
a = flue coefficient 0.03 to 0.14 (usually 0.04);
H_r = height of chimney above grates in meters;
z = mm of water or kilograms per square meter;
γ_1 = weight of flue gases per cubic meter at 0° and 760 mm;
d_u = diameter of bore of chimney at base in meters
t_m = average temperature of flue gases;
p = kilograms of coal burned per second;
Q = waste gases per kilogram of coal, cubic meters;
v_n = velocity of gas flow, meters per second (from 3 to 15).

TABLE 5

Hot Blast Stoves, Empirical Data

B'ast temperatures range from 500° to 1000°.
Heating surface ranges from 7 to 25 square meters (average 20 square meters) per cubic meter per millimeter of blast. (Withwell Stoves.)
Cowper stoves. Velocity of cold blast in checker 1.50 to 2.00 m per sec.
Velocity of hot blast in checker 5.00 to 8.00 m per sec.
Volume to be taken at the temperature and pressure.

Le Chatelier states that the heat interchange rate is

650 cal. per square meter per hour for 250° temperature difference.

TABLE 6

Furnace Temperatures

From Le Chatelier and Damour:	
Glass tank, bottle glass	1425°
Annealing lehr	585°
Gas retort setting, top	1190°
Bottom	1060°
Porcelain kilns, hard porcelain	1400°
Chinaware	1275°
Brick kilns, various sources:	
Building brick, cones 015*a* to 2*a*	800°–1170°
Vitrified pavers, cones 1*a* to 14	1150°–1410°
Glazed brick, cones 6*a* to 9*a*	1250°–1310°
Firebrick, cones 6 to 20	1250°–1530°
From Harbison and Walker Co., Catalogue:	
Glass furnace between the pots	1375°
In pots, refining	1310°
In pots, working	1045°
Tanks melted for casting	1310°
Annealing glassware	444°– 555°
Crucible steel furnace, Siemens varies from	1460°–1590°
A rough method of estimating temperatures:	
Just glowing in the dark	525°
Dark red	700°
Cherry red	908°
Bright cherry red	1000°
Orange	1150°
White	1300°
Dazzling white	1500°

TABLE 7

FUEL CONSUMPTION OF FURNACES, POUNDS PER M (FROM VARIOUS SOURCES)

Brick kilns, Hoffman kilns, well built	280 to 340
Grates and troughs	340 to 450
Ruabon kiln	420 to 660
Staffordshire kiln	340 to 450
Buhrer continuous tunnel	230 to 280

C. E. Longnecker, pulverized coal consumption, pounds per ton (*Iron Age*, Feb. 6, 1919):

Puddling	1000
Busheling	400 to 500
Billet heating	160 to 180
Forging	400
Sheet annealing	200
Sheet and pair	300
Tin plate mill	170
Open-hearth	500 to 600
Copper reverberatory	300
Tin smelting	1400
Galvanizing pots	100
Tire furnaces	330
Wheel	600
Continuous bloom furnaces	100 to 150
Rivet making	90
Steam boilers	3.5 pounds per bhp per hour

Soaking pits, hot ingots	100 to 125
Reheating cold blooms	140 to 200
Continuous furnaces, cold billets	140 to 195

Open-hearth furnaces (G. L. Luetscher):

Natural gas	5600 to 6000 ft^3 per ton
Coal	500 to 650 pounds per ton
Oil	45 gallons per ton
Basic furnaces	750 pounds per ton
Acid furnaces	780 to 860 pounds per ton

Billet heating 4 inches square cold	200 to 250 pounds per ton

TABLE 8

VELOCITY OF CONVECTION CURRENTS RESULTING FROM VARIOUS AVERAGE TEMPERATURE DIFFERENTIALS ACTING FOR DIFFERENT HEIGHTS

Temperature Difference, Degrees F.	Velocity in Feet per Second			
	Height, 3 Inches	Height, 1 Foot	Height, 10 Feet	Height, 40 Feet
1	0.1811	0.3622	1.145	2.291
2	0.2561	0.5123	1.620	3.240
3	0.3137	0.6275	1.984	3.968
5	0.405	0.810	2.561	5.123
10	0.5727	1.1455	3.622	7.245
20	0.810	1.620	5.123	10.245
30	0.992	1.984	6.275	12.55
40	1.1455	2.291	7.245	14.49
50	1.2808	2.561	8.10	16.20
60	1.403	2.806	8.873	17.745
70	1.515	3.031	9.582	19.165
80	1.620	3.240	10.245	20.49
90	1.718	3.436	10.867	21.735
100	1.811	3.622	11.455	22.91
150	2.218	4.436	14.03	28.06
200	2.561	5.123	16.20	32.40
300	3.137	6.275	19.84	39.68
400	3.622	7.245	22.91	45.82
500	4.05	8.10	25.61	51.23
600	4.436	8.873	28.06	56.12
700	4.791	9.582	30.305	60.61
800	5.123	10.245	32.40	64.80
900	5.434	10.867	34.365	68.73
1000	5.727	11.455	36.225	72.45
1200	6.275	12.55	39.68	79.36
1400	6.776	13.552	42.86	85.72
1600	7.245	14.49	45.82	91.64
1800	7.684	15.368	48.60	97.20

TABLE 9

VELOCITY IN METERS PER SECOND OF CONVECTION CURRENTS RESULTING FROM VARIOUS AVERAGE TEMPERATURE DIFFERENTIALS ACTING FOR DIFFERENT HEIGHTS

Velocity in meters per second $= v = \sqrt{2gH\alpha t}$

$T_0 - T_1$, $T_0 - t_1$. Degrees	$H = 0.10$	$H = 1.00$	$H = 10.00$
1	0.0848	0.2683	0.8485
2	0.1200	0.3794	1.200
3	0.1470	0.4648	1.470
4	0.1697	0.5367	1.697
5	0.1897	0.6000	1.897
6	0.2078	0.6571	2.078
7	0.2245	0.7100	2.245
8	0.2400	0.7590	2.400
9	0.2546	0.8050	2.546
10	0.2683	0.8485	2.683
20	0.3794	1.200	3.794
30	0.4648	1.470	4.648
40	0.5367	1.697	5.367
50	0.6000	1.897	6.000
60	0.6571	2.078	6.571
70	0.7100	2.245	7.100
80	0.7590	2.400	7.590
90	0.8050	2.546	8.050
100	0.8485	2.683	8.485
200	1.200	3.794	12.00
300	1.470	4.648	14.70
400	1.697	5.367	16.97
500	1.897	6.000	18.97
600	2.078	6.571	20.78
700	2.245	7.100	22.45
800	2.400	7.590	24.00
900	2.546	8.050	25.46
1000	2.683	8.485	26.83

METRIC CONVERSION TABLE

		Logarithm
1 m 00	=3.2803 ft	0.515 9842
	=39.37 in	1.595 1654
	=1000 mm	3.000 0000
1.00 ft	=0 m 30 480	9.484 0158−10
	=304 mm 80	2.484 0158
	=12 in	1.079 1812
1.00 in	=25 mm 40 005	1.404 8346
1 m² 00	=10.7639 sq ft	1.031 9683
	=10,000 cm²	4.000 0000
1 sq ft	=0.0929034 m²	8.968 0316−10
	=929 cm² 034	2.968 0316
	=144 sq in	2.158 3624
1 cm² 00	=0.155 sq in	9.190 3308−10
	=100 mm² 00	2.000 0000
1 sq in	=6 cm² 45163	0.809 6692
	=645 mm² 163	2.809 6692
1 m³ 00	=35.3145 cu ft	1.547 9525
	=1000 liters	3.000 0000
	=1.58219 oz mols	0.199 2583
	=44.80287 gr mols	1.651 3058
	=264.170 gallons U. S.	2.421 8842
	=220.083 gallons British	2.342 5870
1.00 cu ft	=0 m³ 028317	8.452 0475−10
	=28.317 liters	1.452 0475
	=1728 cu in	3.237 5437
	=7.48052 gallons U. S	0.873 9317
	=6.23210 gallons British	0.794 6345
	=1.26868 gr mols	0.103 3533
	=0.044803 oz mol	8.651 3058−10
1.00 gram mol	=22.32 liters	1.348 6942
	=0 m³ 02232	8.348 6942−10
	=0.788219 cu ft	9.896 6467−10
	=0.035314 oz mol	8.547 9525−10
1.00 oz mol	=22.32 cu ft	1.348 6942
	=0 m³ 63236	9.800 7417−10
	=632.36 liters	2.800 7417
	=28.317 gram mols	1.452 0475
1 kg 00	=2.20462 lb	0.343 3340
	=1000 grams	3.000 0000
	=35.27392 oz	1.547 4540
1.00 lb	=0 kg 453953	9.656 6660−10
	=453 gr 593	2.656 6660
	=16 oz	1.204 1200

METRIC CONVERSION TABLE—*Continued*

		Logarithm
1.00 oz	=28 gr 3496	1.452 5461
	=0.0625 lb	8.795 8800−10
	=437.5 grains	2.640 9781
1 kg 00 per m²	=1 mm 00 of water	0.000
	=0 kg 0001 per cm²	6.000 0000−10
	=10 gr 00 per cm²	0.000 0000
	=0.03937 in of water	8.595 1654−10
	=0.001422 lb per sq in	7.153 0032−10
	=0.204817 lb per sq ft	9.311 3657−10
1 kg 00 per cm²	=1 metric atmosphere	0.000
	=0.967777 atmosphere	9.985 7752−10
	=735 mm 51 of mercury	2.866 5888
	=10000 kg 00 per m²	4.000 0000
	=10 m 00 of water	1.000 0000
	=14.2234 lb per sq in	1.153 0032
	=2048.17 lb per sq ft	3.311 3657
	=32.8083 ft of water	1.515 9842
	=393.699 in of water	2.595 1654
	=28.9570 in of mercury	1.461 7542
1.00 atmosphere	=760 mm of mercury	2.880 8136
	=29.9212 in of mercury	1.475 9790
	=1 kg 0332 per cm²	0.014 2248
	=14.697 lb per sq in	1.167 2280
	=10 m 332 of water	1.014 2248
	=33.9007 ft of water	1.530 2090
1.00 oz per sq in	=0.0625 lb per sq in	8.795 8800−10
	=9.00 lb per sq ft	0.954 2425
	=1.732 in of water	0.238 5479
	=34 mm 90413 of water	1.542 8768
	=34 kg 90413 per m²	1.542 8768
1.00 lb per sq in	=0 kg 0703067 per cm²	8.846 9968−10
	=703 kg 067 per m²	2.846 9968
	=0.068041 atmosphere	8.832 7720−10
	=16.00 oz. per sq in	1.204 1200
	=144.00 lb per sq ft	2.158 3625
	=27.67975 in of water	1.442 1622
	=2.30665 ft of water	0.362 9810
	=2.0359 in of mercury	0.308 7510
	=51 mm 7113 of mercury	1.713 5856

METRIC CONVERSION TABLE—*Continued*

		Logarithm
1.00 lb per sq ft	=4 mm 88241 of water	0.688 6343
	=4 kg 88241 per m^2	0.688 6343
	=0.0069444 lb per sq in	7.841 6175−10
	=0.11111 oz per sq in	9.045 7575−10
	=0.19222 in of water	9.283 7997−10
	=0.0180184 ft of water	8.204 6185−10
	=0.0141380 in of mercury	8.150 3885−10
1 kg 00 per m^3	=0.0624283 lb per cu ft	8.795 3816−10
1 gr 00 per cm^3	=1000 kg 00 per m^3	3.000 0000
	=1 kg 00 per liter	0.000
	=62.4283 lb per cu ft	1.795 3816
	=8.36469 lb per gallon U. S	0.922 4499
	=10.043 lb per gallon British	1.001 7471
	=0.036127 lb per cu in	8.557 8379−10
1.00 lb per cu ft	=16 kg 0184 per m^3	1.204 6184
	=0 kg 01618 per liter	8.204 6184−10
	=0.13368 lb per gallon U. S	9.126 0683−10
	=0.16046 lb per gallon British	9.205 3655−10
	=0.0005787 lb per cu in	6.762 4563−10
	=0.0092592 oz per cu in	7.966 5763−10
1.00 lb per cu in	=27 gr 6797 per cm^3	1.442 1621
	=27 kg 6797 per liter	1.442 1621
	=27,679 kg 744 per m^3	4.442 1621
	=1,728 lb per cu ft	3.237 5437
	=231.00 lb per gallon U. S	2.363 6120
	=277.274 lb per gallon British	2.442 9092

Weight of Air 0° (32° F.) and 760 m/m (29.92 in) mercury.

1 m^3 00 air	=1 kg 2928	0.111 6154
	=2.85069 lb	0.454 9500
1.00 cu ft air	=0.080723 lb	8.906 9975−10
	=0 kg 036615	8.563 6635−10
1 kg 00 air	=0 m^3 77336	9.888 3840−10
	=22.3109 cu ft	1.436 3365
1.00 lb air	=12.3880 cu ft	1.093 0025
	=0 m^3 3508	9.545 9500−10
1.00 gr mol air	=0 kg 02886	8.460 3096−10
	=0.06363 lb	8.803 6442−10
1.00 oz mol air	=0 kg 8172	9.912 3517−10
	=1.8017 lb	0.255 6917
	=28.827 oz	1.459 8117

METRIC CONVERSION TABLE—*Continued*

		Logarithm
1.00 calorie	=3.96832 B.t.u.	0.598 6065
1.00 B.t.u.	=0.2519959 calorie	9.401 3935−10
1.00 calorie per m^3	=0.11238 B.t.u. per cu ft	9.050 6540−10
	=0.02232 calorie per gr mol	8.348 6942−10
	=2.50812 B.t.u. per oz mol	0.399 3482
1.00 B.t.u. per cu ft	=8.8991 calories per m^3	0.949 3460
	=0.1984 calorie per gr mol	9.297 6568−10
	=22.32 B.t.u. per oz mol	1.348 6942
1.00 calorie per gr mol	=44.803 calories per m^3	1.651 3058
	=5.0379 B.t.u. per cu ft	0.702 3532
	=112.472 B.t.u. per oz mol	2.051 0447
1.00 B.t.u. per oz mol	=0.0448 B.t.u. per cu ft	8.651 3058−10
	=0.1593 calorie per m^3	9.202 1352−10
	=0.00889 calorie per gr mol	7.948 9553−10
1.00 calorie per kg	=1.8 B.t.u. per lb	0.255 2725
1.00 B.t.u. per lb	=0.5555 calorie per kg	9.744 7275−10

CONVERSION OF METRIC FORMULAS TO ENGLISH UNITS

Many readers will prefer to use the English units of measurement and Fahrenheit temperatures. This may readily be done in many of the computations. In some of the formulas, however, the temperature enters as a direct factor. These formulas may be transposed to English units by the use of a conversion factor, but the Centigrade temperatures must be retained, as no means has been found to use Fahrenheit temperatures.

P. 11. 1 cu ft of air = 0.0807 lb. 1 cu ft of waste gases at 0° and 760 mm or 32° F and 29.92 in weighs from 0.0805 to 0.0830 lb per cubic foot.

P. 13. When H is expressed in feet, δ will be expressed in pounds per square foot, when the specific weights are expressed in pounds per cubic foot. Pounds per square foot may be converted to inches of water column by multiplying by the factor 0.19245.

P. 27. The formula, $Q=\kappa_1\kappa_2\omega\sqrt{2gh}$, is the same in English units, the coefficients κ_1 and κ_2 are practically the same for gases and fluids flowing at high velocities from thin or sharp-edge orifices, when the area of the orifice is small. The larger the orifice the less the relative value of these coefficients affects the flow or the nearer the value of the coefficients approaches unity.

P. 32. $\left.\begin{array}{l}\text{1 kg 00 per m}^2\\ \text{1 mm of water}\end{array}\right\} = \left\{\begin{array}{l}\text{0.03937 in of water}\\ \text{0.204817 lb per sq ft}\\ \text{0.001422 lb per sq in}\end{array}\right.$

1 atmosphere = 760 mm of mercury
= 29.92 in of mercury
= 1 kg 0333 per cm^2
= 33.9007 ft of water
= 14.697 lb per sq in

P. 41. When dimensions are in feet, the formula of Yesmann is

$$h_t=0.6731A\sqrt[3]{\frac{Q_t^2}{B^2\cdot t}}. \qquad (E)$$

or as given on p. 256.

The values of B on page 41 are: 1 m 00 = 3.28 ft.
2 m 00 = 6.56 ft.
5 m 00 = 16.40 ft.

The values of h_t = 0.30 = 0.984 ft.
0.50 = 1.64 ft.
0.75 = 2.46 ft.
1.00 = 3.28 ft.

P. 52. For English units these formulas read as follows:

$$h_t=0.6731A\sqrt[3]{\frac{Q_t^2}{B^2\cdot t}},$$

$$Q_t=1.811B\sqrt{\frac{h_t^3\cdot t}{A^3}}.$$

$$Q_t=\omega\cdot v=B\cdot h_t\cdot v, \quad B\cdot h_t\cdot v=1.811B\sqrt{\frac{h_t^3\cdot t}{A^3}}.$$

$$v=1.811\sqrt{\frac{h_t\cdot t}{A^3}}.$$

P. 56. This formula is the same as given on p. 52.

P. 76. The volume of gases given off by the combustion of the coal specified on this page will be:

	Cubic Feet per kg	Cubic Feet per lb
Theoretical air	319	145
1.25 times theoretical air supply	396	180
1.50 times theoretical air supply	473	215
1.75 times theoretical air supply	551	250
2.00 times theoretical air supply	627	285

APPENDIX VII

DESIGN OF OPEN-HEARTH FURNACES

By A. D. WILLIAMS

In designing an open-hearth furnace the first point to be settled is the size of the hearth. Molten metal weighs 430 lb per cubic feet, so that 5.23 cu ft of bath will be required per ton of metal capacity. The depth of bath permissible depends upon the work to be done. With a shallow bath the reactions will be completed faster than in a deep bath. The boil will be more violent and the depth allowed for the molten cinder will be greater. Furnaces of the mixer type, used in the Talbot and duplex process, are much deeper than those used in the regular process.

In the pig and ore process the boil will increase the volume of the bath to from 2 to 2.5 volumes. In the scrap process the bath may boil to 1.5 to 1.7 its original volume. The manner of handling the cinder, whether it is retained in the furnace until the heat is tapped or is to be drawn off by a cinder skimming notch, affects the depth. In the early furnaces room had to be provided for all of the cinder, and overflows were not infrequent. One advantage of the Campbell tilting furnace was the facility it offered for running off a portion of a heavy cinder. Later furnaces were built with a cinder notch which permitted draining any excess amount of cinder.

Table 1 gives the approximate hearth area in square feet required for bath depths ranging from 1 to 3 ft. It was computed by the formula:

$$3\frac{V}{d}=A,$$

in which V = volume of molten metal in charge;
d = depth of bath, metal only;
A = area of hearth.

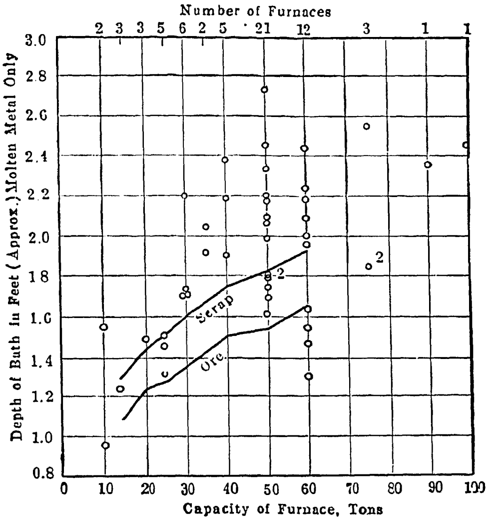


FIG. 138.—Graphical Comparison of the Approximate Depth of Molten Metal in Bath of various Open-hearth Furnaces.

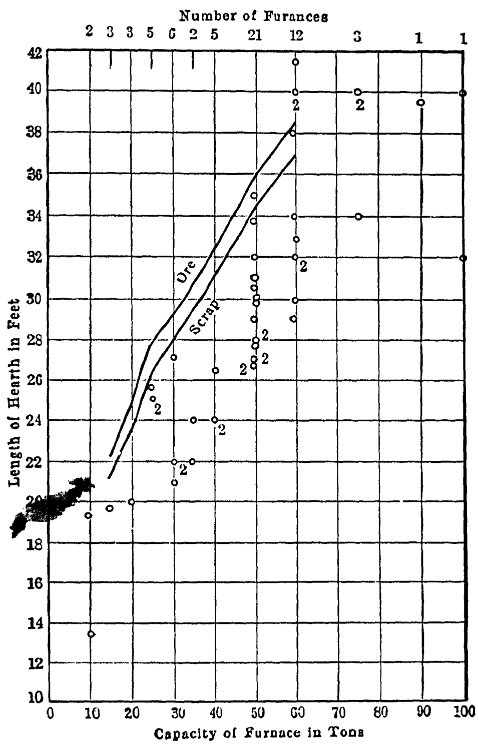


FIG. 139.—Graphical Comparison of the Length of Hearth in Various Open-hearth Furnaces.

While it is necessary to provide additional depth to cover the depth of cinder and the boiling of the charge, additional area is not required for these purposes.

An old rule of thumb made the length of the furnace twice its width. This was all very well with small furnaces, although a longer hearth gives better combustion conditions. The temperature of the metal in the bath is between 1525° and 1600° C. The cinder will be slightly hotter than the metal, the difference ranging from a few degrees just previous to tapping, to 100° or more.

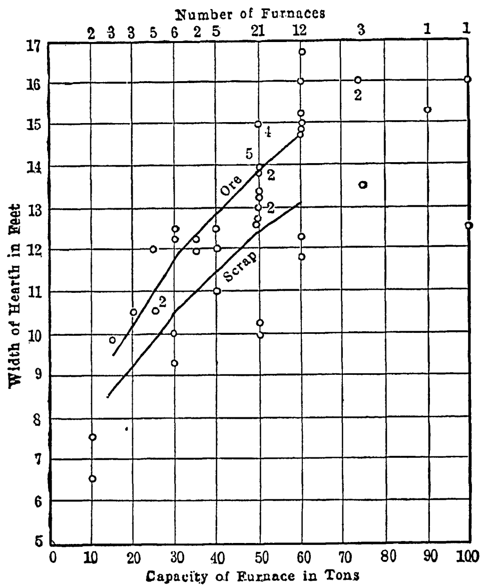


FIG. 140.—Graphical Comparison of the Width of Hearth in Various Open-hearth Furnaces.

The gases leaving the hearth will range between 1600° and 1700°. Told states that the gases should remain in the hearth 2 seconds; Groume-Grjimailo, that the temperature drop through the hearth should be assumed as 200° C. per second. The theoretical flame temperature of a very good producer gas with both the air and the gas preheated to 1000° and 40 per cent excess air will be between 2100° and 2140°.

These data would seem consistent. It requires between one and two seconds for the reaction of combustion to be completed as the large jets of air and gas from the ports must be given time to mix. With too short a furnace the flame will extend beyond the hearth. With too long a furnace the gases of combustion will tend to drop below the bath temperature.

There are three limitations on the width of the furnace:

1. The distance material can be thrown through the door by expert manipulation of a shovel or a dolomite gun.
2. The width the flame can be spread.
3. The limitation due to the strength of the roof.

The first controls patching and making bottom. Furnaces have been built to 16.75 ft wide. This is rather close to the limit.

Table 2 gives the hearth dimensions of a number of furnaces in service. It shows a considerable diversity in the ideas regarding this portion of the furnace. Table 3 gives the minimum and maximum values from Table 2.

TABLE 1

HEARTH AREA OF OPEN-HEARTH FURNACES FOR VARIOUS DEPTHS OF MOLTEN METAL

Capacity of Furnace, Tons	Cu Ft of Molten Metal	Depth of Bath of Molten Metal, In.								
		12	15	18	21	24	27	30	33	36
		(Hearth area in square feet)								
15	78.45	235	188	157	135	118				
20	104.60	314	252	212	180	157	139			
25	130.75	392	314	262	225	196	174			
30	156.90	471	376	314	270	235	209	189		
35	183.05	549	440	356	316	275	244	220	201	
40	209.20	628	504	420	360	314	278	256	229	209
50	261.50	785	629	525	450	392	350	325	286	262
60	313.80	941	755	629	542	471	418	377	343	314
75	392.25	1177	940	785	674	588	523	470	429	392
90	470.70	1412	1130	945	812	706	628	567	515	471
100	523.00	1569	1255	1050	900	785	697	629	573	523

Allow from 3 to 6 in for cinder and boil.

Formula: $3\frac{V}{d}=A$. In which V = volume of molten metal in charge; d = depth of bath; metal only; A = area of hearth required. Weight of molten metal, 430 lb per cubic foot; 5.23 cu ft per ton.

TABLE 2

HEARTH DATA OF OPEN-HEARTH FURNACES FROM PRACTICE

Tons	Reference Number	Length, Ft, L	Width, Ft, W	Area, Sq Ft A	$\frac{L}{W}$	$\frac{A}{\text{Tons}}$	$3\frac{V}{A}$
10	01–A	19.25	6.50	163.6	2.96	16.36	0.95
10	02–F	13.50	7.54	101.0	1.82	10.12	1.59
15	01–GS	21.32	8.52	181.0	2.50	12.06	1.31
15	02–GO	22.37	9.57	215.0	2.33	14.35	1.10
15	03–B	19.67	9.83	191.7	2.00	12.78	1.23
20	01–GS	23.45	9.18	215.3	2.56	10.75	1.46
20	02–GO	24.61	10.33	254.0	2.39	12.70	1.24
20	03–A	20.00	10.50	210.0	1.92	10.50	1.50
25	01–GS	26.25	9.84	258.4	2.67	10.30	1.52
25	02–GO	27.87	11.07	304.6	2.53	12.20	1.29
25	03–A	25.00	10.50	262.5	2.38	10.50	1.49
25	04–A	25.00	12.00	300.0	2.08	12.00	1.31
25	05–A	25.63	10.50	268.0	2.44	10.70	1.46
30	01–GS	27.89	10.50	290.6	2.63	9.6	1.62
30	02–GO	29.20	11.81	344.5	2.48	11.4	1.36
30	03–B	21.00	9.30	195.0	2.26	6.5	2.41
30	04–A	22.00	12.5	275.0	1.77	9.17	1.71
30	05–A	21.88	12.25	267.9	1.77	8.93	1.76
30	06–B	27.00	10.00	270.0	2.70	9.0	1.75
35	01–A	24.00	12.00	288.0	2.00	8.20	1.92
35	02–A	21.87	12.25	267.0	1.79	7.62	2.06
40	01–GS	31.00	11.48	355.0	2.70	8.85	1.77
40	02–GO	32.41	12.89	417.7	2.52	10.50	1.51
40	03–F	24.00	11.00	264.0	2.20	6.60	2.38
40	04–A	26.33	12.50	329.0	2.11	8.23	1.90
40	05–A	24.00	12.00	288.0	2.00	7.20	2.18
50	01–GS	34.45	12.46	430.6	2.77	8.6	1.83
50	02–GO	35.96	13.98	502.7	2.58	10.0	1.56
50	03–A	32.00	10.00	320.0	3.20	6.4	2.46
50	04–A	27.00	13.9	376.0	1.96	7.5	2.08
50	05–A	29.75	15.00	446.0	1.99	8.90	1.76
50	06–A	32.00	14.00	448.0	2.28	8.96	1.75
50	07–A	30.00	15.00	450.0	2.00	9.00	1.74
50	08–A	30.50	13.00	396.5	2.36	7.92	1.98
50	09–A	31.00	15.00	465.0	2.08	9.30	1.68
50	10–A	28.00	12.75	357.0	2.20	7.14	2.20
50	11–A	31.00	14.00	434.0	2.22	8.58	1.81
50	12–A	35.00	14.00	490.0	2.50	9.80	1.60
50	13–A	26.75	13.40	361.1	2.00	7.20	2.18
50	14–A	27.67	13.30	368.7	2.08	7.38	2.12
50	15–A	26.75	12.58	336.5	2.12	6.73	2.33

TABLE 2—*Continued*

HEARTH DATA OF OPEN-HEARTH FURNACES FROM PRACTICE

Tons	Reference Number	Length, Ft, L	Width, Ft, W	Area, Sq Ft, A	$\frac{L}{W}$	$\frac{A}{\text{Tons}}$	$3\frac{V}{A}$
50	16–A	32.00	14.00	448.0	2.28	8.96	1.75
50	17–A	27.00	13.90	375.8	1.95	7.50	2.08
50	18–A	29.00	15.00	435.0	1.95	8.70	1.80
50	19–A	32.00	14.00	448.0	2.28	8.96	1.75
50	20–A	33.67	13.00	438.0	2.59	8.76	1.79
50	21–F	28.00	10.25	287.0	2.73	5.74	2.74
60	01–GS	36.75	13.12	484.4	2.80	8.1	1.95
60	02–GO	38.39	14.76	566.2	2.60	9.3	1.68
60	03–A	30.00	15.00	450.0	2.00	7.5	2.09
60	04–A	38.00	15.00	470.0	2.54	9.5	1.65
60	05–A	43.00	16.75	720.0	2.55	12.0	1.30
60	06–A	40.00	16.00	640.0	2.50	10.67	1.47
60	07–A	34.40	12.30	423.0	2.80	7.05	2.23
60	08–B	32.75	11.80	387.0	2.77	6.45	2.43
60	09–A	32.00	14.75	472.0	2.18	7.67	2.00
60	10–A	29.00	14.92	432.0	1.94	7.20	2.18
60	11–A	40.00	15.25	610.0	2.63	10.16	1.54
60	12–A	32.00	15.00	480.0	2.13	8.00	1.96
75	01–A	40.00	16.00	640.0	2.50	8.53	1.84
75	02–A	34.00	13.50	460.0	2.53	6.14	2.56
75	03–A	40.00	16.00	640.0	2.50	8.53	1.84
90	01–A	39.50	15.25	600.0	2.56	6.67	2.36
100	01–A	40.00	16.00	640.0	2.50	6.40	2.45
100	02–F	32.00	12.50	400.0	2.56	4.00	3.92
150	01–A	41.00	15.00	615.0	2.74	4.06	3.83
200	01–A	42.00	15.75	665.0	2.66	3.32	
200	02–A	40.00	16.00	640.0	2.50	3.2	
200	03–B	47.20	13.00	616.0	3.64	3.08	
200	04–A	40.00	13.00	520.0	3.07	2.60	
300	01–B	47.20	14.75	695.0	3.19	2.32	

A = American. F = foreign. B = Boche (German-Hungarian, etc.)
GS = Groume-Grjimailo scrap process.
GO = Groume-Grjimailo ore process.

TABLE 3

MAXIMUM AND MINIMUM VALUES FROM TABLE 2

Capacity, Tons	Length, Ft		Width, Ft		Area, Sq Ft		$\frac{L}{W}$		$\frac{A}{\text{Ton}}$		$3\frac{V}{A}$	
	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.
10	13.50	19.25	6.50	7.54	101.0	163.6	1.82	2.93	10.12	16.36	0.96	1.55
15	19.67	22.37	8.52	9.83	181.0	215.0	2.00	2.50	12.06	14.35	1.10	1.31
20	20.00	24.61	9.18	10.50	210.0	254.0	1.92	2.56	10.50	12.70	1.24	1.50
25	25.00	27.87	9.84	12.00	262.5	304.6	2.08	2.67	10.30	12.20	1.29	1.52
30	21.00	29.20	9.30	12.50	195.0	344.5	1.77	2.70	8.93	11.40	1.36	2.41
35	21.87	24.00	12.00	12.25	267.0	288.0	1.79	2.00	7.62	8.20	1.92	2.06
40	24.00	32.41	11.00	12.89	264.0	417.7	2.00	2.70	6.60	10.50	1.51	2.38
50	26.75	35.96	10.00	15.00	287.0	502.7	1.95	3.20	5.74	10.00	1.56	2.74
60	29.00	43.00	11.80	16.75	387.0	720.0	1.94	2.80	6.45	12.00	1.30	2.43
75	34.00	40.00	13.50	16.00	460.0	640.0	2.50	2.53	6.14	8.53	1.84	2.56
90	...	39.50	...	15.25	...	600.0	..	2.56	...	6.67	..	2.36

TABLE 4

OPEN-HEARTH FURNACE HEARTH AND HEATING CHAMBER PROPORTIONS, ACCORDING TO GROUME-GRJIMAILO

Capacity, Tons	Scrap Process				Ore Process				Heating Chamber, Cu Ft
	Length, Ft	Width, Ft	Area, Sq Ft	Height Above Bath, Ft	Length, Ft	Width, Ft	Area, Sq Ft	Height Above Bath, Ft.	
15	21.32	8.52	181.7	5.25	22.57	9.57	215.3	4.43	954
20	23.45	9.18	215.3	5.91	24.61	10.33	254.0	5.00	1271
25	26.25	9.84	258.4	6.15	27.85	11.07	304.6	5.21	1581
30	27.89	10.50	290.6	6.56	29.20	11.81	344.5	5.53	1907
40	31.00	11.48	355.2	7.56	32.41	12.89	417.7	6.09	2543
50	34.45	12.46	430.6	7.15	35.96	13.98	502.7	6.32	3178
60	36.75	13.12	484.4	7.87	38.39	14.76	566.2	6.73	3814

The diagrams, Figs. 138, 139, 140 and 141 show, respectively, the approximate depth of bath in feet; the length of the hearth in feet; the width of the hearth in feet and the area of the hearth in square feet. Two lines have been plotted on these diagrams, showing the dimensions of the furnaces given in Table 4, which was abstracted from Professor Groume-Grjimailo's paper, in the *Journal de la Société Russe de Métallurgie*. This table was given in *The Iron Age* of Dec. 26, 1917.

Fig. 142 shows the profile of one side of the hearth suggested in this paper for a 30-ton furnace. These furnace hearths show a certain consistency in their design, a larger hearth and a shallower bath being suggested for the ore process than for the scrap process.

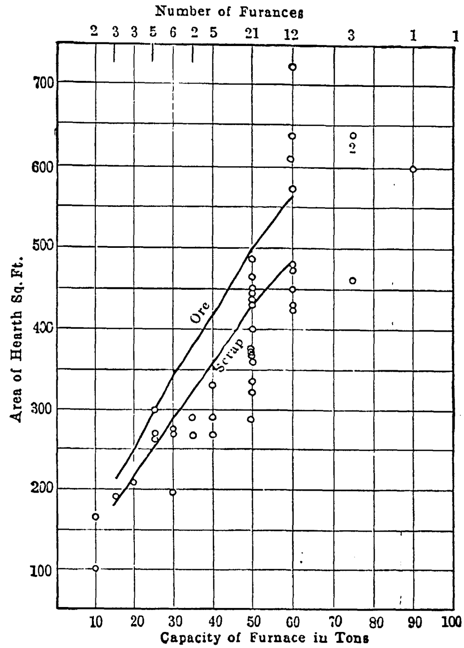


FIG. 141.—Graphical Comparison of the Hearth Area of Various Open-hearth Furnaces.

One feature of the open-hearth and similar furnaces that has caused sorrow and tribulation is that the bottom of the furnace forms a valley below the level of the port sills. One of the basic principles of furnace design is that the flame must lick the hearth or bottom of the furnace. The gas pressure in the heating chamber must be equal to the atmospheric pressure. This last means that a nice balance must be maintained between the volume of gas and air entering the chamber and the gases removed from the chamber. When the pressure in the furnace is permitted to drop below the atmospheric pressure cold air will tend to flow into the furnace around the doors and through all cracks and openings. This chills the bath and causes excessive oxidization to occur.

It is well known that when patching bottom it is necessary to keep the doors closed as much as possible in order to avoid chills, which are likely to develop into cracks and leaks in the bottom. Many beautiful octahedral steel crystals have been discovered in tearing out old furnace bottoms.

When the pressure in the furnace gets higher than the atmosphere a sharp sting of flame is developed and fuel must be burned to maintain it. Some melters and many heaters keep the doors of their furnaces decorated with a halo of flame. In some cases it is necessary to maintain this sting owing to the defective design of the furnace. There is no question whatever that this method of working will prevent cold air being drawn into the furnace. It is "the easiest way"; the company pays for the excess fuel, and the melter or heater has plenty of time to sit down and "watch 'er burn." In foreign plants the technical control of the furnaces is more closely maintained than in this country because fuel is expensive.

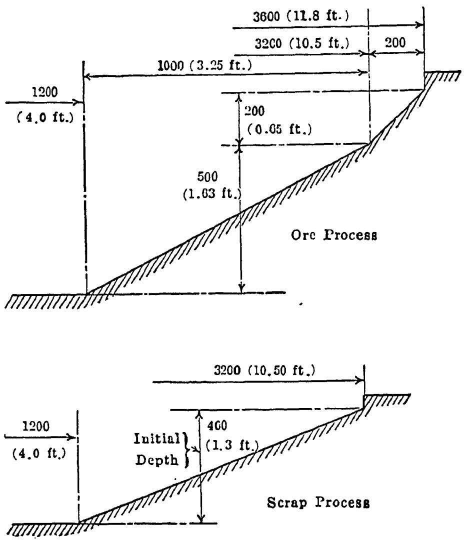


FIG. 142.—Sections Showing Profile of Hearth Slope According to Professor Groume-Grjimailo.

In order to sinter the bottom of the furnace in place it is necessary to direct the jet of flame so that it will lick the hearth. In order to do this the air and gas ports must be given a suitable inclination toward the hearth and the velocity of the flame must be sufficient to carry it down to the bottom. The velocity which will be impressed upon the jet of air entering the furnace will be fixed by the height from the bottom of the regenerators to the port and by the area of the port and the flues leading to it. In the case of the gas a slightly greater head is available as it enters the regenerator under pressure from the producer. The gas pressure can be and usually is increased toward the end of a campaign when the regenerators are partially blocked.

The inclination of the ports is fixed when the heads are built. It frequently happens that these heads are changed several times before they work in a satisfactory manner. In fact, a number of furnace drawings which have been published show heads which must have been altered considerably in order to make the bottom.

In the open-hearth furnace the ports are just below the roof,

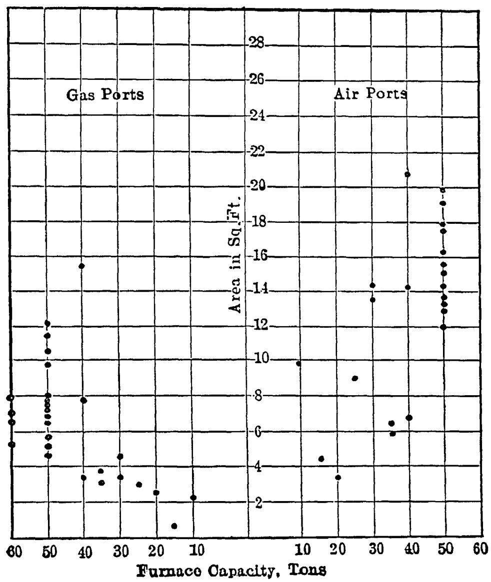


FIG. 143.—Graphical Comparison of the Port Areas of Various Open-hearth Furnaces.

and the hearth forms a pocket below the level of the port and door sills and above the level of the tapping hole. This pocket will become filled with the coolest gases in the heating chamber. In order to sweep these gases out of the pocket and permit the sintering of the bottom of the furnace in place it is necessary to utilize the jet of flame issuing from the ports. This jet must describe a parabola having a middle ordinate equal to the vertical distance from the tapping hole to the port sills. In the earliest types of open-hearth furnace it was endeavored to attain this result by depressing the roof of the furnace. The serious objections to this poor construction arose from its tending to strangle the furnace during the heat and from the extremely short life of the roof. Later experience showed that the flame might be jetted into this pocket by the direction of the ports and the velocity of the gas and air. In European furnaces these velocities range between 12 and 18 m (40 to 60 ft) per second, while in American furnaces the velocity is as high as 50 m per second.

TABLE 5

OPEN-HEARTH FURNACES

Reference Number, Ton	Square Feet, Port Areas		Reference Number, Ton	Square Feet, Port Areas	
	Gas	Air		Gas	Air
10–01–A	2.43	10.12	50–04–A	10.50	18.0
			50–05–A	7.66	19.2
15–03–B	1.07	4.68	50–07–A	5.75	13.8
			50–09–A	7.50	13.7
20–03–A	2.80	3.50	50–10–A	6.60	13.1
			50–11–A	4.92	14.53
25–04–A	3.25	9.00	50–12–A	11.59	17.72
			50–13–A	12.50	18.0
30–04–A	4.75	13.75	50–14–A	6.80	15.38
30–05–A	3.70	14.60	50–15–A	9.85	16.14
			50–16–A	5.62	17.80
35–01–A	3.4	6.40	50–16a–A	7.00	20.00
35–01a–A	3.75	5.98	50–17–A	10.50	12.33
			50–18–A	8.00	16.42
40–05–A	15.70	21.00	50–19–A	7.40	13.30
40–06–A	3.5	7.00			
40–04–A	8.00	14.25	60–05–A	6.75	29.31
			60–06–A	8.00	25.74
			60–13–A	7.74	25.37

A number of different arrangements of the ports of the Siemens furnace have been devised and used with more or less success. The early Siemens furnaces were designed for the use of producer gas, which required preheating. Later these furnaces were used with various manufactured and natural gases and fuel oils, while latterly pulverized coal has been employed. These various fuels require furnace modifications, mainly in the ports and heads, as only the air supply is preheated.

Pulverized coal is only suited for use in furnaces where the ash carried into the furnace with the fuel will not be objectionable. One trouble with early open-hearth or Siemens furnaces was caused by the dirt carried over into the regenerators. This was particularly the case when the chambers were located immediately below the furnace and the uptakes rose directly from the chamber arch. In later designs the chambers were placed below the charging platform and the uptakes were carried up from a cinder pocket or slag chamber. This reduced, but did not eliminate, the cinder trouble.

The carrying power of a flowing stream varies as the sixth power of its velocity; that is, when the velocity is doubled, the mass of the particle which the stream can carry increases sixty-four times. The inertia of these larger particles tends to carry them into any eddies where the stream changes direction, but the finer particles will be carried farther. The ports must be inclined and the velocity of the flame must be sufficient to allow the bottom to be made. This also tends to direct the flame on the surface of the bath and the higher the impinging velocity the greater the tendency to pick up cinder, etc., which will be thrown up during the boil.

Possibly the best illustration of the action of the jet of flame impinging upon the top of the bath may be obtained by observing the action of a stream of water from a nozzle impinging upon a flat plate. When the stream is directed at right angles to the plate there will be a circular flare or film of water traveling outward at a high velocity and a short distance out a tumultuous ring of eddying water eight or ten times as thick as the film it surrounds. The distance out to this ring will depend upon the velocity of the stream. When the jet strikes the plate at an angle it will form a triangular high-velocity film breaking up into a turbulent eddy at the base or side furthest from the apex where the stream strikes the plate. The distance from the stream to the eddy will be affected by two factors, the velocity of the stream and its angle of incidence. If a water surface is substituted for the flat plate the action is complicated by the fact that the jet displaces a certain amount of the surface water

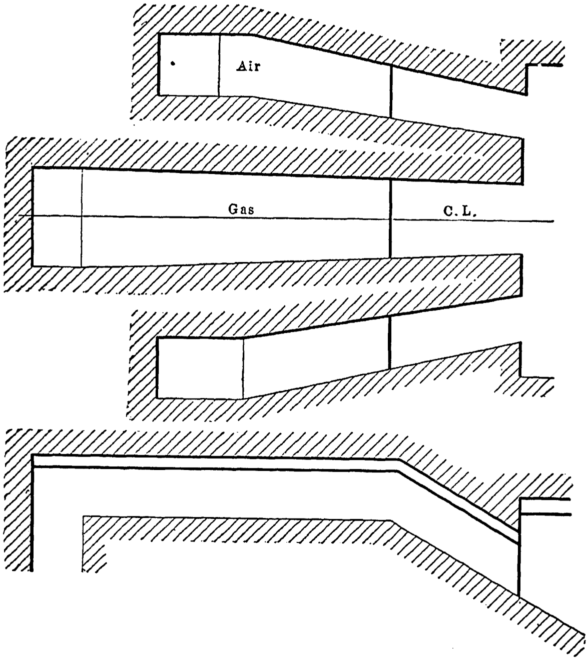


FIG. 144.—Arrangement of Ports of a 25-Ton Open-hearth Furnace at Lyswa, Russia.

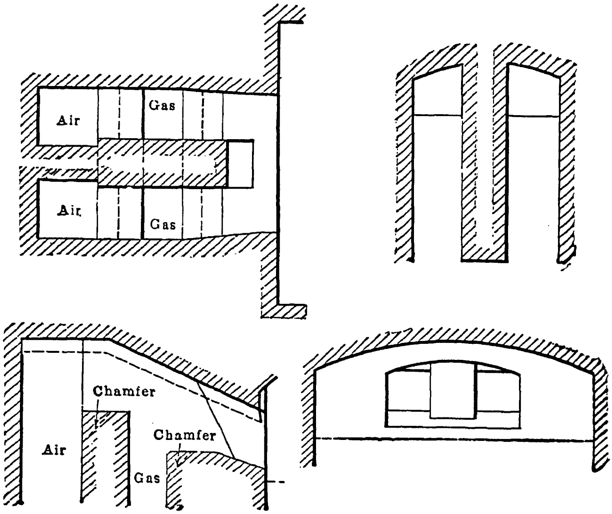


FIG. 145.—Arrangement of Ports of a 50-Ton Furnace at Homestead. This furnace was fired with natural gas but was arranged for producer gas firing.

and the size of the turbulent eddy is considerably increased. If a second stream of water be directed so that it impinges upon the first stream just before the first stream impinges on the plate the condition of the formation of the flames in the furnace will be approximated. The main difference between the action of the two streams of water and that of the streams of air and gas forming the flame will result from the fact that as the reaction of combustion takes place there is a great increase of temperature which

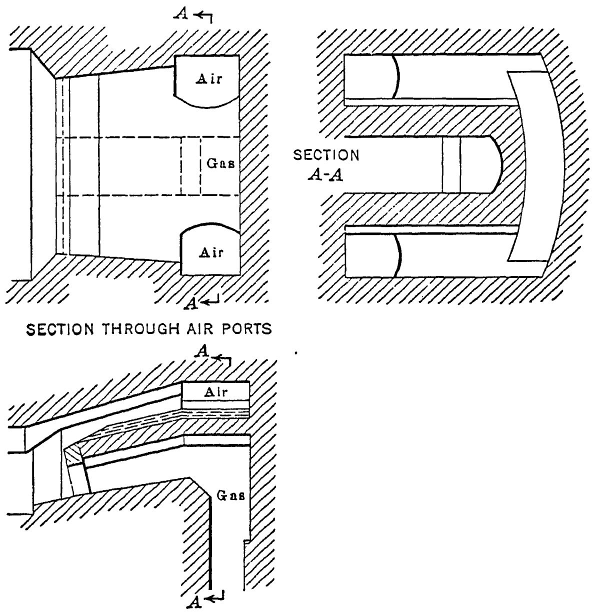


FIG. 146.—Arrangement of Ports on Many American Furnaces. Low air velocity and high gas velocity.

makes the volume of the flame approximately double that of the reacting gases.

A study of the effect of the velocities of the two streams upon their mixing will reveal many interesting facts, particularly if the streams are colored so as to supply a contrast while they remain distinct and a third color by their complete mixture. The degree of the mixing at various points will be revealed by the various tints formed as one or the other color predominates.

The function of the ports is to bring the combustible and the comburent to a point where they will combine in the flame. In the early types of Siemens furnaces there were usually five ports side by side, two for gas and three for air. Figs. 144, 145 and 146 show later designs of ports which were used respectively

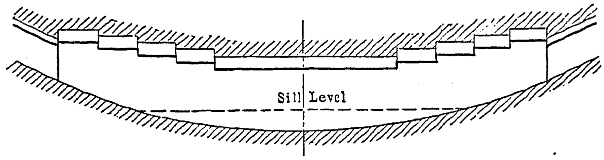


FIG. 147.—Dropped Roof of Early Furnace Design.

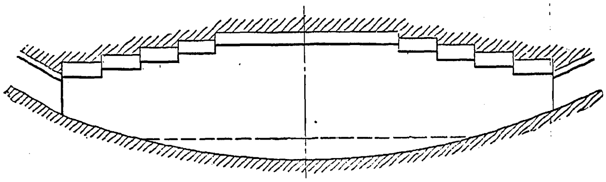


FIG. 148.—Heating Chamber with Raised Roof.

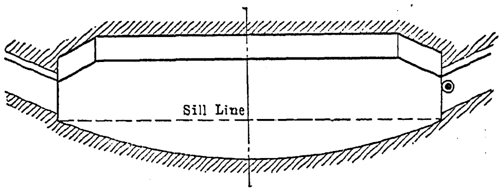


FIG. 149.—Heating Chamber with Raised Straight Roof.

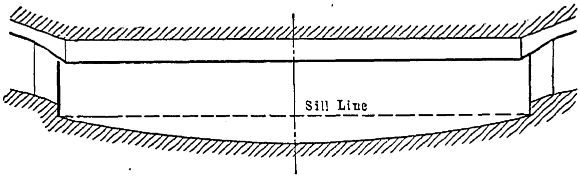


FIG. 150.—Heating Chamber with Straight Roof.

with the heating chambers shown in Figs. 148, 149 and 150. Figs. 144 and 148 were used at the Lyswa Works (Oural), Russia, while Figs. 145, 149, 146 and 150 were used in American furnaces, the last being the later design. A noticeable feature of the Russian furnace is that the bath occupies about 0.60 of the length of the heating chamber. This gives a space at each end of the heating chamber for the formation of the flame. The velocities of the gas and air entering the heating chamber are approximately the same. The air ports are located on each side of the gas port. This furnace works hot and has a good output.

The ports shown in Fig. 145 were designed for use with natural gas, which was jetted into the port at right angles with the stream of preheated air and close to the bottom of the port. These ports were also intended to permit the use of producer gas in case of the failure of the natural gas supply, there being two regenerator chambers at each end of the furnace. With natural gas both chambers were used for air. With producer gas the uptakes nearest the heating chamber were for gas and those further back were for air. With this design of port the stream of gas impinges upon the stream counter to the air current. This would tend to form a mixing eddy at point of junction.

The port arrangement shown in Fig. 146 is that used in many American furnaces. In this design the air velocity is comparatively low, while the gas velocity is from four to ten times the air velocity. One of the reasons that has been advanced for this port arrangement is that it forms a blanket of air between the flame and the roof, reduces the wear on the roof and protects the bath from the oxidizing effect of the air. This design of port gives an extremely long flame. The flame is forced away from the port and the ends of the heating chamber work alternately hot and cold. The introduction of this design of port resulted in an increase in the length of the heating chamber in order to prevent the flame passing beyond the heating chamber. Then the gas velocity was increased to force the flame to the end of the chamber.

Fig. 143 is a diagram in which have been plotted the areas of the gas and air ports as tabulated in Table 5. The wide difference in the ideas of port areas is well illustrated.

With oil, pulverized coal, coke-oven and natural gas, the fuel is piped to the furnace and used without preheating. The fuel is introduced at the end bulkhead or through the sides of the heads. Blue water gas has been used in some foreign furnaces. As this gas contains practically no hydrocarbons it may be preheated. A few attempts have been made to utilize blast-furnace gas in the open-hearth. It may be done by preheating to a higher temperature than is usual with the ordinary mixed producer gas.

Regardless of whether the fuel is used cold or preheated it must be brought into contact with the preheated air so that the flame formed will permit the sintering of the bottom, and the heads of the furnace must be designed to obtain this result.

One of the reasons why blue water gas and blast-furnace gas have not been considered on their heating possibilities is that they burn with a non-luminous flame, it being considered that to obtain high temperatures in the open-hearth and reverberatory furnace a flame with a so-called high radiating effect is necessary. By this is meant a luminous flame. It is well known that the transmission of heat by radiation varies as the difference between the fourth powers of the temperatures of the radiating and recipient surfaces and a coefficient varying from unity for the ideal black body to a very small fraction of unity for a polished surface. Conduction varies with the temperature difference. Convection depends upon the temperature head or difference in temperature and the flow of the fluid.

In the early designs of Siemens furnaces the roof was depressed from each end to the center, as shown in Fig. 147. It was supposed that this type of roof enhanced the heating effect by forcing the flame into contact with the bath and assisted in the sintering of the bottom. This type of roof had a short life, as it had a tendency to burn out, and it was frequently damaged when charging the furnace. Its worst defect was that it choked the furnace. Later designers were obsessed by the radiant-heat idea and this resulted in the forms of roof shown in Figs. 148 and 149. It was soon found, however, that this type of roof resulted in an increased fuel consumption, and the straight roof (Fig. 150) is now used. Fig. 151 shows a form of skewback designed to prevent the wall expansion from interfering with the roof. Twelve-inch roofs are widely used and many American furnaces employ the Orth roof which permits the use of a repair block when the intermediate shapes burn out. It is possible that a roof with cooling ribs spaced closer together than in the Orth roof would be more satisfactory.

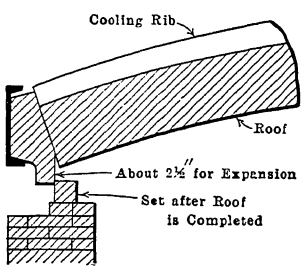


FIG. 151.—Skewback Construction. Designed to permit wall to expand at top.

It is a basic principle of furnace design that satisfactory results cannot be obtained unless the flame licks the sole of the furnace. It is likewise well known that bottom cannot be made in an open-hearth furnace unless the flame can drop down far enough to sinter the bottom in place, no matter how luminous the flame may be. One of the hottest flames is that of the oxy-acetylene torch. Acetylene burned in air gives a very luminous flame, but when this flame is supplied with oxygen it becomes a blue non-luminous flame. The luminous acetylene flame does not emit an excessive amount of radiant heat to any recipient surface. It is rather noticeable that when the oxy-acetylene torch is used in welding it is necessary for the flame to impinge upon the work, and that the work has a tendency to become luminous, while the flame itself has a very slight luminosity, and that at the tip only. The mixed producer gas which is used in numerous Siemens furnaces derives its heating value mainly from carbon monoxide and hydrogen. These two combustibles are the main constituents of blue water gas. In fact, blue water gas bears a fairly close resemblance to a good mixed producer gas from which the nitrogen has been removed.

Radiant heat from a luminous flame may also be considered from another viewpoint. The roof, side walls and bath are at a high temperature and emit a certain amount of light, depending upon their temperature and emissivity. The flame is sufficiently transparent, when the furnace is at a high temperature, to permit the opposite wall and the slag surface to be seen. The red and yellow portion of the flame, which emits the most visible light, is not transparent and is at a lower temperature than the blue transparent portion. A smoky flame is caused by the presence of soot or unignited carbon. Soot may be caused by the dissociation of carbon monoxide when this gas is chilled by impinging upon cold metal.

Stratification is frequently observed in the flame. Cool gases tend to collect below hotter gases. The hottest gases tend to rise to the roof where they are cooled. The coldest gases would have a tendency to collect on the surface of the bath, but the jet of flame from the ports tends to sweep them away. Convection currents are rarely appreciated at their true value. A temperature difference of 1° C. is sufficient to impress a velocity of 0.268 m (0.88 ft) per second and this velocity will increase as the square root of the difference in temperature. This tends to give an angular direction to currents.

The open-hearth furnace works very close to the yield point of the refractories, but it is only recently that water-cooling has been adopted for these furnaces, although it has been used for years in the blast furnace. Water-cooling adds to the life of the brickwork by increasing the thermal gradient through the wall and removing the heat. It adds little if anything to the fuel consumption and increases the life of certain portions of the furnace, thereby reducing the amount of time the furnace is down for local repairs. This means increasing the output. Water-cooled doors and frames were used a number of years ago, but the extended use of cooling devices is rather recent.

In early designs of furnaces the regenerator chambers were under the furnace and the uptakes rose direct to the ports. As a result the upper portion of the checkerwork blocked up rapidly and its life was reduced. The first cinder pockets were small chambers parallel with the regenerator chamber designed to distribute the gases to the checkerwork by a number of small ports and their functioning as cinder pockets was accidental. The way the cinder lodged in them showed the advantage of increasing their size, and ample space for this purpose became available when the checkerwork was removed from below the furnace and placed under the charging platform. The main fault with many of these pockets is their lack of depth. Removable cinder pockets have been devised.

Regenerative methods of firing or the preheating of the incoming gases and the air for combustion have made it possible to attain high furnace temperatures. The numerous furnaces utilizing the heat of the products of combustion for the preheating of the gas and air supply are a grand memorial to the Siemens brothers.

In the designing of any furnace with regenerators it is important to have sufficient regenerator capacity and equally important to avoid overdoing the matter. The products of combustion leave the heating chamber of open-hearth furnaces at a temperature of about 1700°. The incoming air enters the valve at 10° and the temperature of the producer gas in the main will range from 500° to 600°. Coke oven gas, natural gas or any gaseous fuel containing hydrocarbons cannot be passed through a regenerator without a considerable loss in its heating power due to the dissociation of the hydrocarbons. Furnaces fired with these gases, as well as those fired with oils or tars, are designed to preheat the air supply only, but it is usually desirable to design the regenerators for these furnaces in such a manner that producer gas may be substituted without extensive and costly alterations.

VOLUME OF CHECKER BRICK

There seems to be a considerable diversity of opinion regarding the amount of checker brick which should be used. Table 6 gives the data covering the volume of the checker brick in a number of furnaces. These data have been plotted in Fig. 152

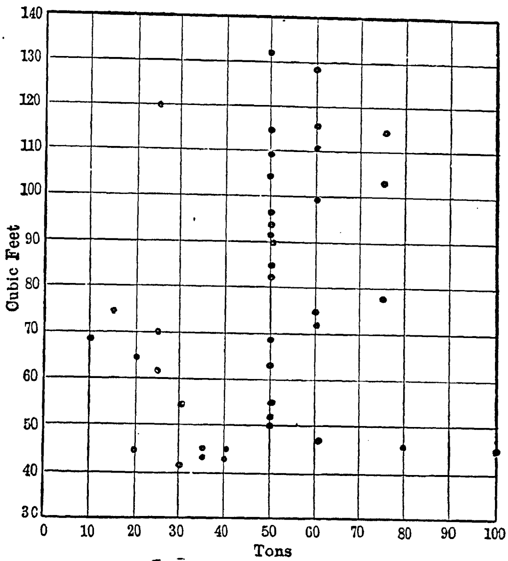


FIG. 152.—Graphical Comparison of the Volume of Air Plus Gas Checker Brickwork in Cubic Feet per Ton of Nominal Capacity for Furnaces of Various Capacities.

which compares the total amount of the air and gas checkerwork per ton of nominal capacity. Table 7 gives the maximum and the minimum volumes of checkerwork.

In all data of this kind there are two uncertain factors. The nominal and the actual capacity of an open-hearth furnace are different. It happens quite frequently that the actual capacity is 10 to 40 per cent in excess of the nominal capacity. When crane and ladle capacity is available this excess can be turned into output; otherwise it may be troublesome. The other uncertain factor is in the checker brickwork itself, the way it is set and its material, the space allowed for the passage of gas and air and thickness of the brick.

A number of more or less empirical rules have been proposed for arriving at the volume of checker or regenerator capacity. A. Consett makes the total regenerator volume 2 cu m per tonne (70 cu ft), and makes the air chamber 10 per cent larger than the gas chamber, for furnaces designed to make four melts per twenty-four hours. In *Notes et Formules* it is stated that "for slow furnaces the empirical relation between the volume of the regenerator and the charge of the furnace in tonnes is 0.8 cu m (28 cu ft) to 1.3 cu m (46 cu ft) per tonne; and for fast working furnaces from 53 to 64 cu ft per tonne." H. H. Campbell says from 50 to 100 cu ft per tonne of melting capacity. Gruner says 50 to 70 kg of checker brick per kilogram of coal burned per reversal. Told suggests 6 cu m or 2850 kg of checker brick per cubic meter of air per second per 100° C. rise in temperature. Another suggestion is the provision of 50 sq m (537.5 sq ft, of checker surface per tonne (metric) of coal per twenty-four hours.

In Euchene's investigations on the carbonizing of coal he developed a formula for the heat-storage capacity of checker brickwork which may be stated as follows:

$$Q_0{}^t = 0.20t + 0.000062t^2,$$

in which $Q_0{}^t$ = kilogram calories per kilogram of checker brick between zero and t degrees C.; and in which t = temperature to which the brickwork is raised, degrees C.

Breslauer cites the regenerator data of a number of furnaces, giving the number of cubic meters of regenerator per ton of steel per hour.

His figures are given for a number of different plants as follows:

Cubic Meters of Regenerator per Ton of Steel per Hour

Dowailis	14.60	Graz	15.50	Haleside	16.75
Krupp, large	18.32	Landore	20.00	Pateg	20.60
Krupp, 3 ton	36.00	Terre Noire	30.00	Swedish	36.60
		Borsigwerke	44.00	Steelton, Pa.	37.45

The wide variety of rules for the proportioning of the regenerators affords an ample opportunity for choice, and in part explains the variation in volume of checkerwork used by different designers. Empirical rules, basing the proportioning of a part by a direct ratio with the capacity of the furnace are very easy to apply, and require very little thought in their use. In addition they save time, as it only takes a few minutes to arrive at the regenerator volume required. Unfortunately, this is not the case when logical methods are employed.

HORIZONTAL OR VERTICAL PASSAGES

Regenerators may be built with the gas passages arranged horizontally or vertically. Where head room is limited a horizontal pass regenerator, at first sight, appears to possess certain advantages, but a consideration of the behavior of heated gases when cooling and of cold gases when being heated, will show that a horizontal regenerator or recuperator introduces a considerable amount of friction in the path of the gases, necessitating an initial pressure for introducing the air and gas and a strong suction to remove the waste gases.

At Steelton, Pa., it was found necessary to use a blower with horizontal regenerators. The writer recently had occasion to investigate the action of some horizontal regenerators. It was found that the products of combustion were flowing mainly through the upper passes and their heat was conveyed to the lower passes by conduction through the brickwork. The air supply flowed through the entire height of the checker. The frictional resistance was excessive. These regenerators, however, contained an enormous weight of brick for the amount of work to be done by them, and for that reason gave very little difficulty in operation.

A basic principle of regenerator design is: the gas passages should be vertical; the cooling gas should pass downward through the checkerwork; the gas being heated should pass upward through the checkerwork. When the gases circulate in this manner they have a tendency to subdivide themselves between the different passes proportionally to the local heating and cooling of the brickwork, with the result that the cooling and heating of the gases will be practically uniform. A few years ago A. E. Maccoun made a series of temperature observations on a Cowper hot-blast stove at

TABLE

VOLUME OF CHECKER BRICK

Reference Number, Tons	Gas Checkers					
	Length, Ft	Width, Ft	Area, Sq Ft	Volume, Cu Ft	Volume, Cu Ft per Ton	Height, Ft
10–01–A	.....	.....	.....	297	29.7	.....
15–03–B	.....	.....	.....	504	33.6	.....
20–03–A	.....	.....	.....	517	25.8	.....
20–04–A	7.00	4.00	28.0	240	21.0	.....
25–03–A	18.00	4.50	81.0	689	27.6	8.5
25–04–A	.....	.....	.....	732	29.3	.....
25–05–A	.....	.....	.....	.....	.....	.....
30–05–A	.....	.....	.....	532	17.7	.....
30–04–A	.....	.....	.....	703	23.4	.....
35–01–A	12.60	5.58	66.5	665	19.0	10.00
35–02–A	12.17	6.33	77.0	656	18.7	8.50
40–03–F	15.00	5.29	79.0	734	18.4	9.25
40–07–E	8.00	7.00	56.0	840	21.0	15.00
50–04–A*	22.00	6.00	132.0	1040	20.8	7.83
50–17–A	.....	.....	.....	930	18.6	.....
50–20–A	25.50	5.50	140.0	1400	28.0	10.00
50–16–A†	22.00	7.92	174.0	1740	34.8	10.00
50–21–F	12.25	9.50	116.0	1070	21.4	9.25
50–07–A	18.25	10.17	185.0	2590	51.8	14.00
50–22–A	18.00	8.17	144.0	2088	41.8	14.50
50–23–A	17.00	10.08	172.0	2093	41.8	12.17
50–09–A	.....	.....	.....	3300	66.0	.....
50–10–A	.....	.....	.....	896	17.9	.....
50–11–A	.....	.....	.....	1936	38.7	.....
50–13–A	.....	.....	.....	1235	24.7	.....
50–16–A†	.....	.....	.....	1912	38.2	.....
50–04–A*	.....	.....	.....	1188	23.7	.....
50–05–A	.....	.....	.....	1928	38.6	.....
50–18–A	.....	.....	.....	1980	39.6	.....
50–19–A	.....	.....	.....	2156	43.1	.....
60–11–A	31.25	7.92	247.0	2510	42.0	10.17
60–13–A	.....	.....	.....	2027	33.8	.....
60–03–A	.....	.....	.....	1953	32.5	.....
60–04–A	.....	.....	.....	1616	26.9	.....
60–05–A	.....	.....	.....	2417	40.3	.....
60–06–A	.....	.....	.....	2515	41.9	.....
60–07–A	24.00	8.4	202.0	2780	46.3	13.83
60–08–B	23.00	8.4	193.0	2630	43.8	13.83
60–14–E	12.00	7.0	84.0	1260	21.0	15.00
75–03–A	.....	.....	.....	2515	33.5	.....
75–02–A	16.00	9.33	149.0	3278	43.7	22.00
75–01–A	.....	.....	.....	3334	44.5	.....
80–01–E	12.50	9.00	112.0	1680	21.0	15.00
100–03–E	14.00	10.00	140.0	2100	21.0	15.00

* Original and rebuilt.

6

ASCERTAINED FROM PRACTICE

Air Checkers						
Length, Ft	Width, Ft	Area, Sq Ft	Volume, Cu Ft	Volume, Cu Ft per Ton	Height, Ft	Air+Gas, Cu Ft per Ton
.....	.....	.....	391	39.1	.....	68.8
.....	.....	.....	616	41.0	.....	74.6
.....	.....	.....	780	39.0	.....	64.8
7.00	4.50	31.5	472	23.6	15.00	44.6
18.00	6.00	108.0	818	32.7	8.50	60.3
.....	.....	.....	1020	40.8	.....	70.1
.....	.....	.....	3000	120.0	.....	120.0
.....	.....	.....	708	23.6	.....	41.3
.....	.....	.....	937	31.2	.....	54.6
12.00	7.58	90.5	905	25.9	10.00	44.9
12.17	8.33	101.5	860	24.6	8.50	43.3
15.00	7.00	105.0	970	24.3	9.25	42.7
8.00	7.83	63.0	944	23.6	15.00	44.6
22.00	10.00	220.0	1735	34.7	7.83	55.5
.....	.....	.....	1590	31.8	.....	50.4
25.50	8.00	204.0	2040	40.8	10.00	68.8
22.00	10.83	238.0	2380	47.6	10.00	82.4
12.25	9.50	116.0	1070	31.4	9.25	62.8
18.25	12.00	218.0	3130	62.6	14.00	114.4
18.00	12.17	218.0	3160	63.2	14.50	105.0
17.00	12.08	203.0	2470	49.4	12.17	91.2
.....	.....	.....	3300	66.0	.....	132.0
.....	.....	.....	1681	33.6	.....	51.5
.....	.....	.....	2904	58.1	.....	96.8
.....	.....	.....	1900	38.0	.....	62.7
.....	.....	.....	2621	52.4	.....	90.6
.....	.....	.....	1980	39.6	.....	62.3
.....	.....	.....	2310	46.2	.....	84.8
.....	.....	.....	2693	53.8	.....	93.4
.....	.....	.....	3310	66.2	.....	109.3
31.25	10.83	340.0	3460	57.7	10.17	99.7
.....	.....	.....	5662	94.4	.....	128.2
.....	.....	.....	2393	39.8	.....	72.3
.....	.....	.....	2835	47.2	.....	74.1
.....	.....	.....	4198	70.0	.....	110.3
.....	.....	.....	3443	57.4	.....	99.3
24.00	12.60	302.0	4160	69.4	13.83	115.7
23.00	12.60	290.0	4000	66.7	13.83	110.5
12.00	7.83	94.5	1416	23.6	15.00	46.6
.....	.....	.....	3442	44.5	.....	78.0
16.00	12.50	200.0	4400	58.7	22.00	102.4
.....	.....	.....	5265	70.2	.....	114.7
12.60	10.00	126.0	1888	24.8	15.00	45.8
14.00	11.25	157.5	2360	23.6	15.00	44.6

† Original and rebuilt.

the Edgar Thomson Works (American Iron & Steel Institute meeting, May 28, 1915), which showed that the checker openings close to the shell took more gas when heating and less air on blast.

In his work, *Fours à Flamme*, Professor Groume-Grjimailo gives a mathematical demonstration of the reason why the portion of the checkerwork close to the wall, losing a considerable amount of heat by radiation, exerts a greater cooling effect on the hot gases of combustion flowing downward and tends to concentrate their flow through this portion of the checker, while the air, which is being heated, tends to seek the central and for that reason more slowly cooled portion of the checkerwork. It is interesting to note that Professor Groume-Grjimailo cites the Cowper stove, as follows: "For example, the checker openings of the Cowper hot-blast stove located nearest to the shell lose a great deal of heat by radiation; they therefore exercise a stronger cooling effect upon the current of gas flowing through them and by reason of this the velocity of the descending current of gas is increased, since if $t_2 < t_1$ then $v_2 > v_1$."

Convection currents in gases and air, due to small temperature differences, are by no means inconsiderable. A temperature difference of 1° C. is sufficient to impress a vertical velocity of 0.268 m (0.88 ft) per second on a gas. This velocity will be increased directly as the square root of the difference in temperature. Convection currents will act to carry the cooled gas or the heated gas or air away from the heating or cooling surface. These currents will exist in any chamber, regardless of whether a current of gas is circulating through the chamber or not. When a hot gas is giving up its heat to a surface the convection currents will be downward, while when a cool gas is absorbing heat from a surface the convection currents will be upward. It naturally follows that the working current or circulation of the air or gases should be in the same direction as the convection currents, as in this case they will tend to reduce the friction loss. When the convection currents are in the opposite direction to the working current they form recirculating loops which entail a direct loss of heat capacity in the checkerwork.

PERIOD BETWEEN REVERSALS

The period between reversals has a direct bearing upon the design of the checker in two important particulars: the heating and cooling time determines the weight of checker brick required and the thickness of the brick composing it. There is a limit to the quantity of heat which may be absorbed and given out by the brickwork. The practical limit is reached when the entire mass of brickwork is raised to a temperature at which heating gases pass through the checker with a very slight or no drop in temperature. The economical limit is reached when the heating gases leave the checkerwork at such a temperature that during the period when the checker is giving up heat the brickwork temperature does not drop below the initial temperature of the incoming gases plus the temperature differential necessary for heat transfer.

The most important variable, the period between reversals, is beyond the control of the designer. The furnaces are operated by two or three shifts of men. Each shift usually has certain definite portions of the furnace to keep in repair and will so operate the furnace, when it is possible so to do, in a manner to favor their section and throw the burden of repair work upon the other shifts. This leads to irregular working. Sidney Cornell (*Chemical and Metallurgical Engineering,* May, 1913) cites the case of a 60-ton furnace. There were 509 reversals in a week. The average time period between reversals was 10 minutes. The longest period was 70 minutes, the shortest 2 minutes. Consecutive periods differed widely in their duration. The following time periods in minutes were scaled from a diagram he presented:

R 15 R 25 R 20 R 15 R 16 R 23 R 24 R 24 R 34 R 22 R 53 R 12 R 9.

(R stands for reversal.)

Quick reversals are necessary at certain stages of the operation, but it is certainly desirable that the work should be divided, as nearly equally as possible, between the two ends of the furnace. When the time factor depends entirely upon the human element it is absolutely impossible to avoid considerable irregularity in operation. At the same time melt variations will prevent automatically timed reversals, except during certain stages of the heat.

However, reversing machines are available which will operate the valves in a predetermined sequence and time the reversals regularly. These reversing machines also permit the operator to reverse the furnace at any time he sees fit. Machines of this type have been installed at a number of by-product coke-oven plants, but there seems to be a tendency upon the part of the " heaters " to cut out the automatic timing device. It is hardly necessary to state that a certain amount of passive or impassive opposition might be expected to the use of a machine of this kind, a species of sabotage which is extremely difficult to overcome. Under ordinary working conditions there should be two to eight reversals per hour, and it is necessary to proportion the checkerwork to suit the longest time period.

TABLE 7

MAXIMUM AND MINIMUM VOLUME OF CHECKER BRICK ASCERTAINED FROM PRACTICE

Capacity, Tons	Gas		Air		Air+Gas	
	Minimum	Maximum	Minimum	Maximum	Minimum	Maximum
10	29.7	....	39.1	....	68.8	
15	33.6	....	41.0	....	74.6	
20	21.0	25.8	23.6	39.0	44.6	64.8
25	27.6	29.3	32.7	40.8	60.3	70.1
30	17.7	23.4	23.6	31.2	41.3	54.6
35	18.7	19.0	24.6	25.9	43.3	44.9
40	18.4	21.0	23.6	24.3	42.7	44.6
50	17.9	66.0	31.4	66.0	50.4	132.0
60	21.0	46.3	23.6	94.4	44.6	128.2
75	33.5	44.5	44.5	70.2	78.0	114.7
80	21.0	....	24.8	....	45.8	
100	21.0	....	23.6	....	44.6	

The time factor has a definite bearing on the thickness of the checker brick. This is shown by the curves of Fig. 153. These curves were plotted for a fire-clay brick. For a silica brick or brick of other material the co-ordinates would be different, but the curves would be similar to those shown. These curves show the rise in temperature at the center line of a brick in per cent of the rise in temperature at the surface, when both faces of the brick are heated during various time intervals. In these curves the ordinates are given per cent values to permit their application to any initial temperature. These curves are computed by a formula developed from Fourier's conduction equation. They show that the rate of temperature rise at the center of the brick will vary according to the square of the thickness of the brick. These curves apply equally to the cooling period. A curve showing the complete heating and cooling cycle will resemble the hysteresis loop, which shows the heating effect of cyclic magnetic changes upon an iron core.

The firebrick makers in this country list a special checker brick, 10.5×4.5×4.5 in (265×115×115 mm), and a checker brick 2.9×2.75×2.75 in (107×70×70 mm), and some designers use 9-in straights, which give a 2.5-in (63-mm) wall. With a 30-minute period between reversals, the temperature change on the central plane of these bricks may be approximated as follows:

Assuming that the initial temperature throughout the bricks is practically uniform, and the surface is heated through any temperature range for a period of 30 minutes. At the end of this period the temperature of the central plane will have increased to 59 per cent, 94 per cent and 96.5 per cent of the surface temperature, respectively, according to Fig. 153. So that the cooling cycle starts with an initial drop in temperature on the central plane of 41, 6 and 3.5 per cent, respectively.

Fig. 153 shows that the period of time required for these drops will be, respectively, 21, 3 and 2 minutes. During this period the portion of the brick between the center and the surface will be transmitting heat both toward the center and the surface of the brick.

In other words, the thinner the checker brick the higher its heat-storage capacity as compared with the volume it occupies, the greater the amount of heating surface for the given weight of brickwork and the smaller the heat-storage capacity per unit of surface. When the checker brick are too thin the heat gradient from top to bottom of the checker becomes a curve instead of a straight line. The brick, instead of working on the sloping portion of the curve (Fig. 153) work over on to the flat upper portion of the curve.

A great many of the formulas covering the heat transfer from one substance to another contain a factor which covers the velocity of flow of the gas or liquid which is absorbing or giving up heat to the stationary surface. For some reason or other there seems to be an impression that the higher the velocity of flow the better. Undoubtedly there is a certain velocity of flow that will result in the maximum heat transfer per unit of heating surface, and in the case of properly designed heat transfer elements, be they regenerators, recuperators, steam boilers, etc., it is extremely probable that the natural convection currents will

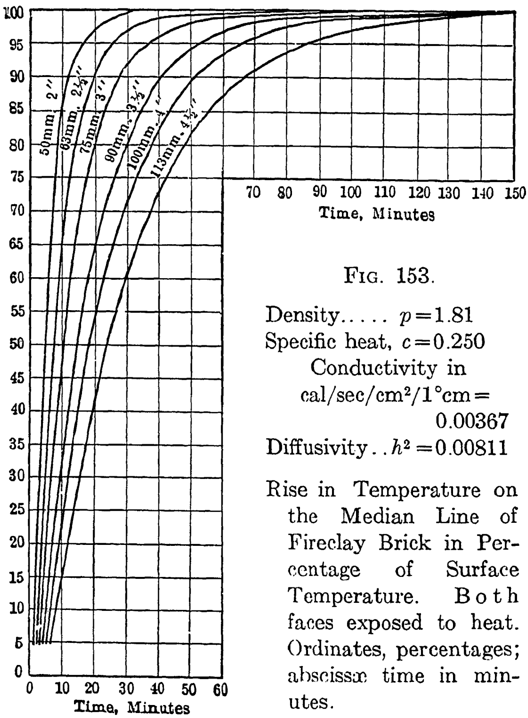


FIG. 153.

Density..... $p = 1.81$
Specific heat, $c = 0.250$
Conductivity in cal/sec/cm²/1°cm = 0.00367
Diffusivity.. $h^2 = 0.00811$

Rise in Temperature on the Median Line of Fireclay Brick in Percentage of Surface Temperature. Both faces exposed to heat. Ordinates, percentages; abscissæ time in minutes.

assume this velocity themselves. Higher velocities, as well as lower velocities, will reduce the rate of heat transfer. The higher velocities may be obtained by forcing the gas or liquid past the surface, when the velocity is too low, that is, when the heated gases are not carried away from the chamber above the regenerator, or the cooled gases cannot flow away from below the checkerwork, recirculation will take place in the form of convection eddy currents.

Concerning forced circulation, it is easy to conceive that a current of hot gas may pass a finite surface at such a high velocity that its temperature drop or loss of heat may be extremely small. In this case the rate of heat transfer will be very low per unit of surface. On the other hand, it may be conceived that the gas passes the heating surface so slowly that it loses practically all of its heat, and has a large drop in temperature. In both of these cases an inordinate amount of heating surface will be necessary. With the high velocity a very great length of surface will be required. With the low velocity an extremely wide surface will be required. Probably the best analogy to these conditions is supplied by the electrical circuit. A series circuit may have such a high resistance that its power transmission value will approximate zero. A multiple circuit may have such a low resistance

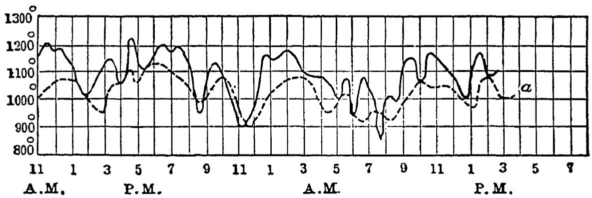


FIG. 154.—Temperature Head or Differential between the Hot Gases in a Fire Box and a Point 1 In. from the Surface of the Wall Exposed to Heat.

that it forms a short circuit and its power transmission value will approximate zero.

The heat differential or temperature head necessary for the heat transfer is a factor concerning which there is considerable disagreement. There is the film theory of high surface resistance due to layers of gas adhering to the surface. Such data as are available indicate that a temperature drop of 300° or more is necessary for gas to gas heat transfer. The curves of Fig. 154 were copied from a pamphlet, "The Flow of Heat Through Furnace Walls" (Bulletin No. 8, U. S. Bureau of Mines) by Ray and Kreisinger. The full line shows the fluctuation in the firebox temperature of the experimental furnace as indicated by a thermo couple and the dotted line gives the temperatures of a couple embedded in the brickwork 1 in from the surface exposed to heat. It will be noticed that the brick temperatures lag behind the furnace temperatures and, due to the heat-storage capacity of the brick, a smoother curve is obtained. The temperature gradient between the two curves includes the resistance of 1 in

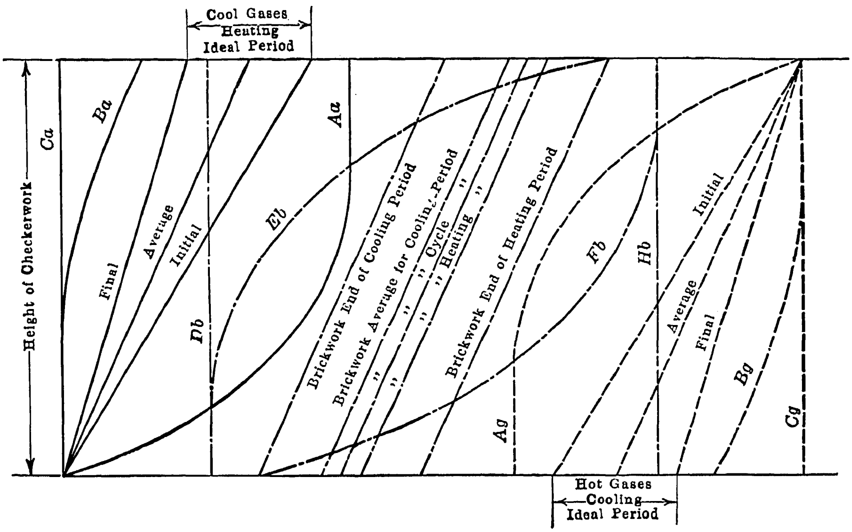


FIG. 155.—Curves Showing the Temperature Changes in the Hot Gases Cooling, the Cool Gases Heating and the Checker Brick.

These curves showing these temperature changes during the normal or ideal period or cycle are indicated by the legends. The abnormal cycle or the changes due to overheating and overcooling are marked with reference letters. The latter condition is the one that occurs when a cold checker is heated in starting.

- *Ag* cooling curve of hot gases passing through a cold checker;
- *Bg* cooling curve of hot gases passing through a checker with too long a heating period;
- *Cg* cooling curve of hot gases passing through a checker after it has absorbed its maximum amount of heat;
- *Aa* heating curve of cold gases passing through an overheated checker;
- *Ba* heating curve of cold gases passing through a checker with too long a cooling period;
- *Ca* cold gases passing through a cold checker;
- *Db* cold checker prior to heating;
- *Eb* checker initial heating curve. The upper portion after becoming heated transmits heat to lower portion by conduction;
- *Fb* checker overheated in upper portion. Heat is transmitted by conduction to raise temperature of lower portion;
- *Hb* overheated checker prior to cooling.

of brick as well as any surface resistance which may exist, and this difference rarely exceeds 150°. This would seem to indicate that a heat differential of less than 300° might be obtainable under good working conditions from gas to gas.

Blocking up of the checkerwork by dust deposits occurs progressively, and a number of schemes have been tried for increasing the time between clean-outs. This dust collects upon the upper portion of the checkerwork and causes more trouble in the air checker than it does in the gas, for the reason that the gas usually enters the regenerators under pressure and this pressure can be increased to overcome increased resistance. The air, however, enters the checker under atmospheric pressure, and its flow is induced by the chimney effect caused by the heat it absorbs acting through a vertical height equal to the distance from the bottom of the checkerwork to the port. It naturally follows that increased resistance will have a considerable effect in reducing the air supply, and it is frequently necessary to put a fan on the air supply after 400 to 500 heats. Running the gas through the air checkers has been the subject of a patent, the idea being based upon the reducing action of the gas, but this deposit contains very little iron ore when there are adequate cinder pockets. The composition of this deposit at the Donetz-Jurjewka works was:

Composition of Deposit in Checkers

	Gas	Air
Additions and limestone, per cent	26.0	37.0
Furnace lining, per cent	0.8	1.5
Cinder, per cent	28.2	16.0
Silica brick, per cent	45.0	45.5

Frank Orth has designed a checker in which the size of the pass increases in four steps, being widest at the top and narrowest at the bottom. While G. L. Danforth, Jr., arranges the top of the checkerwork in a series of blocks surrounded by channels about 6 in wide and about 2 ft deep. Another arrangement, by Danforth, provides a number of pits in the top of the checkerwork. The number, perimeter and depth of the pits giving an entry area equal to the entry area at the top of the checkerwork. In both cases the purpose is to provide space for the deposit and at the same time prevent it from blocking up the checkerwork. The dust deposit will in time attain a thickness of 1.0 to 2.0 in at the top of the checker and its effect in increasing the resistance

TABLE 8

TEMPERATURES, PRESSURES, VELOCITY, TIME DATA FROM VARIOUS SOURCES

	E. Juon at Donetz-Jurjewka Works	
	Pressures in mm of of Water	Temperatures, Average, Degrees C
Gas in main	+15	600
Gas in flues under regenerators	+ 2 to +11	650 to 750
Average	+5	700
Gas at top of regenerators	+4	1100 to 1300
		1200
Gas at top of uptake	+7	1250
	+ 5 to +10	1200 to 1300
Air under regenerators	−1	550
		400 to 700
Air at top of regenerators	∓0	1225
		1100 to 1350
Air at top of uptake	+3	1300
		1250 to 1400
Hearth chamber, top and bottom	∓0	1800
Waste gases top of uptake—Gas	−3	1700
	− 2 to − 3	
—Air	+1.5	1700
Top of gas regenerator	−14 to −15	1600
Top of air regenerator	−14 to −15	1600
Below gas regenerator	−19	750
Below air regenerator	−18	700
At base of stack	−33	600

	Temperatures According to		
	Told	Le Chatelier	Harbison Walker
Gas in flues	400	720	720
Leaving regenerator	850	1200	1200
Average temperature in regenerator	625		
Air at reversing valve	30		
In flues leading to regenerator	270		
Leaving regenerators	950	1000	1000
Average temperature in regenerator	610		
Waste gases entering regenerators	1600		
After reversing valve	300		

TABLE 8—*Continued*

	From Told	
	Velocity, m per Second	Time in Seconds
Air and gas passing through the valves	3.50	
Entering regenerators	3.00	
Passing through regenerators, less than	1.00	
Air in regenerator chamber	....	5.00
Checkerwork	....	3.00
Gas in regenerator chamber	....	4.00
Checkerwork	....	3.00
Air and gas passing through checkerwork, less than	2.00	
Air and gas passing through chamber over checker	0.50	
Air and gas issuing from the ports	8.00	
Gases in the hearth	....	2.00

through the regenerator may be gaged by the following record of the gas pressure below and above the checkerwork published by E. Juon.

MEASURE OF RESISTANCE TO FLOW THROUGH CHECKERS

Heat Number	Pressure in mm of Water		Heat Number	Pressure in mm of Water	
	Below	Above		Below	Above
156–159	+6	+5	600–602	+18	0
160–164	+5	+5	603–606	+18	+1
165–168	+4	+4	607–610	+17	0
169–172	+4	+4	611–613	+16	0
173–176	+5	+0	614–617	+17	0
			618–621	+16	+1

The dust deposit is difficult to cope with, as much of it is extremely fine. It is very difficult to reduce the velocity of the gas current and throw this dust down in the limited space available, but a considerable amount of it will be deposited in a well-designed cinder pocket.

Taking up the design computations for a 50-tonne open-hearth furnace regenerator, the following assumptions are made:

Melting capacity of furnace........................	50 tons
Number of heats made per week..................	15
Average time per heat, charge to tap...............	8 hours 20 minutes
Coal consumption per ton of steel..................	300 kg (660 lb)
Average number of reversals per hour...............	3
Secondary air supply, per cent of theoretical.........	140

GASES OF COMBUSTION ON BURNING 100 VOLUMES OF HAW PRODUCER GAS

	Vol.	Calories	O_2	CO_2	H_2O	N_2
H_2	11.26× 58.2=	655	5.63	.....	11.26	22.52
CH_4	3.24×195.2=	632	6.48	3.24	6.48	25.92
CO	29.65× 68.2=	2022	14.83	29.65	.....	59.32
O_2	0.19	....	.....	.....	.....	.....
CO_2	1.25	....	.....	1.25	.....	.....
N_2	50.69	....	.....	.....	.....	.....
H_2O	3.72	....	.....	.....	3.72	.....
	100.00	3309	26.94	34.14	21.46	158.45
			0.19	.....	.....	0.76
			26.75	34.14	21.46	157.69

COMBUSTION OF 100 MOLECULAR VOLUMES OF GAS

Air			Products of Combustion in Molecular Volumes					
Supply, Per Cent	Molecular Volumes	Excess Air	O_2	CO_2	H_2O	Excess N_2	N_2	Total
100	133.75	0.00	0.00	34.14	21.46	0.00	157.69	213.29
120	160.50	26.75	5.35	34.14	21.46	21.40	179.09	240.09
140	187.25	53.50	10.70	34.14	21.46	42.80	200.49	266.79
180	240.75	107.00	21.40	34.14	21.46	85.60	243.29	320.29

These values may be evaluated as cubic feet, cubic meters, etc.

A portion of this air is required for the oxidization of certain elements in the charge and additions.

Total coal consumption = 300×50 = 15,000 kg (for the 500 minutes). The average coal consumption will be 30 kg per minute, or 0.5 kg per second. The maximum rate of coal consumption will be approximately 1.25 kg per second. This coal gasified in a producer supplies 3.5 cu m of gas per kilogram of coal gasified and this gas, burned with 40 per cent excess air, requires 1.87 volumes of air per 1 volume of gas and the products of combustion will be 2.68 volumes. Fig. 183 shows the heat capacity and calorific intensity curves of the producer gas, the air supply and the waste gases, as computed according to the methods of Mallard and Le Chatelier. These curves are based upon the combustion or burning of 100 molecular volumes or 2.232 cu m of the gas fuel under consideration for use in the furnaces.

		Required for Heating		Available from Products of Combustion	
		Gas Checker	Air Checker	Gas Checker	Air Checker
Temp. top of checkerwork, degrees...		200	1200	1700	1700
Temp. bottom of checker, degrees....		600	300	800	600
Temp. increase or decrease, degrees...		600	900	900	1100
Heat capacity in calories at....	1700°	....	....	4140	4140
(Based upon 100 molecular volumes of the gas burnt from curve of Fig. 183)..	1200°	960	1700		
	800°	....	....	1750	
	600°	450	....	....	1250
	300°	....	480		
Heat in calories per 100 molecular volumes of gas burnt to be given or absorbed from checkerwork........		510	1220	2390*	2890*
Heat in calories per cubic meter of gas burnt to be given out by or absorbed by checkerwork..................		228.5	546.6	1071*	1295*

* These values give the total amount of heat available in the products of combustion of 100 molecular volumes or 1 cu m of the gas cooled through the assumed temperature range of either the gas or the air checker, and which must be divided between the two checkers.

By assuming that 2 cu m of gas are burned the proportion of the products of combustion required for the heating of the two checkers may be arrived at:

	Heat Required for		Heat Available Products Combust	
	Gas	Air	Gas Checker	Ch
For gas: 228.5×2	457			
For air: 546.6×2	....	1093.2		
Available in products of combustion..	....	....	1071	12
Total required and available	1550.2		2366	
Per cent of total required	29.48	70.52		
Heat available required for the gas checker = 2366×29.48	....	....	697.5	
Heat available required for the air checker = 2366×70.52	....	....	....	16
Volume of products of combustion required for gas checker, 697.5÷1071, per cent	....	....	65.13	
Volume of products of combustion required for air checker, 1668.5÷1295, per cent	....	....	....	12
Total per cent waste gases	....	....	193.99	
Correction to make 200 per cent	....	....	6.01	
Corrected per cent values	....	....	67.00	13
Reducing these to a 100 per cent basis	....	....	33.50	6
Calories per cubic meter of gas burned	228.5	546.6	359	86
Volumes per cubic meter of gas burned	1.00	1.87	0.9*	

* 2.68 volumes×0.335 and 2.68×0.665.

The surplus heat over that required for the preheating o
gas and air will be

For the gas checker: 359−228.5=130.5 calories
For the air checker: 861−546.6=314.4 calories

This surplus supplies the heat lost by radiation, condu
and convection from the checker or regenerator chamber
should these losses be not sufficient to absorb this amount of
the final temperature of the products of combustion leaving
checkers will automatically increase until the heat carried
into the furnace by the air and gas plus these losses equals

amount of heat given to the checkerwork by the products of combustion.

The maximum quantity of gas required will be

$$3.5 \text{ (cu m per kg)} \times 1.25 \text{(kg coal)} = 4.375 \text{ cu m per second.}$$

The maximum quantity of air required will be

$$4.375 \times 1.87 = 8.182 \text{ cu m per second.}$$

The maximum volume of the products of combustion will be

$$4.375 \times 2.68 = 11.725 \text{ cu m per second.}$$

These volumes are reduced to zero C. and 760 mm of barometric pressure.

The average volumes will be as follows:

of gas: 3.5×0.5(kg of coal) $= 1.75$ cu m per second
of air: $1.75 \times 1.87 = 3.273$ cu m per second
of products of combustion: $1.75 \times 2.68 = 4.69$ cu m per second

	Reversal Period $= t$	
	20 Minutes, 1200 Seconds	30 Minutes, 1800 Seconds
Gas:		
Maximum calories $= 4.375 \times 228.5 \times t =$	1,200,000	1,800,000
Air:		
Maximum calories $= 4.375 \times 546.6 \times t =$	2,871,000	4,306,000
Gas:		
Average calories $= 1.75 \times 228.5 \times t =$	480,000	720,000
Air:		
Average calories $= 1.75 \times 546.6 \times t =$	1,148,000	1,722,000
Allowing for a range of 200° in the checker brick and 90 per cent of its weight as effective, the weight of brick required will be		
For gas:		
20 min.: $1{,}200{,}000 \div (200 \times 0.25 \times 0.90) =$	26,667 kg	
30 min.: $1{,}800{,}000 \div (200 \times 0.25 \times 0.90) =$	....	40,000 kg
For air:		
20 min.: $2{,}871{,}000 \div (200 \times 0.25 \times 0.90) =$	63,800 kg	
30 min.: $4{,}306{,}000 \div (200 \times 0.25 \times 0.90) =$	....	95,680 kg

The brick will weigh about 1800 kg per cubic meter; therefore the volume occupied will be

	Reversal Period 20 Minutes 1200 Seconds	30 Minutes 1800 Seconds
For gas:		
20 min. period, 26,667÷1800, cu m......	14.32	
30 min. period, 40,000÷1800, cu m......	......	22.23
For air:		
20 min. period, 63,800÷1800, cu m......	35.45	
30 min. period, 95,680÷1800, cu m......	......	53.13
Increase in temp. in checkerwork, degrees	Gas	Air
Gas: 1200−600 = ...	600	...
Air: 1200−300 = ...	...	900
Average temp. in checkerwork, degrees:		
Gas: (1200+600)÷2 = ...	900	...
Air: (1200+300)÷2 = ...	...	750
Average time in checkerwork, seconds:		
At 100° increase in temp. per second...	6	9
At 200° increase in temp. per second...	3	4.5
Gas: Aver. volume per sec. at 900°, m^3, $1.75\times(1+at)$ = ...	7.53	
Air: Aver. volume per sec. at 750°, m^3, $3.273\times(1+at)$ = ...	...	12.28
With 200° per second increase in temp., the volume required for the passes will be, cu m.		
Gas: 7.53×3 = ...	22.59	....
Air: 12.28×4.5 = ...	....	55.26

With a temperature rise of 100° per second the volume required for the passes will be double the above.

Fig. 153 shows that with the usual period between reversals there will be no economy in increasing the thickness of the checker brick over 2.5 in (63 mm), which is a standard shape.

The volume occupied by the checker brickwork will be the sum of the brick volume added to the pass volume, and requires no explanation. With a different producer gas and a different air proportion there would be a corresponding change in the amount of heat interchange required and in the volumes required for the brickwork and for the passes.

The height of the checkerwork should be made as great as possible, say, 5 m (16.4 ft) as a minimum. The flues below the checker should be proportioned to permit free flow to the passes at a low velocity. If desired, the height of these flues may be stepped down, proportionally to the distance from the inlet. This, however, introduces complications in the brickwork with comparatively little gain.

The vertical velocity of the gases leaving the checkerwork may be considerable and it is necessary to provide sufficient space above the checker to get rid of the eddies which will be formed. Unless sufficient space is provided these eddies will cause considerable interference with the flow of the gas from the checker chamber into the slag pocket, converting what should be a smooth regular flow into a series of bursts or blow-throughs. These bursts may have entirely different periods in the air and gas chambers and result in considerable irregularity of combustion and waste of fuel.

In any regenerative fired furnace it is impossible to reduce the temperature of the waste gases leaving the regenerator to the temperatures of the incoming air and gas. This results in a considerable loss of heat up the chimney. Frequently one-third of the heat passes uselessly up the stack, and the amount of heat lost is more often in excess of this value than it is less.

The theoretical temperature of the waste gases at the base of the stack is about 300°. The actual temperature at the base of the stack ranges from 600° to 1000°, and rarely runs below 700°. This would seem an ideal opportunity to install a steam boiler and recover a portion of the heat, but until a comparatively few years ago such an installation was not considered, although the installation of boilers on puddling furnaces had been common practice for many years. To-day few open-hearth furnaces are built without waste heat boiler equipment and it is extremely probable that such boilers could be profitably installed in connection with many of the other furnaces.

Considerable notice has been taken of these waste heat installations in the various technical papers, but it has been in the main simply tabulations of the installations as made with very little real information as to the underlying reasons which led to the selection of the particular equipment installed. A noticeable departure from this practice was presented by the paper of Thomas B. Mackenzie before the Iron and Steel Institute, in 1918.

PROVISION FOR EXPLOSIONS

One of the operating difficulties with the open-hearth furnace lies in the fact that explosions of gas are likely to occur whenever the furnace is reversed. This difficulty is also met in all regenerative furnaces. These explosions vary in their intensity from slight puffs to heavy explosions and are due to the gas trapped in the gas regenerator meeting the air that is drawn into the stack flue. When the furnace is connected directly with the stack the puff of the explosion passes up the chimney and is rarely noticed. Similar explosions are not infrequent in blast furnace practice, and experience in that line has demonstrated the absolute necessity of providing explosion doors to relieve explosion pressures, as well as the necessity of making all the flues and settings gas tight and building them with buck stays of sufficient strength to stand the explosion stresses.

When boilers or economizers are connected with the furnace these explosions become of serious import and unless relief valves are provided of sufficient area to prevent excessive rises in pressure the settings will be damaged and numerous cracks will admit cold air, greatly reducing the efficiency of the waste heat installation.

Another cause of unsatisfactory results with waste heat boilers arises from the loss of sensible heat by the gases in passing through the flues. These flues are generally underground and close to the surface, and the ground above the flue is frequently so hot that it remains dry, except when very heavy rains occur. Exactly what the heat loss from this source will be depends upon the construction of the flue, the depth below the surface and the length of the flue. With waste heat utilization it is desirable that the flues should be well insulated and as short and direct as possible.

The waste-gas flues are frequently far from tight, and when waste-heat boilers are installed the air leakage into the flues is much more serious than when they connect directly to the stack. Low temperatures at the bottom of the chimney in most metallurgical high-temperature furnaces should be viewed with suspicion, until checked by an analysis of the waste gases.

LEAKAGE THROUGH VALVES AND DAMPERS

Leaky valves and dampers are another source of trouble when waste-heat boilers are installed. The simple butterfly valve is the oldest form of reversing valve. When carefully made and new these valves are tight, but they do not remain in that condition very long when exposed to hot gases. In modern practice the butterfly valve is rarely used, except for reversing the air, the gas being reversed by valves better designed for the prevention of leakage.

There are a number of valves on the market which have proved more or less successful in operation. Many of these valves have water seals, which prevent leakage as long as the water supply is maintained and the pressure differential between the flues or the flue and the air is less than the seal. All water-seal valves lose their seal during the reversal period, and while this period, when the sealing lip is lifted above the water surface, may be only a fraction of a minute, a certain amount of loss occurs which cannot be prevented. All water-seal valves add perceptibly to the moisture in the hot gases which pass through them.

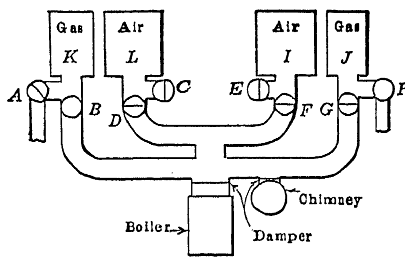


FIG. 156.—An Arrangement of Flues and Valves Used in the United States. Refer to sequence of valve operation.

Fig. 156 shows an arrangement of valves and flues which has been used in the United States. It is rather costly, involving the installation and upkeep of eight valves and two dampers. A method of reversal which experience has shown to be satisfactory with this valve system is as follows: assuming that the air and gas are entering the furnace through the checkers *K* and *L* and passing out through the checkers *I* and *J*, the sequence of operation is:

1. Steam is cut off from the producers;
2. Air-stack valve *F* is closed;
3. Gas-inlet valve *A* is closed;
4. Air-inlet valve *C* is closed;

5. Gas-stack valve B is opened. (This passes the gas trapped in the gas checker K to the flues and boiler, thence to the stack);
6. Gas-stack valve G is closed;
7. Air-inlet valve E is opened. (This admits air to the furnace);
8. Gas-inlet valve H is opened;
9. Air-stack valve D is opened;
10. Steam is turned on to the producers.

The important feature of this system of valve operation is the time interval between the opening of the gas and the air stack valves on the same side of the furnace. This interval must be sufficient to permit the inflammable gases in the gas checker to pass into the flues and to the stack before the air-stack valve is opened. In the plant where this system was introduced the number of explosions was reduced from about 40 per day to 4 in 411 reversals in a period of five days.

Several other systems are in use, and in some installations the valves are interconnected in sets which are operated simultaneously. Different arrangements of flues and different valves will require some modification of this system, the essential point being the interval between the opening of the gas-stack valve and the air-stack valve.

This sequence of valve operation will not eliminate explosions unless the valves and the flues are sufficiently tight to prevent any air entering the system while the gas trapped or pocketed in the checker chamber is passing to the stack.

FAULTY BOILER SETTINGS

Probably the most important portion of the waste-heat boiler installation is the arrangement of the boiler setting with regard to the manner in which the gases pass through it—that is, the baffling and the location of the gas inlet and outlet. The design of boiler settings and their baffling has been the subject of much experimenting, but unfortunately most of these experiments have been made without any very clear conception of the action of the hot gases while flowing past cool metallic surfaces.

The ruling temperature in steam boilers is very low. For this reason it would require designing talent of an extremely high order and much serious study to produce a boiler which would not work. Practically all of the boilers which have been built and installed present extremely gross violations of some of the simplest and most elementary laws of physics, not only in the circulation of the heated gases but in the circulation of the water and steam. Owing to their low ruling temperature these boilers work—that is, they produce steam when hot gases pass through them, but their utilization of the heat is comparatively inefficient when the possibilities of such low-temperature applications are considered.

Some three years ago the writer, in the course of a discussion before the Cleveland Engineering Society, stated that it seemed to him that commercial boilers were much better designed for the production of soot, a form of lampblack, than they were for the production of steam. Since then he has seen no occasion to reverse his opinion. At the same time he will admit that the low rate of evaporation presents a certain margin of safety in boiler operation —that is, the operating safety of a steam boiler depends upon the maintenance of the supply of feed water. When the water supply fails a very few minutes elapse before the water in the boiler will be evaporated to a point where portions of the heating surface will become dry. This is the danger point, and the higher the rate of evaporation as compared with the volume of water contained in the boiler, the quicker it will pass this danger line.

When the method of baffling steam boilers is examined, it will be found that the hot gases are introduced at the lowest point of the setting, that these gases rise in two of the passes and drop through the middle pass, and are carried away from the highest point of the setting. Experience with regenerators and similar heat-absorbing structures has shown that this arrangement of gas passages is absolutely illogical. Some waste-heat boilers without baffling have been installed in connection with copper-smelting furnaces, but in these installations the baffling was removed in order to reduce the resistance to the passage of the gases through the boiler, and the manner in which the gases were introduced and carried away from the setting was not calculated to obtain the best results. The main idea in the design appeared to be that the gases should pass through the boiler setting as rapidly as possible without any consideration of the utilization of their sensible heat, while two boilers were placed in series in order to reduce the temperature of the outgoing gases. The precedent of this design has been followed in other cases.

There is a considerable diversity of opinion in regard to the manner in which waste-heat boilers should be rated. In America boilers are usually rated at 10 sq ft (0.933 m^2) per horsepower, which is equivalent to an evaporation of 3 lb (14.6 kg) per square meter of heating surface. Some have considered that this rating should be reduced to 7.30 kg for waste-service service, in spite of the fact that many boilers are in service at rates of evaporation considerably in excess of the above figures. When it comes to the selection of a boiler for any given waste-heat installation a consideration of some of the installations which have been made indicates wide differences in regard to the area of heating surface required.

Waste Heat Boilers Installed on Open-hearth Furnaces
(Thomas B. Mackenzie, Iron and Steel Institute, 1918)

	I	III	V	VII	IX
Nominal capacity of furnace, tons	30	45	100	60	60
Heating surface,* sq m	151.00	170.00	204.00	204.00	204.00
Economizer heating surface, sq m	66.2	89.4	111.9	111.9	111.9
Steam pressure, absolute, per sq cm, kg	7.14	8.35	5.20	6.18	8.64
Feed water, initial temp	11.6	6.95	40.8	9.5	29.3
Feed water, final temp	117.8	120.6	133.5	133.00	129.00
Gas temperatures, deg. C.:					
Entering boiler	504.00	585.00	422.00	577.00	439.00
Leaving boiler	254.00	273.00	232.00	303.00	304.00
Drop in boiler	250.00	212.00	190.00	274.00	135.00
Leaving economizer	169.00	182.00	184.00	172.00	264.00
Drop in economizer	84.00	91.00	48.00	131.00	40.00
Draft in mm of water:					
At boiler inlet	....	23.00	40.00	25.00	25.00
At boiler outlet	....	62.00	78.00	63.00	95.00
Drop through boiler	....	39.00	38.00	38.00	70.00
At draft fan	20.00	87.00	80.00	76.00	146.00
Drop through economizer	....	25.00	2.00	13.00	51.00
Probable volume of gases passing through boiler per second, cu m at zero and 760 mm	9.74	10.35	15.50	15.50	24.60

* Boilers all Babcock & Wilcox type.

OTHER WASTE HEAT BOILER DATA

Test Number	1*	2	3	4
Plant	Ill. St. Co.	Indiana St. Co.	Bethlehem Co.	Lackawanna Steel Co.
Rated furnace capacity, tons	65	75	80	00.00‡
Actual furnace capacity, tons	72†	85	82.6	00.00
Type of boiler	Stirling	Rust	B. & W.	B. & W.
Boiler heating surface, sq m	371.6	453.3	486.0	502.3
Steam pressure absolute, kg per sq cm.	9.76	9.86		
Superheat, deg. C	71.00	98.00	67.00	54.00
Gas at boiler inlet	664.00	624.00	739.00	530.00
Gas at boiler outlet	327.00	277.00	256.00	242.00
Drop in temperature	337.00	347.00	483.00	288.00
Draft at boiler inlet, mm	....	37.00	39.00	45.00
Outlet	....	100.00	83.00	92.5
Draft loss in boiler	45.2	63.00	44.00	47.5
Weight of gas passing through boiler, kg per second	9.22	10.60	9.50	10.00

* Average of ten tests. † Approximate. ‡ Tilting furnace.

Tests Nos. 1, 2 and 3 from C. J. Bacon's paper at 1915 meeting of the American Iron and Steel Institute.

Test No. 4, Arthur D. Pratt, American Society of Mechanical Engineers, December, 1916.

The following method of arriving at the amount of boiler-heating surface required for absorbing the waste heat from an open-hearth furnace is simply an extension of the method used by the author in arriving at the regenerator capacity required and reference must be made to the heat-capacity curves given in the preceding section of this work.

In practice it might be desirable to divide this heating surface between an economizer or feed-water heating section and a boiler or steam-producing section. This method of construction will result in a reduction of the size of the boiler and its cost, and possibly may reduce the total cost of the installation. When the cost of the foundations and other structure required to install the economizer is considered it is probable that the

	Gas Checker	Air Checker
Temp. products of combustion at bottom of checkerwork, degrees C	800	600
Products of combustion per cu meter of gas burnt, cu m	0.90	1.78
Products of combustion $=0.90+1.78=$	2.68	
Average temperature products of combustion $=[(800\times 0.90)+(600\times 1.78)]\div 2.68$	665	

(No allowance is made for a heat loss in the flues, although in practice a temperature drop of 50° to 100° will occur between the bottom of the checkerwork and the boiler inlet. The heat capacities given by the curves are based on the consumption of 100 molecular volumes or 2.232 cu m. of gas. These values have been reduced to those for a cubic meter of gas burnt in this computation.)

Heat capacity products of combustion of 1 cu m of gas at the bottom of the checkerwork, average $t=665$, calories	635	
Heat capacity products of combustion of 1 cu m of gas at the boiler outlet, average $t=200$, calories	162	
Heat available for production of steam, calories	473	
Assuming that one-third of this heat will be lost in flues and boiler setting, net amount of heat available will be, calories	316	
Heat required for evaporation of 1 kg of water at 5.54 kg per sq cm absolute (78.76 lb per sq in) $=156+500$, calories	656	
Products of combustion from 1 cu m of gas will evaporate: $316\div 656$, kg		0.481
1 sq m of boiler heating surface is equivalent to the evaporation of 15 kg of water per hour; or 0.00417 kg of water per second		
Area of the heating surface required to evaporate 0.481 kg of water per second will be: $0.481\div 0.00417$ sq m.	116	
(This being the area of the heating surface required to absorb the heat in the products of combustion from 1 cu m of the gas burnt)		
As the average amount of gas burnt per second is 1.75 cu m, the area of the heating surface required will be: 116.00×1.75	203	

Quantity of gas burnt per second, p. 231.

total cost will be about the same for both methods of installation.

Another factor that has to be considered, is the area of the gas passage through the boiler must be sufficient to permit the maximum volume of the products of combustion to pass without adding unduly to the draft resistance through the boiler.

In the foregoing computation the temperature of the boiler feed water was taken as zero C. In practice this will not be the case, if the average temperature of the water is considered.

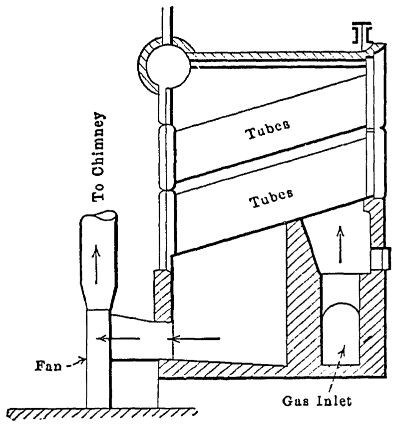


FIG. 157.—Proposed Setting of Marine Type of Waste Heat Boiler.

THE MATTER OF THE DRAFT

Stationary or land practice has persistently used brick boiler settings in spite of their many disadvantages. Brick settings are undoubtedly less costly than air-tight steel-sheathed boiler settings, and the fact that the weight of the boiler setting on land is unimportant has led to the almost exclusive use of the porous brick setting, owing to its lower cost. There are a number of places where the extremely low-draft pressure makes the brick setting comparatively unobjectionable, but in open-hearth and other waste-heat installations where a large draft differential is required the porosity of the brick setting is objectionable. In addition the brick setting will be badly damaged by any explosions which may occur; the brickwork is readily cracked and these cracks admit air which will produce explosions at each reversal.

Practically all waste-heat boilers installed upon open-hearth furnaces have necessitated the installation of an induced draft fan. With the commercial types of boiler set and baffled to the makers' drawings, these fans will be necessary, or else an unduly high chimney. The pressure in the heating chamber of the furnace is equal to the atmospheric pressure and it is necessary to supply a draft depression below the regenerators sufficient to remove the products of combustion from the furnace; additional draft depression must be supplied to overcome any friction in the passages through which the waste gases pass. There are limits to the height of chimney which it is desirable to install in connection with an open-hearth furnace and any further increase in the draft depression must be secured by the installation of a draft fan. When these fans are operated by steam they will consume about one-fifth of the total steam generated and the balance will be available for the gas producers and for supplying the other power demands of the plant.

The primary function of the open-hearth furnace is the production of steel and this should be kept in mind in the design of the waste-heat boiler setting and flues. The boiler should be bypassed so that any failure in these portions of the equipment will not necessitate the shut-down of the furnace. In some cases steam-jet apparatus has been installed to provide against the failure of the fan or its motive power.

The venturi coned ejector form of chimney has been employed in some installations. This type of chimney may be used in either of two methods, the fan may be reduced in size and only handle a portion of the waste gases, or the Louis Prat method may be employed, in which the fan handles cold air only. This latter method is analogous to that of the hydraulic head increaser designed by Clemens Herschel for use with low head hydro-plants.

One of the incidental advantages of introducing induced draft in the operation of the furnace arises from the fact that the fan draft may be increased to compensate for the blocking up of the checkers and the operation of the furnace will be entirely independent of those barometric and weather conditions which affect chimney draft.

Reversing valves for regeneratively fired furnaces have been a source of much trouble. Many different valves have been designed and placed upon the market, and a number of different flue arrangements have been devised to eliminate the reversing valve and accomplish the reversal with a multiplicity of mushroom valves and dampers. The simple Siemens butterfly valve was the first four-way valve used on these furnaces. When in use, however, and exposed, on one side, to the hot gases and on the other, to the cool gas or air passing to the regenerators, it soon warps and becomes leaky. These leaks permit air to pass directly to the stack and cool down the waste gases; or, if the valve is used for gas, there is continual leakage of gas, which burns either in the valve or in the stack flue.

While many reversing valves are water-sealed, most of them, like the butterfly valve, during the operation of the valve, open a direct connection, practically the full area of the valve from gas main or air to the stack. There have been a few valves which cut the furnace, gas and air entirely off from one another and the stack, but these valves have not come into extended use.

Water-sealed valves are used extensively. As long as the seal

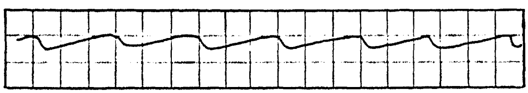


FIG. 158.

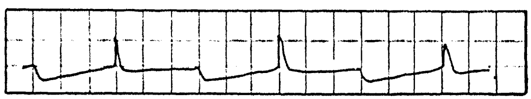


FIG. 159.

FIGS. 158 and 159.—Pyrometer Diagrams of Stack Temperatures. Abnormal conditions shown by sudden upward kick in Fig. 159 are probably due to air leakage and the combustion of checker gases passing along the flue to the stack.

holds they are tight, but there are usually structural limitations to the depth of the seal. When exposed to gas pressure on one side and to the stack depression on the other, the seal is unbalanced and may readily be broken by surges or explosions. In some cases, a considerable water area is exposed to the entering gas or air, as well as to the waste gases.

The producer gas and the stack gases are several hundred degrees hotter than boiling water and will absorb a considerable amount of moisture from a very small area of water surface. Other valves expose very little water surface. With all of these valves the sealing lip must be raised to clear the water surface and the port rims, whenever the valve is operated. From the time the seal is broken until it is reestablished, the full suction of the stack acts to pull air or gas, and in some cases both, into the stack.

With some valves, the furnace itself is directly connected to the stack through both regenerators, at reversal, so that a portion of the stack-pull tends to draw air in at the doors. This may or

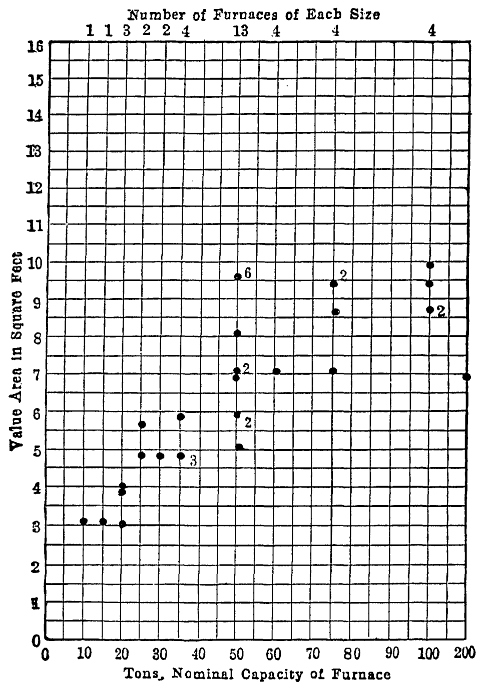


FIG. 160.—Graphical Comparison of Gas Valve Areas of Various Open-hearth Furnaces.

may not be seriously objectionable, according to the distance between the valve and the furnace and the rapidity with which the valve may be operated.

Moisture absorbed from water seals is a direct loss of the amount of heat required to evaporate it and superheat it to the temperature at which it passes out of the regenerator to the stack. In addition, its dissociation probably occurs in the checkers, which may release some oxygen to combine with other combustibles at this point. The reactions here are complex, as certain hydrocarbons dissociate in the checkers, as well as CO_2. A further increment of moisture occurs in certain elements of the charge, and an open-hearth furnace is not particularly efficient as a dryer. At the same time, moisture is carried in by the air supply. All of this water leaves the regenerator for the stack as highly super-

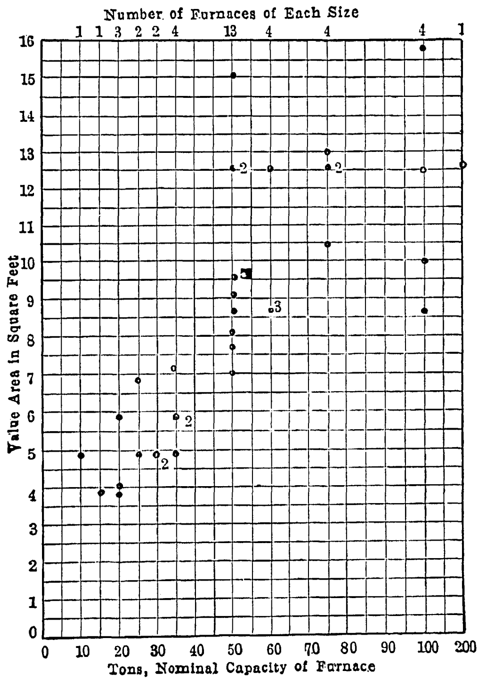


FIG. 161.—Graphical Comparison of Air Valve Areas of Various Open-hearth Furnaces.

heated steam, and its amount is considerable, particularly with large furnaces. In "The Heat Balance of the Open-Hearth," by Sidney Cornell (*Chemical and Metallurgical Engineering*, May, 1913), the weight of moisture passing in the flue gases was given as about 26 per cent of the weight of ingots produced. The water seals in the valves increased the amount of moisture in the producer gas 1 per cent.

The water seal depends upon its water supply, and a very slight stoppage breaks the seal. Frozen water lines in winter frequently cause the superintendent to consign the plant to the tropical regions, particularly when they occur on a cold Sunday morning. Water vapor absorbed by the flue gases reduces not only their temperature, but also the stack draft and the amount of heat available for the waste-heat boiler to convert into steam.

As to the valve area required for a furnace, there is as much diversity in practice as there is with the other portions of the furnace. Figs. 160 and 161 are a graphical comparison of reversing valve practice, while Table 10 shows the same data in tabular form. The reference numbers in the table correspond with those given in preceding chapters.

TABLE 9

REVERSING VALVES

Nominal Diameter Valve			Nominal Area of Valve		
In	Ft	mm	Sq In	Sq Ft	m^2
18	1.50	450	254	1.77	0.164
21	1.75	525	346	2.40	0.223
24	2.00	610	452	3.14	0.292
27	2.25	685	572	3.90	0.362
30	2.50	760	706	4.91	0.456
33	2.75	840	855	5.93	0.548
36	3.00	915	1017	7.07	0.657
40	3.33	1000	1256	8.73	0.808
42	3.50	1070	1385	9.62	0.894
48	4.00	1220	1809	12.57	1.168
54	4.50	1370	2290	15.90	1.477
60	5.00	1525	2827	19.63	1.823
66	5.50	1675	3421	23.76	2.210
72	6.00	1830	4071	28.27	2.930

TABLE 10

REVERSING VALVES AND CHIMNEYS IN PRACTICE

Nominal Furnace Capacity, Tons	Valve Area				Chimney.		
	Gas, Sq Ft	Air, Sq Ft	Sq Ft per Ton		Height, Ft	Area of Sq Ft	
			Gas	Air		Bore, Sq Ft	Per Ton
10–1	3.12	4.87	0.312	0.487	100	9.63	0.963
15	3.12	3.12	0.208	0.208	90	12.56	0.837
20–A	3.90	3.90	0.195	0.195	100	.....	.....
20–3	3.07	4.00	0.154	0.200	.....	.....	.....
20–D	3.90	5.90	0.195	0.295	.....	.....	.....
20–4	.....	.....	.....	.....	.....	5.85	0.293
25–4	4.90	4.90	0.196	0.196	125	12.56	0.502
25–3	5.65	6.90	0.226	0.276	125	19.64	0.786
30–5–4	4.90	4.90	0.163	0.163	125	14.20	0.473
35–2	5.90	5.90	0.169	0.169	.....	.....	.....
35–A	4.90	5.90	0.169	0.169	114	17.00	0.487
35–1	4.90	4.90	0.140	0.140	130	15.90	0.474
35–D	4.90	7.10	0.140	0.203	.....	.....	.....
40–7	.....	.....	.....	.....	.....	9.00	0.225
40–4	.....	.....	.....	.....	140	19.64	0.491
50–4	7.06	7.06	0.141	0.141	150	23.70	0.474
50–10	7.06	15.00	0.141	0.300	.....	21.50	0.430
50–11	5.94	9.18	0.119	0.184	150	26.27	0.525
50–8	9.60	12.50	0.192	0.250	150	28.25	0.565
50–13–15	9.60	9.60	0.192	0.192	150	25.80	0.516
50–16	9.60	12.50	0.192	0.250	160	20.00	0.400
50–X	5.22	7.75	0.104	0.155	.....	.....	.....
50–4α	8.13	8.13	0.163	0.163	.....	.....	.....
50–19	9.60	9.60	0.192	0.192	150	20.00	0.400
50– 5	7.00	9.52	0.140	0.190	.....	.....	.....
50–D	5.90	8.70	0.118	0.174	.....	.....	.....
60–13	7.06	12.58	0.118	0.210	180	33.18	0.553
60–3–5	7.06	8.72	0.118	0.145	153	28.27	0.471

TABLE 10—*Continued*

REVERSING VALVES AND CHIMNEYS IN PRACTICE

Nominal Furnace Capacity, Tons	Valve Area				Chimney		
	Gas, Sq Ft	Air, Sq Ft	Sq Ft per Ton		Height, Ft	Area of Sq Ft	
			Gas	Air		Bore, Sq Ft	Per Ton
75–1	9.42	13.00	0.126	0.173	.....	.....	.....
75–A	9.42	12.56	0.126	0.167	.....	.....	.....
75–3	8.70	12.56	0.116	0.167	.....	.....	.....
75–D	7.10	10.50	0.095	0.140	.....	.....	.....
75–2	.....	.....	.....	.....	165	20.36	0.271
80–1	.....	.....	.....	.....	.....	15.80	0.198
100–3	.....	.....	.....	.....	.....	19.60	0.196
100–H	10.00	10.00	0.100	0.100	180	28.25	0.285
100–E	8.75	8.75	0.088	0.088	180	50.00	0.500
100–4	9.42	15.90	0.094	0.159	160	28.25	0.283
100–D	8.70	12.60	0.087	0.126	.....	.....	.....
150–1	.....	.....	.....	.....	150	23.70	0.158
200–4	7.00	12.56	0.035	0.063	180	38.50	0.385

Some of these valves are so heavy as to require electric motors, or some other form of power, for their operation. In these cases the control is located on a pulpit at a central location on the charging floor, in the rear of the furnace. Smaller and lighter valves are operated by cables or levers led into the pulpit. Heavy valves have considerable inertia, and for this reason cannot be operated as rapidly, even by power, as the smaller valves with lighter moving parts.

Furnaces fired by natural gas, oil, coke-oven gas or tar require reversing valves for the air only, the fuel being reversed by shutting off the jet at one end of the furnace and turning on the jet at the opposite end. Some of these furnaces are supplied with one checker chamber at each end, while others are so constructed, with two chambers at each end, that they may be converted with little difficulty to producer-gas firing.

Those furnaces in which only the air is preheated have a slight advantage over those in which both the gas and the air are preheated, in that no unburned gas has to be wasted up the stack at reversal. The amount of fuel lost in this manner depends upon the gas-filled volume between the reversing valve and the port, and the frequency of reversal. When the gas is preheated this loss cannot be avoided.

This gas likewise creates an explosion hazard; when the conditions are right, it burns and passes up the stack as a puff of flame. Again it may become mixed with air, the mixture being below the ignition temperature; when a portion of this mixture is suddenly ignited an explosion of more or less violence occurs. These explosions damage the walls of regenerative chambers, flues and waste-heat boilers so that large amounts of air are drawn into the system, reducing the stack draft and the output of the boiler by reason of the lowering of the temperature of the waste gases.

One of the factors in regard to valve area that meets with little consideration is the velocity of the gases passing through the valve. In addition, most valves involve a change in direction of flow, totaling 360°, 180° in the valve and two 90° changes in the flues. When a stream of flowing gases passes through passages involving changes in area, velocity changes are involved. The velocity of flow in the normal section of the flue may be represented by $V_{\min}$ and in the contracted area of the valve by $V_{\max}$. The corresponding velocity heads will be $h_{\min}$ and $h_{\max}$. That is, a velocity head $= h_{\max} - h_{\min}$ will be required to produce the increase in velocity. The pressure, in kilograms per square meter or in millimeters of water, required to produce the increase in velocity will be

$$\delta = (h_{\max} - h_{\min})\Delta_t,$$

in which δ = pressure in kilograms per square meter or millimeters of water;

Δ_t = the weight of 1 cu m of the gas in motion at a temperature of $t°$.

When the pressure is desired in inches of water, the weight of the gas being in pounds per cubic foot, the formula is

$$\delta = 0.192(h_{\max} - h_{\min})\Delta_t.$$

The coefficient of contraction varies from unity, when the areas of the two passages are the same, to 0.83, when the area of the smaller passage is 0.01 of the area of the larger passage. This slightly increases the pressure required, but ordinarily a large margin is available to cover this increase. The loss of pressure due to changes of direction may be expressed by the formula,

$$\delta = r\Delta_t \frac{v^2}{2g},$$

in which r is a function of the angle through which the stream is deflected. The following values are given by Weisbach for short bends in pipe:

Angular change $\alpha =$	20°	40°	45°	60°	80°	90°
$r =$	0.046	0.139	0.188	0.364	0.740	0.984

v = velocity in meters per second;
$2g$ = gravitational constant $= 2 \times 9.81$.

When the loss of pressure due to directional changes is desired in inches of water column, the formula is

$$\delta = 0.192 r\Delta_t \frac{v^2}{2g},$$

in which v = velocity in feet per second;
$2g = 2 \times 32.2$;
Δ_t = weight per cubic foot of gas in motion at $t°$.

The losses, due to directional changes of 180° in the valve and 90° in the flues each side of the valve, are approximately from two to four times as great as the loss due to restricted valve area. For this reason, changes of diameter of valve of 6 in (150 mm) or so have a comparatively small effect upon operation. These losses vary with the square of the velocity in the valve and the flues. Many furnaces are, undoubtedly, choked by the small area of the valves used, and much operating trouble is doubtless due to lack of consideration of these details.

A valve small enough to choke the furnace is an expensive luxury, as it exacts its toll every minute the furnace is working. Added chimney height or forced draft must be provided to overcome its resistance. There is very little doubt that the erratic working of some furnaces with different weather conditions is

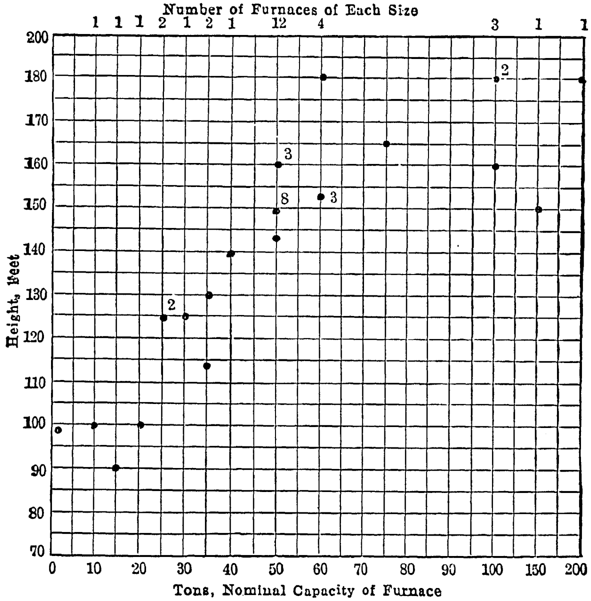


FIG. 162.—Graphical Comparison of Chimney Heights of Various Open-hearth Furnaces.

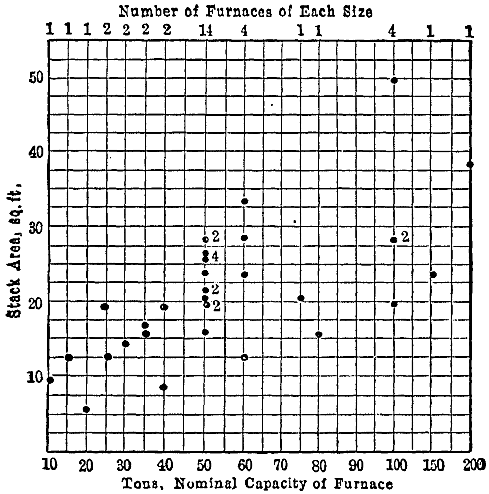


FIG. 163.—Graphical Comparison of Chimney Areas of Various Open-hearth Furnaces.

largely due to the fact that the close balance between draft and resistance is disturbed by barometric and temperature conditions. These various losses will be discussed in more detail in the design computations for a furnace.

Figs. 162 and 163 give a graphical comparison of chimney areas and heights which are tabulated in Table 10. Open-hearth practice in the United States tends to the use of self-supporting steel chimneys lined with firebrick. Even with waste-heat boilers and induced draft, the straight chimney is used, while abroad many of the venturi cone (Prat type) chimneys are used. The chimney is a necessity, as the waste gases must be carried to a sufficient height to prevent their becoming a nuisance, not only in the works, but to its neighbors.

Another factor which must be considered, in connection with the draft required to operate the furnace, is that the gases must be pulled out of the furnace through the ports, down through the checkers and through the valves and flues; that is, the waste-gas end of the furnace is below atmospheric pressure, and there is a constant tendency for cold air to be pulled into the system. When this brickwork is new and tight the leakage of air into the system may be slight, but after a few explosions have shaken things up, the brickwork is liable to leak like a sieve. For this reason, gas samples taken at the base of the stack are liable to show a condition of affairs quite different from that which actually exists in the gases leaving the heating chamber or the regenerators.

A great many tests and considerable investigation of open-hearth furnaces have been made, from time to time, at different plants. Tests of this kind cannot be permitted to interfere with furnace operation. They must be carried on night and day, over several melts. When it is decided to run such a test the question of cost must be considered, not only in apparatus, but in personnel. A large number of observations and chemical analyses must be made and the results analyzed.

Very slight details are liable to vitiate the value of such tests and it is extremely difficult to impress upon the available assistants the factors that are really essential; frequently, in fact, the busy executive is unable to devote the preliminary attention he desires to the test scheme, and, as a result, only a portion of the data desired is obtained. Comparatively few of these tests are made known, hence there is much repetition of the work of others.

THE DESIGN OF OPEN-HEARTH FURNACES

Design computations for an open-hearth furnace do not require any large amount of mathematical knowledge, for the principles involved are comparatively simple. As in all other engineering and chemical propositions, it is necessary to effect a compromise between a number of different requirements and co-ordinate them in the right manner to secure a desired result. All of the different elements of the problem are closely interrelated, and changes and modifications at one point necessitate carrying a corresponding modification through the entire system which it affects.

In order to establish a point of departure, it has been assumed that a furnace having a nominal capacity of 100 tons is to be designed, that it is to be fired with producer gas, and that the actual hearth area will be 650 sq ft. The computations will be limited to those required to establish the design lines. Certain factors may be more or less arbitrarily fixed without regard to current practice or whether they are desirable or not, merely for the purpose of furnishing a working base, the purpose of the computation being to illustrate the formulas used and their application.

The reactions in the open hearth are well known. Their main effect, as far as the flue gases are concerned, is an increase in CO_2 and in the moisture content, the latter in the first part of the heat, and the former during the boil and after the limestone is added. The fuel consumption will vary with the furnace, the method of working, etc. Ranging from 220 kg (485 lb) per ton, using molten pig, up to 350 kg (770 lb) per ton and higher. The fuel consumption is not uniform during each furnace cycle, from charge to charge, but varies about as follows:

Percentage of Cycle (Time)	Percentage of Fuel Consumed
63.00	76.00
17.00	15.20
14.00	8.40
1.00	0.40
5.00	......
100.00	100.00

TABLE 11

COMBUSTION OF PRODUCER GAS (SC–DSW)

Volumetric Composition of Gas, Per Cent		B.t.u. per Cubic Feet of Element	B.t.u. per 100 Cubic Feet of Gas	O_2 Required	Products of Complete Combustion, Cubic Feet		
					CO_2	H_2O	N_2*
H_2	12.10	× 293.2	= 3,548	6.05	.....	12.10	24.20
CH_4	2.60	× 983.4	= 2,557	5.20	2.60	5.20	20.80
C_2H_4	0.40	×1610.0	= 644	1.20	0.80	0.80	4.80
CO	21.78	× 343.6	= 7,484	10.89	21.78	.....	43.56
O_2	0.02	.....	.....	−0.02	.....	.....	−0.08
CO_2	5.68	.....	.....	.....	5.68	.....	.....
H_2O	3.82	.....	.....	.....	.....	3.82	.....
N_2	53.60	.....	.....	.....	.....	.....	53.60
	100.00	*Low*	14,233	23.32	30.86	21.92	146.88
H_2O	18.10	× 54.06	= 988				
		High	15,221				

Theoretical air supply $= 23.32 \times 5.00 = 116.60$ cu ft per 100 cu ft of gas.

* Assuming the atmosphere to consist of 80 per cent N_2 and 20 per cent O_2.

Air Per Cent	Supply Volumes	Excess Air	Products of Combustion of 100 Volumes of Gas				
			O_2	CO_2	H_2O	N_2	Total
100	116.60	0.00	0.00	30.86	21.92	146.88	199.66
120	139.92	23.32	4.64	30.86	21.92	165.53	222.98
140	163.24	46.64	9.09	30.86	21.92	184.19	246.30
180	209.88	93.28	18.18	30.86	21.92	221.50	292.94
			Percentage Basis (Wet)				
100	....	....	0.00	15.64	11.11	73.25	....
120	....	....	2.00	14.03	9.97	73.94	....
140	....	....	3.75	12.72	9.03	74.50	....
180	....	....	6.31	10.71	7.50	75.37	....
			Percentage Basis (Dry)				
100	....	....	0.00	17.60	....	82.40	....
120	....	....	2.29	15.58	....	82.13	....
140	....	....	4.12	13.97	....	81.91	....
180	....	....	6.83	11.59	....	81.58	....

For the case in hand it is assumed that the fuel consumption will be about 300 kg (660 lb) per ton converted into producer gas of the following composition: H_2, 12.10; CH_4, 2.60, C_2H_4, 0.40; CO, 21.78; O_2, 0.02; CO_2, 5.68; H_2O, 3.82; N_2, 53.60, having a thermal value of 1265 calories per m^3 or 142 B.t.u. per cubic foot (low values), the high values being 1352 calories per m^3 or 152 B.t.u. per cubic foot. The heat capacity of this gas, the air supply required for its combustion and for the products of combustion are shown by the curve (Fig. 184). Table 11 gives the combustion and air-supply data for this gas. One pound of coal produces 70 cu ft of gas, 1 kg, 4 m^3 39 of gas. Its specific weight is 1 kg 125 per m^3 (0.07024 lb per cubic foot). Its products of combustion, with 40 per cent excess air, weigh 1 kg 32 per cubic meter (0.08241 lb per cubic foot). At the maximum rate of working, the gas will be burned at the rate of 5 m^3 32 (188 cu ft) per second, requiring 8 m^3 51 (301 cu ft) of air for combustion, and the products of combustion will be 12 m^3 77 (451 cu ft), all volumes at 0° C., 761 mm barometer. As the volumetric corrections for pressure are comparatively small, they will be neglected in the computation, and the temperature corrections alone will be used.

As the capacity of the furnace is to be 100 tonnes, the metal volume will be 523 cu ft, and the hearth area has been fixed at 650 sq ft. The approximate depth of the bath will be:

$$\text{depth of bath} = d = \frac{3v}{a} = \frac{3 \times 523}{650} = 2.41 \text{ ft} = 29 \text{ in of metal.}$$

As the bottom will slope toward the tap hole and be banked at both ends and sides, the actual metal depth will depend upon the way this is done. There will be at least 12 inches of bottom. Therefore, the depth to the brick will be about 42 inches from the sills of the charging doors, and the port sill or bridge will be fixed as 6 inches higher, making the total depth that the flame must drop below the port 48 inches, or about 1200 mm.

The brick lines of the hearth will have to be fixed outside of the bath area of 650 sq ft. Therefore, the inside width between walls will be made 5 m 00 (16.40 ft), and the length between port sills 13 m 50 (44.25 ft), giving an area of 67 m^2 50 (726 sq ft). If the length is cut to 13 m 00 (42.65 ft.) the area will be reduced to

65 m² 00 (700 sq ft). The width might be reduced slightly by increasing the length. There are advantages in reducing the span of the roof, as well as disadvantages in making the furnace too long, but it is possible that a length of 14 m 00 (45.90 ft) with a width of 4 m 650 (15.25 ft), giving an area of 65 m² 11 (701 sq ft), would be satisfactory.

The temperature in the heating chamber will be 1800° C. (3272° F.) for the gases, the roof being 50° to 100° C. (90° to 180° F.) cooler, and the bath from 150° to 250° C. (270° to 450° F.) cooler. The height of the chamber may be approximated by Yesmann's formula

$$h_t = A\sqrt[3]{\frac{Q_t^2}{B^2 t}},$$

in which h_t = the thickness of the layer of gas in motion in meters (or feet);
Q_t = the volume of gas in cubic meters (or cubic feet) at $t°$ temperature;
B = the width of the furnace in meters (or feet);
A = a coefficient which varies for each value of h and B;
t = temperature Centigrade

TABLE 12

TABLE GIVING THE VALUES OF A FOR METRIC UNITS

h_t	Values of B		
	1 m 00	2 m 00	5 m 00
0 m 30	3.42	3.54	3.62
0 m 50	3.29	3.46	3.57
0 m 75	3.13	3.37	3.54
1 m 00	2.97	3.28	3.53

This formula may be translated to English units as follows:

TABLE 13

TABLE GIVING THE VALUES OF A FOR ENGLISH UNITS

h_t, Feet	Values of B				
	3 Feet	6 Feet	9 Feet	12 Feet	16 Feet
1.0	2.275	2.36	2.40	2.42	2.43
1.5	2.215	2.32	2.38	2.40	2.41
2.0	2.15	2.29	2.35	2.38	2.39
2.5	2.08	2.245	2.32	2.36	2.38
3.0	2.01	2.20	2.30	2.35	2.375

For the case in hand,

$Q_t = Q(1+\alpha t) =$ 12 m³ 77 $\times 7.606 =$ 97 m³ 14
451 cu ft $\times 7.606 = 3430$ cu ft
$B = 5$ m 00 (16.4 ft) or 4 m 650 (15.25 ft)
$t = 1800°$ C. (3275° F.)
$A = 3.53$ and 3.48 (approximately for metric units)
$= 2.37$ and 2.34 (approximately for English units)

The formula may now be written with numerical values, as follows:

For $B = 5$ m 00 (16.4 ft),

$$h_t = 3.53\sqrt[3]{\frac{97.14^2}{5.0^2 \times 1800}} = 2 \text{ m } 097 = 6.88 \text{ ft}$$

Or, for English units,

For $B = 15.25$ ft (4 m 65),

$$h_t = 2.34\sqrt[3]{\frac{3430^2}{15.25^2 \times 1800}} = 7.11 \text{ ft (2 m 169).}$$

This will be the distance from the surface of the bath to the center of gravity of the roof segment; it will give an approximate height of the skewbacks above the door sills of 5 ft 4 in (1630 mm) for the wider chamber and 5 ft 7 in (1700 mm) for the narrower chamber.

With a chamber area of 65 m² 00 (700 sq ft) the chamber volume will be approximately 144 m³ 00 (5087 cu ft) for the wider chamber and 149 m³ 50 (5280 cu ft) for the narrower chamber. The gases will remain in the chamber approximately 1.50 seconds, which, with a temperature drop of 200° C. (360° F.) per second, means a temperature of approximately 1500° C. (2732° F.) for the gases leaving the chamber. Referring to the curve (Fig. 184) and allowing for a drop in calorific intensity of about 200°, it will be seen that a preheat of the air and gas of between 800° and 1000° C. (1472° to 1832° F.) will be required. Allowing for a possible loss of temperature in the necks, cinder pockets and uptakes, the checkerwork will be proportioned to supply a preheat of 1200° C. (2192° F.).

The distance the jet of flame must drop below the bridge or port sill, in order to permit the sintering of the bottom, has been fixed at 1200 mm (48 in). A resultant velocity of the air and gas can be assumed, and this will fix the resultant angle of the two jets. The higher the resultant velocity is assumed, the less the resultant angle will be, and the further beyond the center of the chamber will be the point where the maximum depression of the jet of flame occurs.

High velocities, in addition, cause the incoming end of the chamber to work cold and the outgoing end to work hot, while it is desirable that both ends of the chambers work as uniformly as possible. High velocities for either the gas or the air mean reduced port areas, and high velocities for the outgoing products of combustion, which in turn call for an increased draft depression in order to pull the gases through the ports. This draft depression creates a suction acting to pull air in through the valves, flues and chamber walls. At the same time, it is necessary to have sufficient draft to draw the waste gases out of the chamber and down through the checkerwork, but the lower this draft depression the less the tendency to induce air leakage or infiltration.

As the flame has to drop 1200 mm (48 in) in one-half of 14 m 00 (46.00 ft), which is the length of the furnace, an angle somewhat greater than 10° must be allowed for the trajectory of the jet. Yesmann's formula for this case is

Metric: $$H=\frac{v^2 \sin^2 \delta}{2g}\times\frac{273+t_i}{t_m-t_i},$$

English: $$H=\frac{v^2 \sin^2 \delta}{2g}\times\frac{459+t_i}{t_m-t_i},$$

in which H = the middle ordinate of the parabola, in this case 1 m 200 or 4.00 ft;

v = the resultant velocity of the two jets uniting to form the flame;

$\sin \delta$ = the sine of the resultant angle of the two jets;

$2g$ = gravitational constant $= 2 \times 9.81 = 19.62$ in metric units or $2 \times 32.2 = 64.4$ in English units;

t_i = the temperature of the gases within the chamber;

t_m = the temperature of the gases in the jet of flame.

When a furnace is heating up, the stream of flame tends to follow the roof, until the interior of the chamber becomes raised to a temperature sufficient to permit it to drop, and the drop of the flame is an index of the progress made in heating the furnace. It is likewise desirable to be able to sinter the bottom, when the furnace, for any reason, is cooler than usual, or the ports are eroded. The temperature of the jet of flame, t_m, may be assumed as 1800° C. (3272° F.) and that of the gases in the chamber as 800° C. (1472° F.) $= t_i$; velocities of $v = 15$ m 00, 20 m 00 and 25 m 00 (49.2, 65.6 and 82.0 ft) per second will be tried out in the formula, solving for $\sin \delta$. The resultant angle for these velocities will be as follows:

For $v = 15$ m 00 (49.2 ft) per second, $\delta = 18° \ 12'$
$v = 20$ m 00 (65.6 ft) per second, $\delta = 13° \ 33'$
$v = 25$ m 00 (82.0 ft) per second, $\delta = 10° \ 46'$

In solving to obtain these angles

$t_i = 800°$ C. (1472° F.) $\quad 273 + t_i = 273 + 800 = 1073$
$t_m - t_i = 1800 - 800 = 1000$
$H = 1$ m 200 (4.00 ft) $\quad 2g = 2 \times 9.81 = 19.62$

The formula may now be written:

$$H = 1.20 = \frac{v^2 \sin^2 \delta \times 1073}{19.62 \times 1000},$$

which becomes

$$\sin \delta = \sqrt{\frac{21.94}{v^2}} = \frac{4.684}{v}.$$

For $v = 15$ m 00 (49.2 ft) per second, $\sin \delta = 0.3123$
$v = 20$ m 00 (65.6 ft) per second, $\sin \delta = 0.2342$
$v = 25$ m 00 (82.0 ft) per second, $\sin \delta = 0.1873$

These computations may be made in the English units, if desired. Should t_4 be given a higher value than 800° C. (1472° F.), the angle will be less. Two components may be selected to suit the resultant angle and velocity, but this cannot be intelligently done until the pressures available for impressing velocity upon both the gas and the air have been approximated. For the air, the pressure available will be entirely due to the chimney effect of the system, diminished by the resistance to the flow of the air, unless a fan is used. The same pressure is available in the case of the gas, plus the pressure in the gas main, which is more or less under control through the steam blower on the producer. In order to determine the chimney effect, it is necessary to obtain the height of the regenerator.

In a previous chapter several empirical rules were given for the amount of checkerwork required. According to Gruner, from 50 to 70 kg (50 to 70 lb) of checker brick are required per kilogram (pound) of coal burned per reversal. In the case considered, 5 m^3 32 (188 cu ft) of gas is to be used each second, each kilogram of coal producing 4 m^3 39 of gas (70 cu ft of gas per pound). The coal consumption, therefore, is 1 kg 212 (2.686 lb) per second. The weight of checker brick for this amount of coal would be from 60 to 85 kg (135 to 188 lb). According to this rule, the amount of checker brick would be:

TABLE 14

	50 to 1	70 to 1
For 15-min. (900-sec.) reversals	54,540 kg 120,220 lb	76,360 kg 168,320 lb
For 20-min. (12 0-sec.) reversals	72,720 kg 160,300 lb	101,810 kg 224,430 lb
For 25-min. (1500-sec.) reversals	90,910 kg 200,410 lb	127,250 kg 280,550 lb

Told suggests 6 m^3 00 (212 cu ft) or 2850 kg (6283 lb) of brick per cubic meter (35.3 cu ft) of air per second per 100° C. (180° F.) rise in temperature. According to this rule, the checker brick would be

$$2850 \times 8.51 \times 9 = 218{,}250 \text{ kg } (478{,}900 \text{ lb})$$

$$6 \times 8.51 \times 9 = 459 \text{ m}^3\ 55 \ (16{,}228 \text{ cu ft})$$

This last is reasonably close to the quantities arrived at, providing Told referred to the chambers at one end only.

The weight of the checker brick will be determined upon the basis of a temperature change of 100° C. (180° F.), 70 per cent of the brick being considered effective. The average specific heat of firebrick is about 0.25; its weight is about 1800 kg. per m^3 (112 lb per cubic foot).

The regenerator computations are made as follows:

TABLE 15

	Gas Checker on Gas	Air Checker on Air	Gas Checker on Waste Gases	Air Checker on Waste Gase
Temp. top of checkerwork, C....	1200°	1200°	1600°	1600°
F....	2192°	2192°	2912°	2912°
Temp. bottom of checker, C....	500°	300°	800°	600°
F....	932°	572°	1472°	1112°
Average temperature, C.........	850°	750°	1200°	1100°
F.........	1562°	1382°	2192°	2012°
Temperature increase or decrease in checker, C................	700°	900°	800°	1000°
F................	1260°	1620°	1440°	1800°
Heat capacity in calories per cubic meter from curve (Fig. 184):				
Waste gases at 1600°.........	....	....	1632	1632
Air or gas at 1200°.........	445	646	....	....
Waste gases at 800°.........	....	....	711	....
Waste gases at 600°.........	....	....	....	518
Gas at 500°.........	166	....	....	....
Air at 300°.........	....	148	....	....
Calories absorbed by gas and air or given up by waste gases.....	279	492	921	1114
Assuming 2 m^3 00 (70.6 cu ft) of gas are burned: 279×2=...	558	....	....	....
492×2=..........	....	984	....	....
Total: Required 558+ 984=	1542		....	
Available 921+1114=	....		2035	
Percentage required for gas: 558÷1542=................	36.19	....	....	....
Percentage required for air: 984÷1542=................	....	63.81	....	....

TABLE 16

	Gas on Gas	Air on Air	Gas on Waste Gas	Air on Waste Gas
Calories available for gas checker:				
$2035 \times 0.3619 =$	....	....	736	....
Calories available for air checker:				
$2035 \times 0.6381 =$	....	....	....	1299
Percentages for air and gas:				
$736 \div 921 =$	....	....	79.91	....
$1299 \div 1114 =$	....	....	....	116.61
Correction to make sum 200.....	....	....	1.41	2.07
Corrected values..............	....	....	81.32	118.68
Reducing to 100% basis.........	....	....	40.66	59.34
Calories per cubic meter of gas				
burned: $921 \times 0.4066 =$	....	....	374	....
$1114 \times 0.5934 =$	....	....	....	661
Calories required by preheat.....	279	492	....	....
Volumes of gas, air supply and products of combustion:				
Gas at 0° 760 mm...........	5 m³ 32	....	....	....
Air at 0° 760 mm...........	....	8 m³ 51	....	....
Products of combustion:				
$12.77 \times 0.4066 =$	....	....	5 m³ 19	....
$12.77 \times 0.5934 =$	....	....	....	7 m³ 58
	(188 cu ft	(300 cu ft)	(181 cu ft)	(270 cu ft)
Calories per second required:				
$5.32 \times 279 =$	1484	....	....	....
$5.32 \times 492 =$	....	2618	....	....
$5.32 \times 374 =$	....	....	1990	....
$5.32 \times 661 =$	....	....	....	3516
Calories per cycle:				
15 minute = 900 seconds......	1,335,000	....	1,791,000	....
		2,356,000		3,164,000
20 minute = 1200 seconds......	1,781,000	....	2,388,000	....
		3,141,000		4,219,000
25 minute = 1500 seconds......	2,226,000	....	2,985,000	....
		3,627,000		4,974,000

TABLE 16—*Continued*

	Gas on Gas	Air on Air	Gas on Waste Gas	Air on Waste Gas
Weight of checker brick required to store up heat in waste gas (100° change in brick temperature 70% effective):				
15-minute cycle:				
Gas				
[1,791,000 ÷ (100 × 0.25)] ÷ 0.7 =	....	....	102,350 kg	....
Air:				
[3,164,000 ÷ (100 × 0.25)] ÷ 0.7 =	....	....	....	180,800 kg
20-minute cycle:				
Gas:				
[2,388,000 ÷ (100 × 0.25)] ÷ 0.7 =	....	....	136,460 kg	....
Air:				
[4,219,000 ÷ (100 × 0.25)] ÷ 0.7 =	....	....	....	241,100 kg
25-minute cycle:				
Gas				
[2,985,000 ÷ (100 × 0.25)] ÷ 0.7 =	....	....	170,600 kg	....
Air:				
[4,974,000 ÷ (100 × 0.25)] ÷ 0.7 =	....	....	....	284,300 kg

Any change in the base assumptions made will alter these values in a corresponding manner. The change in temperature of the gas, air and waste gases will vary the quantity of heat available. The cyclic change in temperature may be varied; this will change the volume of brick required to store up the heat, and the surface required for heat interchange. The heat-storage capacity must be sufficient to supply the desired preheat, as well as the loss through the chamber walls, when on air or gas. Too much checker adds to the first cost; too small a checker increases the operating cost. The conditions will be changed by the use of other fuels, as well as by changes in the ratio of the air supply.

TABLE 16—*Continued*

	Gas on Gas	Air on Air	Gas on Waste Gas	Air on Waste Gas
The volume occupied by the checker brick will be = weight $\times \frac{1}{1800}$:				
15-minute cycle = ...	. . . .	. . . .	56 m^3 86	100 m^3 90
20-minute cycle = ...	. . . .	. . . .	75 m^3 82	133 m^3 94
25-minute cycle = ...	. . . .	. . . .	94 m^3 78	157 m^3 38
Pass volume in checkerwork:				
Q_0 of gas, air, waste gases	5 m^3 32	8 m^3 51	5 m^3 19	7 m^3 58
$t°$ average in checker	850°	750°	1200°	1100°
$1+\alpha t$ for average $t°$	4.1195	3.7525	5.4040	5.0370
$Q_0 \times (1+\alpha t) =$	21 m^3 92	31 m^3 93	28 m^3 05	38 m^3 18
Temperature change	+700°	+900°	−800°	−1000°
Average time in checker seconds, 200° per second change in t . . .	3.50	4.50	4.00	5.00
Basing pass volume on gas and air heating time, the pass volume =	76 m^3 72	143 m^3 68	. . . .	. . . .
Average time in checkers of waste gases, revised to pass volume, seconds	. . . .	. . . .	2.735	3.763
Total checker volume:				
15-minute cycle	133 m^3 58	244 m^3 58	4775 cu ft	8613 cu ft
20-minute cycle	152 m^3 54	277 m^3 62	5372 cu ft	9778 cu ft
25-minute cycle	171 m^3 50	301 m^3 06	6040 cu ft	10600 cu ft
Percentage of space occupied by				
Brick, 15-minute cycle	42.57	41.26	. . . .	. . . .
Pass	57.43	58.74	. . . .	. . . .
Brick, 20-minute cycle	49.70	48.25	. . . .	. . . .
Pass	50.29	51.75	. . . .	. . . .
Brick, 25-minute cycle	55.26	52.28	. . . .	. . . .
Pass	44.74	47.72	. . . .	. . . .
In order to provide a margin for all possible contingencies, the 25 minute cycle will be used				
Using square passes, the diameter of unit pass = $\sqrt{0.4474}$ = ...	0.6689	. . . .	. . . .	. . . .
= $\sqrt{0.4772}$ = ...	. . . .	0.6808	. . . .	. . . .
Brick = 1.0000 − 0.6689 = ...	0.3311	. . . .	. . . .	. . . .
1.0000 − 0.6808 = ...	. . . .	0.3192	. . . .	. . . .
Make both alike based on gas, using a 2.5 in. (63 mm.) brick:				
Dimension of unit pass = 63 ÷ 0.33 = b = ...	190 mm	7.5 in.	. . . .	. . . .
Brick thickness	63 mm	2.5 in	. . . .	. . . .
Diameter square pass	127 mm	5.0 in	. . . .	. . . .

The pass unit will be different for a different wall thickness, and it may be desirable to try several variations, in order to secure the most desirable arrangement. An old rule is that the active heat-storage capacity of brickwork is comprised in the depth of 1.25 in from the surface, giving a wall thickness for maximum results of 2.5 in (63 mm). When the brickwork is laid up to secure stability, 9-in straights give very good results. With thicker walls the time of the cycle must be increased, to develop the full weight of the brickwork. With the time cycle assumed, which agrees fairly closely with practical working, a wall thickness of 2.5 in (63 mm) utilizes its full heat-storage capacity. With a fifteen-minute cycle a 2-in wall would be desirable, but is probably a little too thin to give the best operating results.

TABLE 16—*Continued*

	Gas on Gas	Air on Air	Gas on Waste Gas	Air on Waste Gas
By using a 2-in (50 mm) brick, the size of the pass and the space occupied will be reduced:				
$50 \div 0.33 = b =$	150 mm	6.0 in	....	....
Brick thickness	50 mm	2.0 in	....	....
Diamet r square pass	100 mm	4.0 in	....	....
Wall thickness around pass	63 mm	2.5 in	50 mm	2.0 in
Area unit pass $= b^2 =$	0 m^2 0361	....	0 m^2 0225	....
Area of pass opening $=$	0 m^2 0162	....	0 m^2 0100	....
Area occupied by brick $=$	0 m^2 0199	....	0 m^2 0125	....
As the unit pass was made the same, the volume of brick and pass in the air checker will be changed to				
Brick $= 301.06 \times 0.5526 =$	166 m^3 37	....	....	....
Pass $= 301.06 \times 0.4474 =$	134 m^3 69	....	....	....

The lineal amount of pass and brickwork is found by dividing the brick and pass volumes by the area of the brick and pass in the pass unit. It will generally be found that there is a slight disagreement between the lengths determined for the pass and the brickwork. It is preferable to take the highest of the values so found for determining the size of the space to be occupied by the checkerwork. Another variant which will affect the actual checkerwork lies in the number of pass units making up the length and width of the chamber.

TABLE 17

	Gas Checker	Air Checker
The lineal amount of pass and brickwork required will be determined:		
For 63 mm (2.5 in) walls:		
Brick for gas = 94.78 ÷ 0.0199 = ...	4,763 m 00	....
air = 166.37 ÷ 0.0199 = ...	....	8,361 m 00
Pass for gas = 76.72 ÷ 0.0162 = ...	4,736 m 00	....
air = 134.69 ÷ 0.0162 = ...	....	8,314 m 00
For 50 mm (2.0 in) walls:		
Brick for gas = 94.78 ÷ 0.0125 = ...	7,830 m 00	....
air = 166.37 ÷ 0.0125 = ...	....	13,309 m 00
Pass for gas = 76.72 ÷ 0.0100 = ...	7,672 m 00	....
air = 134.69 ÷ 0.0100 = ...	....	13,469 m 00
The heating surface available in these checkers will be:		
For 63 mm (2.5 in) wall:		
4763.00 × 4 × 0.127 = ...	2,419 m² 00	....
8361.00 × 4 × 0.127 = ...	....	4,247 m² 00
	26,028 sq ft	45,680 sq ft
For 50 mm (2.0 in) wall:		
7,830.00 × 4 × 0.100 = ...	3,132 m² 00	....
13,469.00 × 4 × 0.100 = ...	....	5,388 m² 00
	33,715 sq ft	58,000 sq ft
The corrected volume of checkerwork will be:		
For 63 mm (2.5 in) wall:		
4763.00 × 0.0361 = ...	171 m³ 95	....
8362.00 × 0.0361 = ...	....	301 m³ 82
	6,073 cu ft	10,657 cu ft
For 50 mm (2.0 in) wall:		
7,830.00 × 0.0225 = ...	176 m³ 18	....
13,469.00 × 0.0225 = ...	....	303 m³ 02
	6,225 cu ft	10,703 cu ft

TABLE 18

	Gas Checker	Air Checker
A checkerwork height has been assumed, in order to illustrate the effect of a high checkerwork. On this basis:		
The number of passes required will be determined by dividing the lineal amount of pass required by the height of checkerwork:		
n for gas $=4763 \div 7.00=$	681	
n for air $=8361 \div 7.00=$	....	1195
Any desired length or width of chamber that will contain the number of passes may be selected. When a square pass is used, it is necessary that the chamber dimensions should be multiples of the pass dimension, plus one wall. The length of both chambers will be alike.		
Let n = number of passes in length of chamber, then $l=190n+63$.		
Assume $n=40$, then the length of the chamber will be: $190\times40+63=$	7 m 663	25.14 ft
For the width of the chambers, n may be assumed as some number which will give a number of passes slightly in excess of that computed above. For the case in hand, let $n=18$ for gas and 30 for air. The corrected number of passes will be, therefore:		
40×18 for gas $=$	720	....
40×30 for air $=$	....	1200
The width of the chambers required will be:		
For gas: $190\times18+63=$	3 m 483	....
For air: $190\times30+63=$	....	5 m 763
	11.425 ft	18.90 ft
The area occupied by the checkerwork will be:		
For gas: $7.663\times3.483=$	26 m² 69	....
For air: $7.663\times5.763=$	....	44 m² 16
	287.4 sq ft	475.4 sq ft
The corrected checkerwork volume will be:		
For gas: $26.69\times7.00=$	186 m³ 83	....
For air: $44.16\times7.00=$	....	309 m³ 12
	6602 cu ft	10,916 cu ft

TABLE 18—*Continued*

	Gas Checker	Air Checker
The heat surface per pass was computed as.....	3 m² 556	3 m² 556
The corrected value for total heating surface will be:		
For gas: 3.556×720 =....................	2560 m² 00	....
For air: 3.556×1200 =...................	....	4267 m² 20
	27,558 sq ft	45,935 sq ft
The corrected total area of the passes will be:		
For gas: 0.0162× 720 =..................	11 m² 664	....
For air: 0.0162×1200 =..................	....	19 m² 44
	125.58 sq ft	209.32 sq ft
The corrected average velocity per second of the gases will be:		
For gas: 21.92÷11.664 =.................	1 m 879	....
For air: 31.93÷19.44 =..................	....	1 m 642
	6.17 ft	5.38 ft
For waste gases: 28.05÷11.664............	2 m 405	....
38.18÷19.44 =..........	....	1 m 964
	6.71 ft	6.44 ft

TABLE 19

	Gas on Gas	Air on Air	Gas on Waste Gas	Air on Waste Gas
The heat transfer, total per second,				
in calories is.................	1484	2,618	1990	3,516
in B.t.u. is...................	5900	10,400	7900	13,900
Calories per sq m per second.....	0.5797	0.6135	0.7773	0.8239
B.t.u. per sq ft per second.......	0.214	0.226	0.287	0.306
Average temp. differential brick to gas or gas to brick, C.........	175°	175°	175°	175°
F.........	315°	315°	315°	315°
Convection factor to still air, according to curve plotted from Langmuir's experiments (Fig. 164) is calories per m² per second	0.294			
Correction factor for velocity =				
$K=\sqrt{\frac{35+v}{35}}=$..............	2.516	2.384	2.803	2.569
v = velocity in cm per second.				
Convection corrected for velocity:				
in calories per m² per second...	0.7397	0.7009	0.8241	0.7553
in B.t.u. per sq ft per second...	0.273	0.258	0.304	0.278

These heat-transfer values computed from Langmuir would seem to indicate the provision of sufficient heating surface to permit the desired rate of heat transfer to take place. Langmuir's

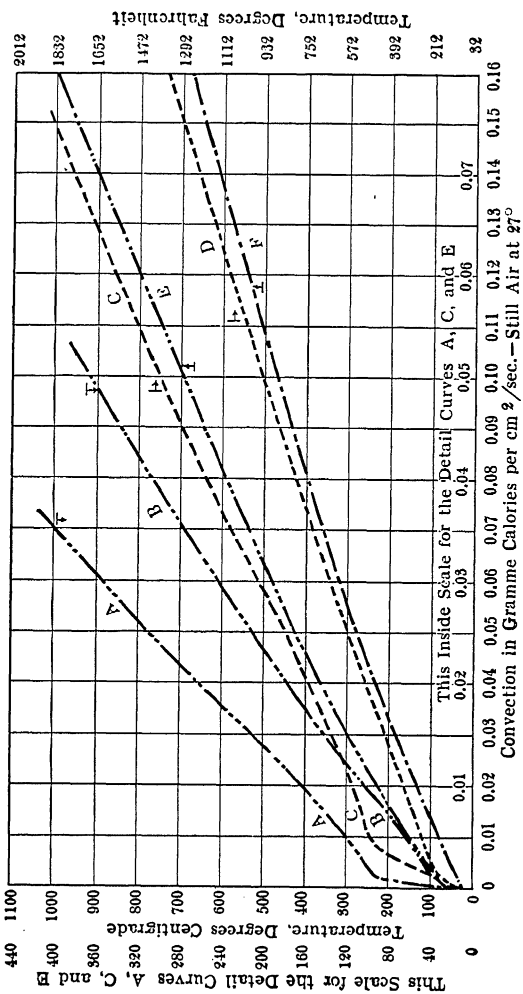


FIG. 164.—Convection Factor Curve Based upon Langmuir Experiments. Correction factor for velocity of flow equals $K=\sqrt{\frac{35+v}{35}}$, v = velocity in cm. per second.

work is probably as accurate as any obtainable to-day, but the writer has assumed that his temperature differential holds with higher initial temperatures, his values being based upon still air at 27° C. (80° F.). Whether these assumptions are correct cannot be determined without further research.

In the foregoing, a checker work of square passes has been assumed. The method of arriving at the heating surface of the usual checker construction is somewhat similar to the above.

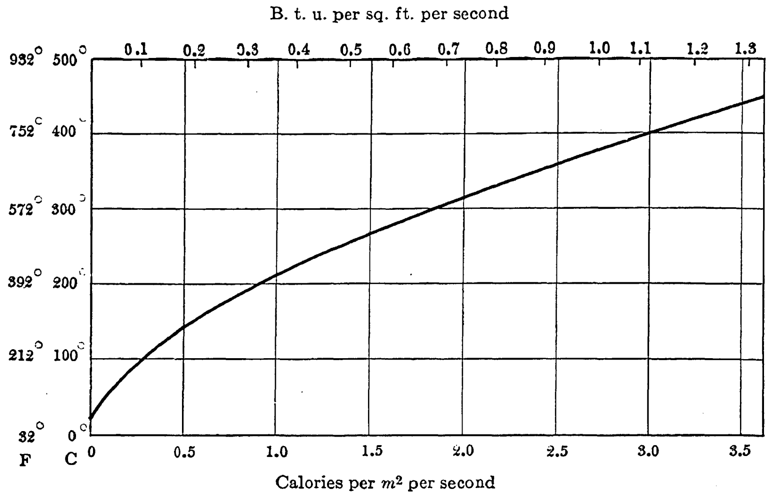


FIG. 165.—Curve of Heat Loss from Exterior Walls of Furnaces According to Experiments of Charles R. Darling.

The bricks occupy 0.5526 of the volume and the pass 0.4474; hence the number of rows per meter will be

$$n_b = 552.6 \div 63 = 8.771 \text{ rows}$$

or $$0.5526 \div 0.21 = 2.63 \text{ rows per foot.}$$

The gauge or space between the rows of brick will be

$$b_g = 447.4 \div 8.771 = 51 \text{ mm (about 2 in)}$$

or $$0.4474 \div 2.63 = 0.17 \text{ ft (about 2 in).}$$

From these figures the heating surface per square meter, or per square foot, may be readily computed, as well as all of the other data necessary to determine the sizes of the checker chambers for gas and air. When properly laid to secure stability, the 9-in straight makes a checker with the maximum heat storage utilization of the volume occupied.

All of the computations are based upon an assumed rate of fuel consumption, and will vary as that rate changes. Other variations will be introduced by the use of different fuels and different working conditions, than those which were fixed initially as a basis for the computations. However, there is no basic reason why the design of a furnace cannot be reduced to rational methods. When the method of operation and the design are fixed, it is possible to predict the fuel consumption within a reasonable margin, as well as the performance of the furnace as a heat-transfer apparatus. The main difficulty in obtaining accurate results with these computations—results which will check with practice—lies not in the computations, but in the

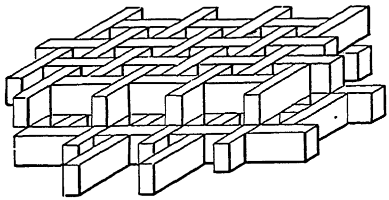


FIG. 166.—Method of Laying Up Checker Brick to Form Vertical Passes.

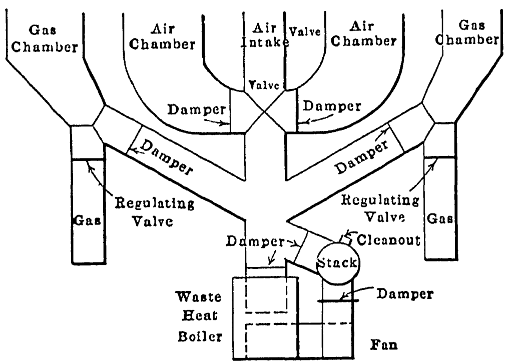


FIG. 167.—Schematic Arrangement of Flues and Dampers with Waste-heat Boiler.

initial assumptions, for unless the foundation is true the entire superstructure will be wrong. For instance, in computing the heat losses from the exterior walls of the furnace and chambers, the exposure to stray air currents and variations in the cycle of operation introduce complex variables which are difficult to consider. When the cycle is fixed and proper allowance made for the exposure, these losses may be reduced to mathematical analysis.

The frictional resistance in the checkers and flues may be approximated by the formula of Mojarow. This formula is

$$\gamma = m\frac{SL}{\omega}v_t\Delta_t,$$

in which γ = the frictional resistance expressed in inches of water or, if the metric system is used, millimeters of water or kilograms per square meter;

m = coefficient of friction, as determined by Mojarow = 0.003076 for English units or 0.016 for metric units. This coefficient was determined by observations on a Cowper hot-blast stove. Murge fixed a coefficient for mine galleries of one-third these values—namely, 0.001026 for English and 0.0053 for metric units;

S = perimeter of passage or summation of the perimeters of passages through which the gases flow;

L = length of the passage or flue;

SL = total surface of passage or flue in contact with flowing gases;

ω = cross-sectional area of the passage; or, where this varies, the mean sectional area; or, where the stream of gas does not fill the flue, the area of the flowing stream;

v_t = average velocity of flow per second at $t°$;

p_t = specific weight of a unit volume of gas at $t°$;

t = average temperature of the flowing gas in the section examined.

	Gas on Gas	Air on Air	Gas on Waste Gas	Air on Waste Gas
Frictional resistance in checkers:				
SL = heating surface = m^2 = ...	2560	4267.5	....	....
ω = total area of passes m^2......	11.664	19.440	....	....
$SL \div \omega$ = ...	219.5	219.5	....	....
v_t = velocity, m per sec.........	1.879	1.642	2.405	1.964
Δ_0 = specific weight gases........	1.125	1.293	1.320	1.320
$t°$ = average temperature.......	850°	750°	1200°	1100°
$1+\alpha t$ = gas factor for $t°$ = ...	4.1195	3.7525	5.4040	5.0370
$\Delta_t = \Delta_0 \div (1+\alpha t)$ = ...	0.2731	0.3445	0.2443	0.2621
γ = kg per m^2 friction loss				
$X = 0.016 \times 219.5 \times 1.879 \times 0.273$ =	1 kg 802	....	....	....
$0.016 \times 219.5 \times 1.642 \times 0.3445$ =	......	1 kg 987	....	....
$0.016 \times 219.5 \times 2.405 \times 0.244$ =	....	....	2 kg 064	....
$0.016 \times 219.5 \times 1.964 \times 0.2621$ =	....	....	....	1 kg 808

As the gases enter and leave the checkerwork they make a 90° change in their direction of flow, which is equivalent to a total absorption of their velocity head. The pressure necessary to impress upon the flowing gases their velocity in the checkers will be:

	Gas on Gas	Air on Air	Gas on Waste Gas	Air on Waste Ga
$h = v_t \div 2g$ = meters of gas olumn.	0.180	0.137	0.295	0.197
$\delta = h\Delta_t$ = kg per m^2 = ...	0. 49	0.047	0.072	0.052

	Gas Checker	Air Checker
Flues below checkerwork and walls:		
Chamber width	3 m 483	5 m 763
Number of 115-mm walls	5	9
Number of 465-mm flues	6	10

	Gas on Gas	Air on Air	Gas on Waste Gas	Air on Waste Gas
Frictional resistance in flues below checker work:				
$t°$ = temp. below checkerwork	500°	300°	800°	600°
$1+\alpha t =$	2.835	2.101	3.936	3.202
Q_0	5 m³ 32	8 m³ 51	5 m³ 19	7 m³ 58
$Q_t = Q_0(1+\alpha t) =$	15 m³ 08	17 m³ 88	20 m³ 43	24 m³ 27
B = width of flues, total	2 m 790	4 m 650	2 m 790	4 m 650
H = height of flues	1 m 000	....	....	....
Total length of chamber	7 m 663	....	....	....
The resistance to flow is about equivalent to the full volume for one-third of the length $L =$	2.554	....	....	....
S = perimeter of flues, total	17.580	29.300	....	....
ω = area of flues, total	2.790	4.650	....	....
SL = exposed surface m²	44.90	74.83	....	....
$SL \div \omega =$	16.09	16.09	....	....
Δ_0 = specific weight gas	1.125	1.293	1.32	1.32
$\Delta_t = \Delta_0 \div (1+\alpha t)$ = kg per m³	0.397	0.615	0.335	0.412
$v_t = Q_t \div \omega$ = m per sec	5.404	3.84	7.324	5.22
γ_t = friction loss in kg per m² =				
$0.016 \times 16.09 \times 5.404 \times 0.397 =$	0 kg 553	....	....	....
$0.016 \times 16.09 \times 3.840 \times 0.615 =$	....	0 kg 548	....	....
$0.016 \times 16.09 \times 7.324 \times 0.335 =$	....	....	0 kg 632	....
$0.016 \times 16.09 \times 5.220 \times 0.412 =$	....	....	....	0 kg 554
The pressures requisite to impress their velocity on the gases in the flues below the checker-work:				
$\delta = \Delta_t(v_t^2 \div 2g)$ = kg per m² =	0.590	0.460	0.916	0.570

	Gas on Gas	Air on Air	Gas on Waste Gas	Air on Waste Gas
Space required in chamber above checkerwork by Yesmann's formula, which was used in obtaining the height of the heating chamber				
For the case in hand:				
Q_0 = volume of gases per second	5 m³ 32	8 m³ 51	5 m³ 19	7 m³ 58
$t°$ = temperature............	1200°	1200°	1600°	1600°
$1+\alpha t$ = gas temperature factor....	5.404	5.404	6.872	6.872
$Q_t = Q_0(1+\alpha t)$ = ...	28 m³ 75	45 m³ 99	35 m³ 66	52 m³ 08
Q_t^2 = ...	827	2115	1272	2712
B = width of chamber = ...	3m 483	5 m 763	3 m 483	5 m 763
B^2 = ...	12.13	33.21	12.13	33.21
B^2t = ...	14,543	39,852	19,408	53,135
$Q_t^2 \div B^2t$ = ...	0.05686	0.05306	0.06555	0.05104
$\sqrt[3]{Q_t^2 \div B^2t} = X$ = ...	0.3845	0.3758	0.4032	0.3710
A = coefficient..............	3.40	3.53	3.40	3.53
$h_t = AX$..................	1 m 306	1 m 327	1 m 372	1 m 310
Rise of arch over chamber.......	0 m 420	0 m 700	....	....
Checker to skewback...........	1 m 180	0 m 900	....	....
Chamber above checkerwork:				
The frictional resistance to the flow of the gases will be about equivalent to the full volume for one-third the length				
m = coefficient of friction = ...	0.016			
S = perimeter = (2×3.483) + (2×1.250) =	....	....	9.466	....
L = length................	....	....	2.554	....
S = perimeter = (2×5.763) + (2×1.250) =	....	....	....	14.026
ω = area = 3.483×1.250 = ...	....	....	4.354	....
ω = area = 5.763×1.250 = ...	....	....	....	7.203
$SL \div \omega$ = 9.466 ÷ 4.354 = ...	....	....	2.174	....
$SL \div \omega$ = 14.026 ÷ 7.203 = ...	....	....	....	1.947
$v_t = Q_t \div \omega$ = m per sec.......	6.60	6.39	8.19	7.23
Δ_0 = specific weight gas......	1.125	1.293	1.32	1.32
$\Delta_t = \Delta_0 + (1+\alpha t)$ kg per m³...	0.2082	0.2393	0.1921	0.1921

	Gas on Gas	Air on Air	Gas on Waste Gas	Air on Waste Gas
γ = frictional resistance in kg per m^2 =				
$0.016 \times 2.174 \times 6.60 \times 0.208$ =	0 kg 048	....	....	....
$0.016 \times 1.947 \times 6.39 \times 0.239$ =	....	0 kg 048	....	....
$0.016 \times 2.174 \times 8.19 \times 0.192$ =	....	....	0 kg 055	....
$0.016 \times 1.947 \times 7.23 \times 0.192$ =	....	....	....	0 kg 043
γ may be considered as kg per m^2 or as mm of water				
The pressure required to create the air and gas velocity above the checkerwork =				
$\delta = (v_t^2 \div 2g)\Delta_t$ kg per m^2.......	0 kg 462	0 kg 498	0 kg 657	0 kg 512

The height from the bottom of the checker chambers to the port sill or bridge of the furnace can now be fixed from dimensions that have been determined. This permits the approximation of the chimney effect of the checkers and uptakes. The height will be found as follows:

Flues below checker chamber........................	1 m 000	3.280 ft
Checkerwork height..................................	7 m 000	22.963 ft
Checker to arch at center...........................	1.600	5.249
	9 m 600	31.492 ft
From bottom of arch to port sill....................	3m 000	9.844
H = total height to port sill........................	12 m 600	41.336 ft

Chimney Effect of Checkers and Uptakes	Gas on Gas	Air on Air	Gas on Waste Gas	Air on Waste Gas
$\delta = H(\Delta_{air} - \Delta_{gas}\, t°)$...........	....	....	....	....
$t°$ = average temperature.......	850°	750°	1200°	1100°
$1 + \alpha t$ = gas factor.............	4.1195	3.7525	5.4040	5.0370
Δ_0 = specific weight of gases....	1.125	1.293	1.32	1.32
$\Delta_{tgas} = \Delta_0 \div (1 + \alpha t)$ kg per m^3.....	0.273	0.345	0.244	0.262
Δ_{air} = specific weight of air.......	1.293	....	....	....
$\Delta_{air} - \Delta_{tgas}$ = kg per m^3..........	1.020	0.848	1.049	1.031
$\delta_1 = 12.60 \times 1.02$ = kg per m^2..	12 kg 851	....	....	....
$\delta_2 = 12.60 \times 0.848$ = kg per m^2..	....	10 kg 683	....	....
$\delta_3 = 12.60 \times 1.049$ = kg per m^2..	....	....	13 kg 217	....
$\delta_4 = 12.60 \times 1.031$ = kg per m^2..	....	....	....	12 kg 990

The gas and air necks, the cinder pockets and the uptakes are next in order. The efficiency of the cinder pocket depends upon the change in the direction of the flow, as well as upon the reduction in the velocity of flow. The width of the pockets fixes the taper of the necks. Both pockets will be made the same width, say, 2 m 00 (6.56 ft) with a space of 1 m 250 (4.10 ft) between them. The gas neck will be 3 m 00 (9.84 ft) wide where it leaves the chamber, tapering to 2 m 00 in a distance of about 3 m 750 (12.3 ft). The air neck will taper from a width of 5 m 00 (16.47 ft) to 2 m 00 (6.56 ft) in the same distance. These figures have been fixed arbitrarily and the assumption made that the same or greater areas will exist than above the checkerwork.

	Gas on Gas	Air on Air	Gas on Waste Gas	Air on Waste Gas
Air and gas necks:				
The frictional resistance will be based on the following:				
L=length of neck..........	3 m 750	4 m 000	....	....
S=perimeter average.......	9 m 000	11 m 000	....	....
SL=surface...............	33.75	44.00	....	....
ω=area in square meters....	5 m² 00	7 m² 00	....	....
$SL \div \omega$=....................	6.75	6.28	....	....
Q_t=volume of gases flowing..	28 m³ 75	45 m³ 99	35 m³ 66	52 m³ 08
$v_t = Q_t \div \omega$=m per sec.......	5 m 75	6 m 57	7 m 13	7 m 44
Δ_t=kg per m³ gas..........	0.208	0.239	0.192	0.192
γ=frictional resistance in kg per m²=				
$0.016 \times 6.75 \times 5.75 \times 0.208 =$	0 kg 129	....	....	....
$0.016 \times 6.28 \times 6.57 \times 0.239 =$	....	0 kg 144	....	....
$0.016 \times 6.75 \times 6.57 \times 0.192 =$	....	....	0 kg 147	....
$0.016 \times 6.28 \times 7.44 \times 0.192 =$	....	....	....	0 kg 144
The gas velocity in the necks is slightly less than in the chamber above the checkerwork and is assumed the same as in the cinder pockets. The pressure in kilograms per m² required to impress these velocities on the waste gases will be:				
$\delta = \Delta_t (v_t^2 \div 2g)$ = kg per m² = ...	0.354	0.525	0.495	0.543

The frictional resistances met with in the cinder pockets will be assumed as equal to those computed for the neck.

The uptakes will be proportioned as follows, and their frictional resistances computed:

	Gas on Gas	Air on Air	Gas on Waste Gas	Air on Waste Gas
Q_t = volume of gases flowing..	28 m³ 75	45 m³ 99	35 m³ 66	52 m³ 08
v_t = velocities assumed......	15 m/sec	10 m/sec	....	....
ω = area of uptake, total....	2 m² 00	4 m² 60	....	....
N = number of uptakes......	1	2	....	....
v_t = velocity, actual.........	14 m 4	10 m 00	17 m 83	11 m 32
S = perimeter (assumed)....	5 m 00	12 m 0	....	....
L = height = ...	3 m 00	....	....	....
$SL \div \omega$ = ...	7.50	7.83	....	....
γ = frictional resistance in kg per m² =				
$0.016 \times 7.50 \times 14.40 \times 0.208$ =	0 kg 360	....	....	....
$0.016 \times 7.83 \times 10.00 \times 0.239$ =	....	0 kg 300	....	....
$0.016 \times 7.50 \times 17.83 \times 0.192$ =	....	....	0 kg 411	....
$0.016 \times 7.83 \times 11.32 \times 0.192$ =	....	....	....	0 kg 274
The pressure necessary to impress their velocity upon the flowing gases will be:...........				
$\delta = \Delta_t(v^2 \div 2g)$ = kg per m²......	2 kg 199	1 kg 219	3 kg 111	1 kg 254

The assumption is made that the turn into the uptakes for the incoming gas and air absorbs the velocity in the necks and cinder pockets.

	Gas on Gas	Air on Air	Gas on Waste Gas	Air on Waste Gas
Frictional resistance of fantails for air and gas:				
S = perimeter (assumed)....	6 m 00	10 m 00	....	....
L = length................	5 m 00	6 m 00	....	....
ω = area..................	3 m² 00	5 m² 25	....	....
t = average temperature....	500°	150°	800°	600°
$1+\alpha_t$ = gas factor.............	2.835	1.5505	3.936	3.202

	Gas on Gas	Air on Air	Gas on Waste Gas	Air on Waste Air
Frictional resistance of fantails for air and gas:				
Q_0 = volume of gases........	5 m³ 32	8 m³ 51	5 m³ 19	7 m³ 58
$Q_t = Q_0(1+\alpha t)$ = ...	15 m³ 08	13 m³ 19	20 m³ 43	24 m³ 27
$v_t = Q_t \div \omega$ = m per sec.......	5 m 03	2 m 51	6 m 81	4 m 62
Δ_0 = specific weight gases....	1.125	1.293	1.32	1.32
$\Delta_t = \Delta_0 \div (1+\alpha t)$ = kg per m³..	0.397	0.834	0.335	0.412
$SL \div \omega$ = ...	10.00	11.43	10.00	11.43
γ = frictional resistance in kg per m² =				
$0.016 \times 10.00 \times 5.03 \times 0.397$ =	0 kg 320	....	....	....
$0.016 \times 11.43 \times 2.51 \times 0.834$ =	....	0 kg 383	....	....
$0.016 \times 10.00 \times 6.81 \times 0.335$ =	....	....	0 kg 338	....
$0.016 \times 11.43 \times 4.62 \times 0.412$ =	....	....	....	0 kg 359
The pressure necessary to impress their velocity upon the flowing gases will be:				
$\delta = \Delta_t(v_t^2 \div 2g)$ = kg per m²	0.512	0.268	0.795	0.454

	Stack Flue to Boiler or Chimney	Stack Flue from Air Valve	Stack Flue from Gas Valve
Frictional resistance of flues:			
L = length in meters................	5 m 00	4 m 00	8 m 00
S = perimeter in meters............	8 m 00	8 m 00	7 m 00
ω = area in square meters..........	4 m² 00	4 m² 00	3 m² 00
$SL \div \omega$ = ...	10.00	8.00	18.67
Q_0 = volume of gases flowing per sec...	12 m³ 77	7 m³ 58	5 m³ 19
$t°$ = average temperature...........	630°	550°	750°
$1+\alpha t$ = gas factor....................	3.312	3.0185	3.7525
$Q_t = Q_0(1+\alpha t)$ average volume gas....	42 m³ 29	22 m³ 89	19 m³ 48
$v_t = Q_t \div \omega$ = velocity m per sec........	10 m 58	5 m 72	6 m 50
Δ_0 = specific weight of flue gases.......	1 kg 32	....	....
$\Delta_t = \Delta_0 \div (1+\alpha t)$ = kg per m³ gas.......	0.399	0.437	0.352
γ = frictional resistance in kg per m² =			
$0.016 \times 10.00 \times 10.58 \times 0.399$ =	0 kg 676	....	....
$0.016 \times 8.00 \times 5.72 \times 0.437$ =	....	0 kg 320	.....
$0.016 \times 18.67 \times 6.50 \times 0.352$ =	....	....	0 kg 684
The pressure necessary to impress the velocity upon the gases in these flues will be:			
$\delta = \Delta_t(v_t^2 \div 2g)$ = kg per m² = ...	2.250	0.730	0.760

	Gas on Gas	Air on Air	Gas on Waste Gas	Air on Waste Gas
Valve resistances:				
Q_0 = volume flowing	5 m³ 32	8 m³ 51	5 m³ 19	7 m³ 58
$t°$ = average temperature	500°	50°	750°	550°
$1+\alpha t$ = gas factor	2.835	1.1835	3.7525	3.0185
$Q_t = Q_0(1+\alpha t)$ =	15 m³ 08	10 m³ 07	19 m³ 48	22 m³ 89
Valve size in inches	54 in.	66 in.	54 in.	66 in.
ω = valve area	1 m² 48	2 m² 21	1 m² 48	2 m² 21
v_t = velocity through valve = $Q_t \div \omega$ = m per sec	10 m 19	4 m 56	13 m 16	10 m 36
$h = v^2 \div 2g$ =	5 m 291	1 m 06	8 m 83	5 m 47
Δ_0 = specific weight gases	1.125	1.293	1.32	1.32
$\Delta_t = \Delta_0 \div (1+\alpha t)$ = kg per m³	0.397	1.093	0.352	0.437
As the gases make a 180° bend in passing through the valve, the velocity must be created twice; therefore, the pressure required will be:				
$\delta = 2h\Delta_t$ = kg per m²	4 kg 201	2 kg 317	6 kg 216	4 kg 781

Where changes of direction of flow occur, there is a loss of pressure, or, more exactly, additional pressure is required to impress the necessary velocity in the new direction of flow. In addition, where the gases flow from a larger passage into a smaller, their velocity of flow increases and additional pressure must be provided to impress this change in velocity upon the gases. The pressures necessary for this purpose must be approximated in the following manner:

	Gas on Gas	Air on Air	Gas on Waste Gas	Air on Waste Gas
Pressures required for impressing velocity and velocity changes upon flowing gases in kg per m^2 or mm of water: $\delta =$				
Uptakes	2.199	1.319	3.111	1.254
Cinder pockets	.....	.....	0.495	0.543
Necks 0.525 − 0.498 =	.....	0.027	.....	.....
Chamber above checkerwork	0.462	0.498	.....	.....
0.657 − 0.495 =	.....	.....	0.162	.....
Checkerwork	0.049	0.047	0.072	0.052
Flues below checkerwork	.....	.....	0.916	0.570
0.590 − 0.512 =	0.078	.....	.....	.....
0.460 − 0.268 =	.....	0.192	.....	.....
Fantails	0.512	0.268	.....	.....
δ total between valves and ports, kg per m^2 or mm of water	3.300	2.351	4.756	2.419
The friction losses $= \gamma$ in kg per m^2 between the valves and the ports are:				
Uptakes	0.360	0.300	0.411	0.274
Cinder pockets	0.129	0.144	0.147	0.144
Necks	0.129	0.144	0.147	0.144
Chamber above checkerwork	0.048	0.048	0.055	0.043
Checkerwork	1.802	1.987	2.064	1.808
Flues below checkerwork	0.553	0.548	0.632	0.554
Fantails	0.320	0.383	0.338	0.359
γ total between valves and ports, kg per m^2 or mm of water	3.341	3.554	3.794	3.326
Valve resistance $\delta =$ kg per m^2	4.201	2.317	6.216	4.781
$\delta =$ velocity pressures between valves and ports	3.300	2.351	4.756	2.419
$\gamma =$ friction losses between valves and ports	3.341	3.554	3.794	3.326
Summation, kg per m^2 or mm of water	10.842	8.222	14.766	10.526
Chimney effect of checkerwork, uptakes, etc.	12.851	10.683	13.217	12.990
Chimney effect available for port velocity, kg per m^2 or mm of water	2.009	2.461	.....	.....
Draft depression required at reversing valves, kg per m^2 or mm of water	.....	.....	27.983	23.516

	Gas on Gas	Air on Air	Gas on Waste Gas	Air on Waste Gas
Draft depression in kg per m^2 or mm of water at reversing valves..............	.....	.....	27.983	23.516
γ = frictional resistance in flue from valve, kg per m^2.......................	.....	.....	0.684	0.320
δ = velocity pressure in flue from valves, kg per m^2.......................	.....	.....	0.760	0.730
Total draft depression required at point of junction of flues.....................	.....	.....	29.427	24.566
γ = frictional resistance in flue to stack or boiler in kg per m^2 or mm of water..	.....	.....	0.676	.....
δ = velocity pressure required for stack flue $= 2.250 - 0.730 =$ kg per m^2...........	.....	.....	1.520	.....
Total draft depression at base of stack or inlet to boiler, kg per m^2 or mm of water	.....	.....	31.623	.....

In the foregoing, Mojarow's value for the coefficient of friction ($m = 0.016$) was used. This coefficient was derived from observations made on Massick and Crook and Cowper hot-blast stoves. It is probably correct where a considerable portion of the flue system consists of checkerwork. Murge, however, has determined the coefficients of friction of air at low temperatures in mine galleries. The work was done very carefully. The coefficient is one-third of that obtained by Mojarow, and is probably a closer approximation when the flues are large and open. A further confirmation of the fact that Mojarow's value is too high lies in the fact that, in several cases where furnaces in service have been analyzed, the computations showed the necessity for forced draft. These furnaces were in operation without forced draft, but in a more or less defective manner, giving operating troubles and having a shorter campaign than normal. Applying Murge's value for the coefficient of friction, the friction losses are reduced, as is also the total draft depression required. The revised values follow:

	Gas on Gas	Air on Air	Gas on Waste Gas	Air on Waste Gas
γ = friction loss between port and valves, revised, kg per m² or mm of water....	1.114	1.185	1.265	1.109
δ = pressure to impress velocities between port and valves, kg per m² or mm of water..........................	3.300	2.351	4.756	2.419
Valve resistance, δ = kg per m²..........	4.201	2.317	6.216	4.781
Summation, kg per m² or mm of water..	8.615	6.853	12.237	8.309
Chimney effect of checkerwork, etc., kg per m² or mm of water..................	12.851	10.683	13.217	12.990
Chimney effect available for port velocity.	4.236	3.830	.....	.....
Draft depression required at reversing valve	.....	.....	25.454	21.299

	Gas on Waste Gas	Air on Waste Gas
Revised figure for draft depression required at reversing valve, kg per m² or mm of water......................	25.454	21.299
γ = revised frictional resistance in flue from valve, kg per m² or mm of water................................	0.228	0.107
δ = velocity pressure required in flues from valves..........	0.760	0.730
Revised total draft depression required at point of junction of flues, kg per m² or mm of water......................	26.442	22.136
γ = revised frictional resistance in flue to stack or boiler, kg per m² or mm of water........................	0.226	.....
δ = velocity pressure required in stack flue, kg per m² or mm of water......................................	1.520	.....
Total draft depression required at base of stack or boiler inlet, kg per m² or mm of water....................	28.188	.....

The port areas and velocities may now be fixed. In so doing, it should be borne in mind that the waste gases must pass out through the gas- and air-ports in proper proportions to supply the required amount of preheat. This is very rarely accomplished when chimney draft is depended upon, particularly when the gas is given a high velocity as compared to the air. The angle necessary for the jet of flame to sinter in the bottom will be fixed by the resultant of the air and gas angles and velocities. That is, either the air or the gas may be given a maximum velocity; but any change in air velocity will entail a corresponding change in gas velocity, and *vice versa*, as well as a change in port angles, in order to produce the fixed resultant velocity and angle.

For instance, if the gas-port is lined on the resultant angle, the gas velocity will be maximum and the air velocity a minimum. The desirable condition is a velocity which will make the furnace hot throughout its full length. Frequently the trouble with overheated port blocks at the outgoing end is due to the high velocity of the incoming gases. With a high velocity at the incoming end a portion of the jet is practically shot into the outgoing port.

In most American furnaces, the air-port forms a segment over the gas-port; it is claimed that this construction forms a blanket of cooler air between the jet of flame and the roof. As the entering air is several hundred degrees cooler than the flame itself or the hot products of combustion, it is probable that this blanketing action is largely a matter of the imagination. Cooler gases have a tendency to sink below hotter gases; but any swift-moving jet has a tendency to hang together and will carry a layer of colder gases on top of it; these colder gases will have a tendency to move diagonally to the edges of the jet, while they are absorbing heat. The hottest gases, as soon as they escape from the influence of the jet, lose their velocity and tend to seek the roof of the furnace.

The air pressure available for creating velocity in the port is 3 mm 830 of water (0.15 in water column) due to the stack effect of the checkers and flues; it is not, however, advisable to utilize the full pressure available. In the case of the gas, 4 kg 236 (0.166 in of water) are due to the stack effect of the checkers and flues, but in addition there is the pressure created in the producer, which may be varied, amounting to one-half inch to one inch of water (12 to 25 kg per square meter). However, for every unit of pressure expended in creating velocity in the gas-port, a unit of pressure must be available at the outgoing port in order to pull a proportional part of the waste gases through the port. If this is not the case, the quantity of waste gases passing through the gas regenerator will be reduced and the amount of heat available for preheating the gas will be reduced.

Allowing 2 kg 50 for impressing velocity upon the air, the formula $\delta=(v_t^2 \div 2g)\Delta_t$ can be transposed, as follows:

$$v=\sqrt{\frac{\delta 2g}{\Delta_t}}.$$

For the case in hand, $\delta=2$ kg 50

$$\Delta_t=0.239$$
$$2g=19.62$$

and $v=14$ m 325 per second (47.00 ft per second)

The air and gas velocities have to be corrected by a coefficient, according to their relative masses and volumes. These coefficients are derived as follows:

$$\frac{5.32\times1.125}{8.51\times1.293}=\frac{5.9855}{11.003}=\frac{0.3523 \text{ for gas}}{0.6477 \text{ for air}}$$

Therefore, the value of the air component will be

$$14.325\times0.6477=9.28.$$

The area of the air-port will be

$$45.99\div14.325=3 \text{ m}^2\ 21=34.55 \text{ sq ft}.$$

Considerable latitude may be allowed in proportioning the gas- and air-port velocities and angles, to obtain the necessary resultant angle and velocity. Theoretically, the entire velocity may be supplied by either the gas or the air, or it may be divided between them. The ports may both be given the resultant angle, or the port angles may differ, provided the angular velocity components of the air and the gas give the angle and velocity necessary for the resultant. In practice there are certain structural limitations in regard to the location of the ports. These limits must be considered. In this case it will be assumed that the port angles are the same, being the resultant angle. The resultant velocity is 20 m 00 (65.62 ft) per second. Under these conditions the velocity component for the gas will be

$$20.00-9.28=10.72;$$

from which the gas-port velocity will be

$$10.72\div0.3523=30 \text{ m } 43 \text{ per second (99.84 ft per second).}$$

The gas-port area will be

$$28.75 \div 30.43 = 0 \text{ m}^2\ 945 = 10.17 \text{ sq ft.}$$

The gas velocity arrived at is the minimum for the assumed air velocity. If a higher gas velocity is used, the port angles will be changed; if a lower gas velocity is desired, it will entail a corresponding increase in the air velocity. There are an infinite number of combinations which will satisfy the resultant velocity and angle; but any change in one component will react upon the other. An apparently slight change, made without analyzing the conditions, is liable to lead to unexpected results. As there are, necessarily, certain angles and velocities which will give the best results, as well as others which give unreasonable results, the designer must use a certain amoun to fjudgment in coping with the problem. The pressure required to impress this velocity upon the gas will be

$$\delta = (v_t^2 \div 2g)\Delta_t$$
$$\delta = (30.43^2 \div 19.62)0.208 = 9 \text{ kg } 817 \text{ per m}^2$$
$$9 \text{ mm } 817 \text{ of water}$$
$$0.387 \text{ in of water}$$

The velocity of the waste gases passing through the gas-port, based upon the proportional division between the ports, will be

$$v = Q_t \div \omega$$
$$v = 35.66 \div 0.945 = 37 \text{ m } 74 \text{ per second}$$
$$124 \text{ ft per second}$$

The draft required will be

$$\delta = (37.74^2 \div 19.62)0.192 = 13 \text{ kg } 94$$
$$0.55 \text{ in of water}$$

The velocity of the waste gases, through the air-port, will be

$$v = 52.08 \div 3.21 = 16 \text{ m } 25 \text{ per second}$$
$$53.06 \text{ ft per second}$$

The draft pressure required will be

$$\delta = (16.25^2 \div 19.62)0.192 = 2 \text{ kg } 577$$
$$0.101 \text{ in of water}$$

The chimney effect of the gas checkers available for impressing velocity upon the entering gas was 4 kg 236 (0.166 in of water). The gas velocity requires a pressure of 9 mm 817 (0.304 in) of water), therefore the pressure to be supplied from the gas main will be

$$9.817-4.236=5 \text{ kg } 581$$
$$0.22 \text{ in of water}$$

The chimney draft required for removing the waste gases from the heating chamber will be

	Gas on Waste Gas
Draft required at flue junction............................	26 kg 442
Draft required for port velocity............................	13 kg 940
	40 kg 382
Draft in inches of water column............................	1.60

The total draft at the base of stack or waste-heat boiler inlet will be

$$40.382+0.228+1.520=42 \text{ kg } 130 \text{ per m}^2$$
$$42 \text{ mm } 413 \text{ of water}$$
$$1.67 \text{ in of water}$$

The height of the chimney will be based upon the assumption that the furnace will be workable by natural draft with the waste-heat boiler out of service. The temperature of the waste gases at the base of the stack will be 630° C. (1166° F.) and the volume of the waste gases 42 m^3 29 (1494 cu ft) per second at this temperature. The area of the chimney may be fixed by assuming a desired velocity of 10 m 00 (32.7 ft) per second. A velocity head of 5 m 10 (16.7 ft) will be required to impress this velocity upon the gases. The area of the stack may be approximated by the following formula:

$$Q_t=\kappa_1\kappa_2\omega\sqrt{2gH\frac{\Delta_{\text{air}}-\Delta_{\text{gas}}}{\Delta_{\text{gas}}}},$$

in which Q_t = the volume of gases flowing per second; for the case in hand, this is 42 m^3 29;

H = the head at the base of the stack, assumed as 5 m 10;

$2g$ = gravitational constant $=2\times 9.81=19.62$;

ω = the area of the stack in square meters, to be determined;

κ_1 and κ_2 = coefficients for velocity at contracted vein and the ratio of the area of the contracted vein to ω; for the case in hand these may be assumed as unity;

Δ_{air} = weight per cubic meter of air; for the case in hand $t=30°$ C. (86° F.), $1+\alpha t=1.110$, $1.293 \div 1.110 =$ 1 kg 165;

Δ_{gas} = weight per cubic meter of gases, $\Delta_0 = 1$ kg 320, $t=630°$ C. (1166° F.), $1+\alpha t = 3.3121$, $1.320 \div 3.312 = 0.399$.

The formula may now be written with numerical values, as follows:

$$42.29 = 1 \times \omega \sqrt{19.62 \times 5.10 \frac{1.165-0.399}{0.399}},$$

from which $\omega = 3.07$, say, 3 m² 14 (33.10 sq ft), giving a stack diameter of 2 m 00 (6.5 ft).

t = average temperature of gases in chimney..........	600° (1112° F.)
Q_0 = volume of waste gases at 0° (32° F.)..............	12 m³ 77
	451 cu ft
$1+\alpha t$ = gas factor for $t°$..............................	3.202
Δ_0 = specific weight of waste gases per cubic meter.......	1 kg 32
Pounds per cubic foot......................	0.0821
$Q_t = Q_0(1+\alpha t) = 12.77 \times 3.202 =$	40 m³ 89
	1440 cu ft
$\Delta_t = \Delta_0 \div (1+\alpha t) = 1.32 \div 3.202$ = weight per m³.........	0 kg 412
Pounds per cubic foot....................	0.0257
ω = area of chimney bore.........................	3 m² 14
	33.10 sq ft
Diameter of chimney bore = ...	2 m 00
	6.5 ft
v_t = average velocity of gases in chimney = $40.89 \div 3.14$ =	13 m 00 per sec
	42.8 ft per sec
t_{air} = temperature of external air, assumed to be.........	30° (86° F.)
Δ_{air} = weight per cu m of air at 30° (preceding operation)..	1.165 kg
Pounds per cubic foot....................	0.0725
$\Delta_{air} - \Delta_{gas} = 1.165 - 0.412$ = differential per m...........	0 kg 753
Draft depression required at base of stack = ...	42 kg 130
Entry pressure = 5.10×0.399 = ...	2 kg 040
Velocity pressure = $\delta = \Delta_t(v^2 \div 2g) = 0.412(13^2 \div 19.62)$ = ...	3 kg 550
	47 kg 720
First approximation of height of stack = $47.720 \div 0.753$ =...	63 m 50
	208 ft

Additional stack height will be required to cover the friction in the stack; this additional height may be approximated as follows: Assume about 5 m 00 in height required for this purpose; then

$$h=\frac{0.0053\times\frac{6.28\times68.50}{3.14}\times13.00\times0.412}{0.753}=5 \text{ m } 15.$$

The total height of the chimney will be

$$63 \text{ m } 50+5 \text{ m } 15=68 \text{ m } 65=225 \text{ ft}.$$

Additional chimney height might be desirable in order to provide a larger margin for increased draft resistance toward the end of a campaign, due to blocking up of the checkerwork. A different fuel, a different method of operation, or a reduction or increase in fuel consumption will modify these figures, but this method of analysis may be applied, not only to such cases, but likewise to the design of other types of regenerative furnaces.

In connection with the reversing valves, a factor of importance is the depth of the water seal. In the case of the gas valve, this seal is exposed upon one side to the pressure within the gas main, while the other is influenced by the draft depression at the valve. The air-valve seal is influenced by the stack depression. In the case of the gas valve, the gas pressure will be approximately 0.50 in of water and the draft depression about 1.50 in of water, making a total differential of 2.00 in of water. Gases, when their flow is interrupted, have a tendency to produce a rise in pressure, as water does. Therefore, with a 3-in water seal, only 1 in. of which is effective, there is the possibility of a blow-through.

Practically all reversing valves present more or less of a direct connection between the gas main and the stack flue or the air, during reversal, unless special dampers or other means are provided to prevent this. Some of the devices designed to operate dampers for this purpose are so arranged that the water seal on the valve is broken before the dampers have cut off either the gas main or the chimney flue. There is, however, no particular difficulty in arranging a mechanism or a valve which will completely cut the furnace and flues off from the stack and the air during reversal.

One of the main reasons why open-hearth progress has been very slow is that space around the valves is limited, and it is practically impossible to secure the necessary clearances to install different equipment or to provide the necessary rearrangement of the flues. This obstacle is somewhat difficult to overcome. The first installation is always difficult to secure, and in many cases it is the only one made.

A certain amount of producer gas will be lost every time the furnace is reversed. The gas regenerator and certain other parts of the furnace are full of gas prior to reversal. A portion of this trapped gas passes into the furnace and burns, but part of it will be drawn backward into the chimney flue. This loss cannot be eliminated. The more frequently the furnace is reversed, the greater the loss.

Wind exposure has a certain effect upon furnace operation. It affects the chimney draft appreciably. At the same time, it will have an effect upon the velocity with which the air enters the reversing valve, according to its exposure. Mr. Allyn Reynolds stated, at the 1913 meeting of the British Iron and Steel Institute, that a wind blowing at the rate of 20 miles an hour caused a variation in the rate of flow of air into the reversing valve of 70 to 350 ft per minute. The entry rate desired was 180 ft per minute. As variations of this kind are sudden and extremely irregular, it is difficult to compensate for them.

The use of a fan for introducing the air will not eliminate such variations entirely; the fan merely causes a motion of the air within itself, taking the air from the low-pressure side and delivering it to the high-pressure side. Any increase in the suction pressure will increase the delivery rate of the fan. A great advantage of a fan, in delivering the air, lies in the fact that it renders the furnace independent of the stack effect of the regenerators and uptakes in impressing the air velocity at the port, bringing this variable more thoroughly under control. If stove-type regenerators extending above the platform are used, a fan will be necessary to force the air through them.

Considerable heat is dissipated from the wall and roof surface of the regenerator chambers, the amount depending upon their exposed surface and the air or wind currents to which they are exposed. The proposal to insulate this surface has been actively considered. The problem is similar to that of the insulation of the hot-blast stove, except for the fact that the regenerator usually works at much higher temperatures than the hot-blast stove.

The heat loss takes place continually during both the heating and cooling cycles. This has suggested the possibility of recovering a portion of this heat by enclosing the chambers, with an air space all around them, in a sheet-iron housing arranged in such a manner that the air supply for the furnace is drawn from the highest and hottest portion of the enclosure. This housing should provide sufficient room to permit the inspection of the chamber walls.

All regenerative furnaces are more or less subject to explosions at reversal. These explosions crack and otherwise damage the brickwork, producing air leakage which reduces the temperature of the waste gases and increases their volume. The effect of this leakage is intensified when a strong draft must be used to pull the waste gases out of the furnace. The intensity of the draft depression increases progressively from the port to the base of the stack; and the seriousness of the air infiltration necessarily increases as the draft depression increases, the greatest amount of air entering the system through the walls of the flues leading to the chimney. Leakage outward undoubtedly occurs with both the air and the gas; but, as the pressure within the flues is extremely low, the volume lost is much less than the infiltration through the same wall. The only way of preventing air from leaking into the flues is to provide them with a covering of sheet steel of suitable thickness. This will, at the same time, necessitate a consideration of the fact that an explosion within a tight casing may be much more violent than one occurring in a brick flue. Such a casing and the heat insulation of these flues will increase considerably the amount of heat available for the generation of steam.

One of the factors brought out in the preceding computation was the draft depression necessary to pull a proportionate amount of the waste gases through the gas-port of the furnace. This indicates that a better-working furnace would result from having large uptakes through which the waste gases would pass with a low velocity down to the cinder pockets. A further advantage resulting from this increase in uptake area on the outgoing end will be a greatly increased efficiency of the cinder pocket in catching cinder and other material which is carried out of the furnace and thus increasing the life of the regenerators. The carrying power of a flowing stream varies as the sixth power of its velocity. That is, when the velocity is doubled, the weight of the particle carried will increase sixty-four-fold. When the waste gases pass down the uptakes with a high velocity they will not make a sharp turn at the cinder pocket, but will impinge upon its bottom, unless a considerable depth is provided. The formula of Yesmann, giving the trajectory of the jet of flame in the furnace, may be modified to give the distance vertically downward a flowing jet of gas will penetrate a cooler medium, as follows:

Metric: $$H=\frac{v^2}{2g}\times\frac{273+t_i}{t_m-t_i}$$

English units: $$H=\frac{v^2}{2g}\times\frac{459+t_i}{t_m-t_i}$$

The temperature of the gases leaving the heating chamber, t_m, is 1600° (2912° F.) and t_i, the temperature of the gases in the cinder pocket, is assumed as 1400° (2552° F.). The velocity of the waste gases in the gas and air uptakes has already been determined as 17 m 83 (58.5 ft) and 11 m 32 (37.14 ft) per second, respectively. When these values are substituted in the above formula, the following values of H will be found:

in gas cinder pocket $H=16$ m 94 (55.58 ft)
in air cinder pocket $H=6$ m 83 (22.41 ft)

Naturally, if the temperature of the immobile gases in the cinder pocket is lower than it is assumed to be, the penetration of the jet will be less as the difference in temperature increases. But reducing the velocity of the waste gases will decrease the value of H, according to the ratio of the squares of the velocity.

Both McKune and Egler have attacked this problem and produced head constructions which bring the air and the gas together in a rational manner at the incoming end and provide for an increased area of uptake at the outgoing end. Both of these methods are radical departures from previous head construction. The writer has devised a method of obtaining a similar result and at the same time presenting the possibility of adjusting the angle of the flame.

A large segmental air-port over a smaller gas-port is a construction much favored in American furnaces. The high gas velocity and the low air velocity result in a long flame. It was also considered that this form of port produced a blanket of cooler air on top of the jet of flame, and in this manner protected the roof. As the coldest gases tend to seek the lowest portion of the chamber, it is somewhat difficult to see just how this argument is tenable. At the same time the waste gases reaching this large port should be somewhat hotter than those reaching the smaller gas-port at a lower level. This difference in temperature might not be very great, as the vertical distance between the ports is not great.

However, a much larger proportion of the waste gases would be drawn off through this port, and, as a result, the air would be preheated to a higher temperature than the gas. With this construction the gas regenerator is strangled; that is, it cannot obtain the proportion of the waste gases necessary to preheat the incoming gas to the same temperature as the air supply, unless a considerable draft differential is available to force a correct division of the waste gases. A further disadvantage of this type of head is the tendency of the bath to chill at the incoming end with irregular reversals.

Port erosion, increasing the area of the gas-port and reducing the jet velocity, has been very troublesome, as a point is speedily reached at which the proper sintering of the bottom becomes impossible. This entails shutting down the furnace for repairs with consequent loss of production. Water-cooled ports are a partial cure for the trouble; they increase the time between port repairs, but they do not contribute to the correct division of the waste gases between the regenerators.

Another result of the improper division of the waste gases between the regenerators is a tendency for the furnace to work cold; that is, the time per melt is increased, owing to the improper combustion conditions produced. At the same time, the wear upon the furnace is greater, particularly upon the roof, gas ports, cinder line, etc.

The durability of any material exposed to heat is dependent upon its ability to conduct this heat away from the heated end and emit a sufficient amount of heat from its cool end to prevent the hot end from overheating. When the hot end commences to absorb heat faster than the cool end can emit heat, the temperature of the hot end will commence to rise until it fails.

Water cooling supplies a more rapid method of removing heat than air currents; it is particularly valuable in those cases where the refractory is subject to erosion or chemical action, as both of these forms of attack are more rapid at high temperatures. Reducing their temperature increases their resistance to such attack. Another function of water cooling is in reinforcing the rigidity of the binding of the furnace. Metal work exposed to heat will warp; cooling the metal prevents warping and holds the furnace to line. It is possible, however, to overdo the cooling; it, therefore, requires careful consideration, and the apparatus must be designed in such a manner that all steam and air pockets are avoided. With an open-hearth furnace, as with a blast furnace, cooling should only be used to hold vital points which cannot be held in any other way.

Thermal insulation for the conservation of heat will only be successful where the maximum temperature to which the refractories are exposed is less than their yielding temperature and where a sufficient thickness of the refractory is interposed to protect the insulation from temperatures above its yield point. In some cases the insulation will replace a certain amount of refractory brick, which serves to reduce the cost of the insulation; in other cases the insulation will be an addition to the cost of the refractories. The whole question must be settled on the basis of "*Will it pay?*" The balance between the cost of the thermal insulation and the value of the heat which it makes available for other purposes must not only pay the interest upon the investment, and supply funds for maintenance, etc., but must replace the capital invested. It is needless to say that the output of the waste-heat boiler will be increased by the delivery of high-temperature gases.

Waste-heat boilers are an indirect means of heat recovery. The main purpose of the furnace is the production of steel, and auxiliary apparatus cannot be permitted to interfere with this. Considerable diversity of opinion exists in regard to waste-heat boiler design and installation, the type of boiler and the method of baffling.

High gas velocities have been considered necessary for rapid heat transfer from gas to water. Such velocities mean a considerable draft differential through the boiler and are based upon the idea that, at high velocities, stream-line flow of the gases is replaced by confused eddies. Stream-line flow cannot occur with gases which are in contact with surfaces hotter or cooler than themselves. The difference in density created by temperature changes creates local recirculating loops which eliminate streamline flow.

A good idea of these confused currents may be gained by observing any current of hot air rising alongside of a hot furnace. These currents may be rendered visible by using a bright light which will be partially polarized by the eddies created, causing them to cast a shadow. A light which contains an appreciable proportion of the blue end of the spectrum will render visible the heat waves beyond the visible red of the spectrum.

In a gas-to-fluid heat transfer through a metal partition, there are two very important factors, either one of which will limit the value of the test. The heat transmission of the metal will be limited by the manner in which the fluid circulates past the wet surface, to a much greater degree than it will be limited by the manner in which the hot gases circulate past the dry surface. Practically all the recent experiments regarding gas-to-fluid heat transfer through metal have entirely neglected the part played by the fluid in carrying off the heat. Until the fact is recognized that the gases cannot transfer heat to the metal any faster than the water, in turning to steam, is able to carry it away from the metal, very little progress will be made in boiler design.

In metallurgical furnace work it has long been recognized that there are limitations upon the rate of temperature drop. In this work there is frequently only a small temperature differential between the gas giving up heat and the material to which heat is imparted. In the steam boiler the temperature differential between the hot gases and the water turning into steam is very large, and the main obstacle to a high rate of heat transfer is the poor arrangement of the water circulation.

Table 20 gives the quantity of heat available for a waste-heat boiler at various initial and final temperatures, together with the drop in temperature, initial and final gas volumes and the change in the gas volume due to the drop in temperature. These values clearly illustrate the large amount of heat lost by the drop in temperature between the regenerator and the boiler and carried away from the boiler by high waste-gas temperatures. With leaky gas flues there is not only the drop in gas temperature, but the added volume of air, which may increase the volumes to be dealt with by 30 to 40 per cent, or more.

The question of waste-heat utilization must be considered upon an economic basis. A further factor to be taken into account is the question of dividing the heat, which leaves the laboratory of the furnace, between the regenerators and the waste-heat steam generator. The higher the temperature at which the gases are passed to the boiler, the greater its steam-generating capacity; there is thus a possibility of reducing the cost of the regenerator. The whole question of the design of the open-hearth shop and its equipment is a matter of compromise and of balancing one thing with another in order to secure a desired result—ingot tonnage of the desired quality, at a profit.

In the foregoing discussion, producer gas has been the only fuel considered. The fuel question depends largely upon local conditions for the particular plant. Natural gas and coke-oven gas eliminate the producer plant; but in many localities natural gas is becoming scarce and coke-oven gas is not available at a price and in a quantity which will permit its use. Tar is used in a few plants, in order to get rid of the enormous quantities produced by the by-product coking plants. Water gas has been used, as well as blast-furnace gas mixed with producer or some other gas; but the use of these last was more or less forced by war conditions. Pulverized coals are used, as is also oil, which may be atomized by steam or compressed air, or by mechanical means. Pulverized coal adds a certain proportion of its ash to the normal cinder, while finer portions are carried further and about 25 per cent of ash passes out of the stack suspended in the gases. With most of these fuels, only the air is preheated.

One of the most important elements in the fuel for an open-hearth is sulphur. The less sulphur present the better; it has a tendency to pass into the cinder and metal, under certain conditions, adding to the expense and time of the melt.

TABLE 20

Temperature, Degrees C.			Volume of Waste Gas, Cubic Meters			Heat Available, Calories per Second		
Initial	Final	Drop	Initial	Final	Change	Initial	Final	in Boiler
900	400	500	54.91	31.54	23.37	4336	1846	2490
	350	550		29.11	25.80		1622	2714
	300	600		26.82	28.09		1383	2953
	250	650		24.52	30.39		1144	3192
800	400	400	50.31	31.54	18.77	3814	1846	1968
	350	450		29.11	22.20		1622	2192
	300	500		26.82	23.49		1383	2431
	250	550		24.52	25.79		1144	2670
700	400	300	45.59	31.54	14.05	3400	1846	1554
	350	350		29.11	16.48		1622	1778
	300	400		26.82	18.77		1383	2017
	250	450		24.52	21.07		1144	2256
600	400	200	40.87	31.54	9.33	2814	1846	968
	350	250		29.11	11.76		1622	1192
	300	300		26.82	14.05		1383	1431
	250	350		24.52	16.35		1144	1670

TABLE 20—*Continued*

Temperature, Degrees F.			Volume of Waste Gas, Cubic Feet per Second			Heat Available, B.t.u. per Second		
Initial	Final	Drop	Initial	Final	Change	Initial	Final	in Boiler
1652	752	900	1939	1114	825	17,206	7326	9,880
	662	990		1028	911		6337	10,869
	572	1080		947	992		5488	11,718
	482	1170		866	1073		5450	12,666
1472	752	720	1777	1114	663	15,134	7326	7,808
	662	810		1028	749		6337	8,797
	572	900		947	830		5488	9,646
	482	990		866	911		4540	10,594
1292	752	540	1610	1114	496	13,492	7326	6,166
	662	630		1028	582		6337	7,155
	572	720		947	663		5488	8,004
	482	810		866	744		4540	8,952
1112	752	360	1443	1114	329	11,168	7326	3,842
	662	450		1028	415		6337	4,831
	572	540		947	496		5448	5,680
	482	630		866	577		4540	6,628

TABLE 20—*Continued*

Base Data:

Q_{og} = volume of gas burned per second = 5 m³ 32 = 188 cu ft.

Q_{owg} = volume of products of combustion with 40 per cent excess air = 12 m³ 77 = 451 cu ft.

Temperatures, initial, C.	600°	700°	800°	900°
F.	1112°	1292°	1472°	1652°
Gas factor, $1+\alpha t$	3.20	3.57	3.94	4.30
Temperatures, final, C.	250°	300°	350°	400°
F.	482°	572°	662°	752°
Gas factor, $1+\alpha t$	1.92	2.10	2.28	2.47

Heat capacity of products of combustion from curve for producer gas, SC–DSW, Fig. 184:

Calories, initial	2,814	3,400	3,814	4,336
B.t.u.	11,168	13,492	15,134	17,206
Calories, final	1,144	1,383	1,622	1,846
B.t.u.	4,540	5,488	6,337	7,326
Volumes, initial, m³	40.87	45.59	50.31	54.91
cubic feet	1,443	1,610	1,777	1,939
Volumes, final, m³	24.52	26.82	29.11	31.54
cubic feet	866	947	1,028	1,114

APPENDIX VIII

DESIGN OF HOT-BLAST STOVES

By A. D. WILLIAMS

A. E. Maccoun, in his paper before the American Iron and Steel Institute, May 28, 1915, showed the temperature distribution and the approximate isotherms for a Cowper stove at the Edgar Thomson Furnaces. The isotherms, being those taken immediately after the stove went on gas and just before going on blast, indicate that the hot gases, in cooling, tend to flow through those passes which lose the most heat, those closest to the shell of the stove, and that the blast, in heating, tends to pass through the hottest passes, those in the central portion of the stove. This distribution follows natural laws and cannot be changed except by the application of sufficient insulation around the outside of the brickwork to reduce the amount of heat lost through the shell of the stove. Changing the height of the bridge wall or the dome of the stove, or increasing the number of chimney valves, will not affect this distribution, provided, of course, that the one chimney valve has sufficient area. Strangulation by insufficient valve area is a common fault, not only in hot-blast stoves, but in many other types of furnaces.

In a paper before the American Iron and Steel Institute in October, 1916, Arthur J. Boynton gave a number of illustrations of current designs in hot-blast stoves, the number of passes varying from two to four. This paper and its discussion seemed to indicate that the laws governing the subdivision of streams of heating and cooling gases were not clearly understood, and this was also shown in the stove designs illustrated. The laws governing the flow of heat and the relationship between the thickness of the wall of the checker openings and the time required to saturate the heat-storage capacity of the brickwork are not understood. The checker wall thickness ranged from 1.5 inches (38 mm) to 3 inches (75 mm), with variations in pass diameter.

Tables No. 1 and No. 2 were approximated from a curve showing the temperature on the central line of a wall of firebrick, in percentage value of the temperature of the surface of the brick, which was given on page 222, Appendix VII. The curve is based upon a flat wall. In applying this to brickwork built up to form square passes it is necessary to allow for the effect of the corners where a square column of brick is formed. As the ratio between the diagonal and the side of a square $=1.41=\sqrt{2}$, the existence of such columns will double the heating time. These times are only approximate and would not apply to large passes with thin walls between, nor to small passes with thick walls.

Table No. 3 shows the relationship between the brick and the free area for different ratios of pass and wall thickness. As a general rule the stove is on gas from two to three times as long as it is on air. Therefore the cooling period will determine the volume of the brickwork and the checker-pass volume required. Tests made by the Bureau of Mines (Bulletin No. 8, "The Flow of Heat Through Furnace Walls"), indicate a transfer drop in temperature from a hot gas to a brick wall of less than 150°. Numerous other data indicate a temperature drop in a gas-to-gas transfer of heat through checkerwork of 300°. This latter drop is less than that shown in the Cowper stove tested by Mr. Maccoun, where the blast temperature was 650° and the gas temperature around 1200°, a drop of 550.°

Table No. 4 consists of data abstracted from the test of a two-pass Cowper stove at the Edgar Thomson Furnaces in 1913 and contained in Mr. A. E. Maccoun's paper before the American Iron and Steel Institute on May 28, 1915, and the computations based upon these data.

The combustion chamber has an area of 3 m^2 79 and a height above the burner of 23 m 10, giving a volume of 87 m^3 54. The gases in this chamber have a very high temperature, about 1220°. Assuming their specific weight to be equal to that of air, this temperature would be sufficient to give them an ascensional velocity of

$$v_{1200}=\sqrt{2gH\frac{1.29-\dfrac{1.29}{1+\alpha t}}{\dfrac{1.29}{1+\alpha t}}}=44\text{ m }70\text{ per second,}$$

which would carry the gas to the dome in half a second. The average time the gases remain in this chamber as computed from their average temperature in the stove will be about five seconds. The volume of gases formed each second in the combustion chamber is

$$4.31\times5.404=23\ \mathrm{m}^3\ 29 \text{ with 40 per cent excess air,}$$
$$4.75\times5.404=25\ \mathrm{m}^3\ 67 \text{ with 80 per cent excess air,}$$

which would make the average area of the ascending column less than 0 m² 70 or less than 20 per cent of the combustion chamber area. The gas shoots across the short diameter of the chamber and strikes the wall with considerable force. As a result of these conditions, the flame in the combustion chamber may take on a resonant vibration similar to that of the " singing flame " of the physical laboratory.

The mushrooming of the burner jet is one of the causes for the flame coming out around the burner. The hot column of blazing gas will rise through the stagnant gases in the combustion chamber and, since convection currents for gas are readily set up there will be considerable recirculation. Noisy burning is a well-known feature of stove operation. The mushrooming of the gas jet assists in the mixing, and further mixing is effected when the gases reach the dome. Mr. Maccoun's test proved that unburned gases reached the dome but did not pass down into the checkerwork.

The draft conditions existing in a stove are peculiar, inasmuch as there may happen to be a greater draft depression at the chimney valve than at the foot of the stack. This arises from the fact that upward pressure exists in the combustion chamber and likewise, to a lesser extent, in the checker, and these pressures counterbalance each other. Assuming for the sake of simplicity that the waste gases, etc., have the same specific weight as air, the upward pressure in the combustion chamber will be

$$\Delta_{1200}=23.10\left(1.29-\frac{1.29}{1+\alpha t}\right)=+26.20 \text{ mm of water.}$$

In the checker this upward pressure will be

$$\Delta_{740}=26.10\left(1.29-\frac{1.29}{1+\frac{1200+280}{2\times273}}\right)=24.60 \text{ mm. of water.}$$

	Mm. of Water Column
Upward pressure in combustion chamber	+26.20
Upward pressure in checkerwork	+24.60
Difference or positive pressure acting at chimney valve	+ 1.60
Draft depression at chimney valve	−38.00
Draft depression at chimney valve due to stack will therefore be...	−36.40
Upward pressure in checker	+24.60
Draft depression in dome of stove due to chimney	−11.80
Upward pressure in combustion chamber	+26.20
Making a draft depression at the burner available to draw in the air required for combustion	−38.00

Professor Groume-Grjimailo has suggested that combustion conditions in the two-pass Cowper stove may be improved by springing an arch across the chamber with an opening proportioned to retard the flow of the gases to such an extent that combustion occurs in the chamber. The location of the arch will be fixed by the amount of gas burned and the time required for combustion. Two seconds may be allowed for combustion, which will be very rapid under incandescent chamber conditions, with good mixing. The size of the chamber and the port may be arrived at as follows:

$$\text{Chamber volume} = 25.67 \times 2 = 51 \text{ m}^3\ 34,$$
$$\text{Height of chamber} = 51.34 \div 3.79 = 13 \text{ m } 60.$$

It is necessary to provide an opening in the arch over the top of the combustion chamber which will permit the waste gases to escape and yet restrict their flow sufficiently to supply time for the completion of combustion. The area of this opening may be determined by the use of Yesmann's formula, as follows:

$$Q = \kappa_1 \kappa_2 \omega \sqrt{2gH \frac{\Delta_{\text{air}} - \Delta_{\text{gas}}}{\Delta_{\text{gas}}}},$$

in which Q = the quantity of gas flowing, in cubic meters per second; for the case in hand, $Q_{1200} =$ 25 m^3 67;

H = the head at the orifice or the vertical distance from the orifice to the lower free surface of the gases; for the case in hand $H = 13$ m 60;

$2g$ = gravitational constant $= 2 \times 9.81 = 19.62$;

Δ_{air} = specific weight of air = 1 kg 29 per cubic meter;
Δ_{gas} = weight of gases at combustion-chamber temperature; in this case $t=1200$ and $1.29 \div 1+\alpha t = 0$ kg 239;
$\kappa\omega$ = area of orifice in square meters;
κ_1 = coefficient of contraction of the jet, or the ratio between the area of the contracted vein and ω (in hydraulics, the contracted vein for a circular orifice is about 0.67ω);
κ_2 = velocity coefficient, or the ratio between the actual velocity in the contracted vein and the theoretical velocity (in hydraulics, the velocity in the contracted vein is about $0.97v$).

κ_1 and κ_2 have not been accurately determined for gases. For the case in hand they may be assumed as equal to unity, and the formula becomes, for the particular case,

$$25.67 = 1 \times 1 \times \omega\sqrt{19.62 \times 13.60 \times \frac{1.29-0.239}{0.239}} = 0 \text{ m}^2\ 749.$$

The velocity through the orifice will be

$$25.67 \div 0.749 = 34 \text{ m } 27 \text{ per second.}$$

A combustion chamber of this kind might be constructed in an old stove, but it would be difficult to maintain with dirty gas. The furnace dust has a fluxing tendency upon the brickwork and in many cases forms a heavy deposit. Mr. R. J. Wysor, in his discussion of Mr. Boynton's paper, presented some interesting photographs showing the fluxing action experienced in certain furnaces at South Bethlehem. The fume at these furnaces is high in alkali.

The stove tested was provided with three chimney valves, being operated in some runs with all three and in others with only two. The valves were 20.5 inches in diameter, giving an area of 0 m² 212 each. The average temperature at the chimney valve was about 280°. The volume of gas flowing per second, $Q_{280} = 4.75 \times 1+\alpha t = 9$ m³ 63; therefore the velocities would be

$$\text{with 3 valves, } v = \frac{9.63}{3 \times 0.212} = 15 \text{ m } 10 \text{ per second,}$$

$$\text{with 2 valves, } v = \frac{9.63}{2 \times 0.212} = 23 \text{ m } 70 \text{ per second.}$$

The pressure, in millimeters of water or kilograms per square meter, required to impress this velocity upon the gases would be:
With 3 valves:

$$\delta_2 = \frac{15.10^2}{2g} \times \frac{1.29}{1+\alpha t} = 7 \text{ kg } 392 \text{ per square meter;}$$
$$= 7 \text{ mm } 392 \text{ of water.}$$

With two valves:

$$\delta_2 = \frac{23.70^2}{2g} \times \frac{1.29}{1+\alpha t} = 18 \text{ kg } 22 \text{ per square meter;}$$
$$= 18 \text{ mm } 22 \text{ of water.}$$

The draft depression at the chimney valves was -38 mm of water. Therefore, from 20 to 48 per cent of the draft at these points was required to supply a sufficient velocity to remove the waste gases. This would seem to indicate insufficient valve area. Two valves tend to strangulate the stove, but with three the strangulation is lessened. The chimney valve is a weak point; any leakage of blast at this point is effectually concealed. A multiplicity of valves increases the opportunity for valve defects, but at the same time permits blanking a valve, if necessary, without shutting down the stove for any length of time. Large valves are costly, but one large valve reduces the number of potential leakage points. The lowered velocity reduces the draft required in proportion to the square of the velocity, and the friction and other losses are likewise reduced.

The distribution of the gases through the checkerwork will not be affected by the number of chimney valves, dampers and partitions, but by the friction, heat loss, etc. The rapidity with which the current of gas gives up its heat in any particular checker-pass will determine the quantity of gas flowing in that pass. A mathematical demonstration of this is given in Groume-Grjimailo's work, and the distributions of the isotherms, as shown in Fig. 6 of Mr. Maccoun's paper, confirm this. The natural velocity of convection currents is higher than is generally realized. Table No. 8, Appendix VI, gives the convection velocities for various average temperature differentials acting through heights of 0.10, 1.00 and 10.00 meters. The checkerwork is 24 meters high; the velocity average for the blast and the hot gases is 2 m 63 and 2 m 41 per second, which is less than the convection velocity for an average temperature differential of 10° acting for 10 m 00. The temperature differential for the blast in the checker is about 580° and of the gases about 940°. Through the checkerwork the convection currents act in the same direction as the flow of the blast and the gases. Therefore the convection circulation tends to make every particle of the gases brush every unit of surface. This fact was demonstrated many years ago by Peclet at the *Institut des Arts et Métiers*, at Paris, but is still ignored by many.

In the combustion chamber the convection currents will be in the reverse direction to the flow of the blast and the gases. For that reason, it is probable that temperatures taken slightly below the top of the dome would show a higher blast temperature than would be found at the hot-blast valve, lower down.

The study of Table No. 4 reveals several interesting facts in regard to the two-pass stove which was tested. Hot-blast stove temperatures are comparatively low, both as regards the temperatures realized from the burning of the gas and the hot-blast temperature. In this stove the gas-to-gas differential was about 600°, which is nearly double that for other types of regenerators. The rate of heating and cooling, that is, the average temperature change per second in the blast heating and the gases cooling is low. The checker contains 28 m^2 32 of heating surface per cubic meter of free air per second, or, if the average volume of the blast in the checker is considered, this ratio becomes 24 m^2 35. The rate of heat absorption by the blast averages 0.659 calories per square meter of heating surface per second. The fact that the isotherms at the upper end of the checker are much closer together than those lower down may be interpreted as indicating that this end of the checker worked too cold, due, possibly, to the greater heat loss through the walls and shell of the stove.

The usual thickness of the outer walls of stoves is 18 inches of brick, 450 mm, with an air space of from 1 to 2 inches. A cyclic change in temperature occurs in the inner portion of this wall while the outer portion will give a temperature gradient toward the air space. Due to the fact that the heating and cooling periods are not equal and there is a constant heat or temperature loss through the wall, the analysis of this cycle is not simple. The central portion of the wall has a small temperature change. There is a constant flow of heat to the outer surface and a periodic flow from and to the inner wall surface, according to whether the stove is at the beginning or end of the heating or cooling period. The heat loss at the top of the checkerwork will naturally be several times as rapid as at the cooler end. Hence, if heat insulation is applied, it is necessary to use a greater thickness at the top. In determining the amount of insulation required, a study of the temperature cycle of the wall should be made, as it will throw an important light upon the subject.

The frictional loss through the checkerwork may be approximated by the formula developed by W. A. Mojarow and given in the *Revue de Métallurgie* for May, 1914, page 320.(1) This formula is as follows:

$$\gamma = m\frac{SL}{\omega}v_t p_t.$$

Applying this formula to the conditions existing when the stove is on air, the values of the quantities are as follows:

m = coefficient of friction determined by Mojarow as = 0.016 per meter, by observations made upon Cowper stoves;

SL = the heating surface, S, being the perimeter of the passes and L their length, for this case = 4738 m² 00;

ω = the area of the passes = 7 m² 40;

v_t = average velocity of gas = 2 m 63 per second;

p_t = weight of the gas at temperature and pressure = 1 kg 115 per cubic meter for air at 1 atmosphere and an average temperature of 360°.

Substituting these numerical values in the formula, the frictional resistance to the passage of the blast is found to be equal to 30 mm of water.

The following computations have been made with a view to the possibility of obtaining higher blast temperatures, say, 900° (1650° F.), with the same volume of blast as the stove tested (16 m³ 78 of free air per second).

Blast temperatures: Maximum of hot blast....	900°
Cold-blast main..........................	70°
Rise in temperature......................	830°
Average temperature......................	485°

(1) Refer to Appendix IV.

The heat capacity of the blast, per 100 molecular volumes, is obtained from data in Table No. 8, Appendix X.

Heat capacity of hot blast $t=900°$......	660 calories	
Cold blast $t=70°$......	50	"
Absorbed by blast in an increase of 830°.	610	"

The total amount of heat absorbed by the air in heating, during each second the stove is on air, will be

$$(16.78\times 44.80\div 100)\times 610=4586 \text{ calories per second.}$$

(The value 44.80 is the number of molecular volumes of 22.32 liters in 1 m^3.)

With a cycle of one hour upon air, the total amount of heat carried away from the stove by the blast will be

$$4586\times 3600=16{,}590{,}000 \text{ calories.}$$

In making the following computations, no allowance has been made to cover the heat lost through the walls of the stove or the necessary volume of gas to be burned to supply this heat. The amount of gas required will depend in part upon the heat loss and the insulation applied to prevent such loss, upon whether the gas and air from its combustion are preheated or not, upon the design of the combustion chamber, etc. The possibilities of hot-blast stove design have not been as well appreciated as they might be. Detail improvements in burners and valves, with their comparatively slight opportunity for improving the results obtained in producing hot blast, have obscured the greater possibilities of increasing the temperature of the blast by rational construction. Multiple-pass stoves have comparatively little excuse for existence. The main advantage of an even number of passes is the location of valves, etc., near the bottom of the stove. Three-pass stoves are not only irrational in design but require that valves be located near the top of the stove, where trouble is not only difficult to detect but hard to remedy.

Assuming that a temperature change of 100° in the checker brick is permissible, the weight and volume of brick required will be

$$\frac{16{,}590{,}000}{100\times 0.25}=663{,}600 \text{ kg}\div 1800=368 \text{ m}^3\ 80.$$

These values do not allow for heat loss; that is, this volume of brickwork, when heating, would have a temperature change from 20° to 50° greater than when on blast, as the 100° change is the change covering the heat transferred to the blast.

The temperature of the blast increases 830° in the checker. The average rate of increase assumed will determine the time allowed for heating and, with the blast volume, the space required in the checkerwork. Assuming a heating rate of 100° per second, the time necessary will be 8.3 seconds. Slower or faster rates of heating will correspondingly affect the time as well as the space required to contain the blast while heating. The volume of free air blown per second is 16 m³ 78, its average temperature in the checker is 485°, and it is under a pressure of one atmosphere, the volume under these conditions being 23 m³ 33, which, multiplied by 8.3, the heating time, fixes the space required as 193 m³ 70. The total checker volume will be, therefore, 193.70+368.80= 562 m³ 50.

The brick coefficient = 368.80 ÷ 562.50 = 0.6557.
The pass coefficient = 193.70 ÷ 562.80 = 0.3443.

The side of the square for the pass unit $= \sqrt{0.3443} = 0.5868$.

The portion of the unit square occupied by the brickwork will be 1.0000 − 0.5868 = 0.4132.

As the cooling time of the stove is one hour, it is not desirable to make the wall thickness between the passes greater than 75 mm; therefore the side of the unit square will be 75 ÷ 0.4132 = 181.5 mm or, say, 180 mm. The diameter of the square pass will be 180 − 75 = 105 mm = 4.125 ins.

The area occupied by a checker unit	$= 0.180^2 =$	0 m² 0324
The area occupied by the pass	$= 0.105^2 =$	0 m² 0110
The area occupied by brick	=	0 m² 0214

The lineal amount of checker required, based upon the pass = 193.70 ÷ 0.0110 = 17,609 m, or, if based on the brick = 368.80 ÷ 0.0214 = 17,230 m. Using the largest of these values and assuming a checker height of 25 m, the number of passes = 17,609 ÷ 25.00 = 704.

The total area occupied by the passes	$=0.0110\times$ 704 =	7 m^2 75
The total area occupied by brick	$=0.0214\times$ 704 =	15 m^2 07
Checker area	$=7.75\ +15.07=$	22 m^2 82
The perimeter of one pass	$=0.105\ \times$ 4 =	0 m 42
The total perimeter	$=0.420\ \times$ 704 =	295 m 68
The heating surface	$=295.68\times$ 25 =	7392 m^2 00
Heating surface per cubic meter of free air per second	$=7392\ \div 16.78=$	44 m^2 05
Heating surface per cubic meter at pressure and average temperature	$=7392\ \div 23.33=$	31 m^2 68

The average heat transfer rate will be $4586 \div 7392 = 0.6204$ calories per square meter per second for the existing differential.

The average velocity of flow of the air through the checker will be $23.33 \div 7.75 = 3$ m 01 per second.

The frictional resistance in the checkers by Mojarow's formula will be 42 mm 80 of water, while there will be a hydrostatic pressure upward of 9 mm of water, so that the drop through the checker will be 33 mm 80 of water.

The possibility of increase in blast temperatures is limited by the possible temperature which may be realized from the combustion of blast-furnace gas. Mallard and Le Chatelier, in the course of their work for the *Commission de Grisou,* determined the heat capacity of various gases. Working with these data, it is possible to approximate the instantaneous calorific intensity of combustion of various fuels under athermal chamber conditions, as well as the effect of preheating the air or the gas, or both. Such a curve has been computed for a blast-furnace gas having a volumetric composition of $H_2 = 3.92$, $CO = 23.95$, $O_2 = 0.39$, $CO_2 = 12.96$, $H_2O = 1.65$ and $N_2 = 57.13$. This gas has a thermal value of 1861 calories per 100 molecular volumes (2 m^3 232) equivalent to 835 calories per cubic meter and 93 B.t.u. per cubic foot. This curve (Fig. 181) shows intersection points which may be read to approximate the temperature which may be realized from the combustion of this gas with various air supplies and preheating conditions.

Table No. 5 gives computations, which are believed to be self-explanatory, for the plotting of the curve. In making them, however, it has been assumed that the air is composed of 1 volume of O_2 and 4 volumes of N_2. While this is not absolutely accurate, it is sufficiently close for the purpose and saves considerable time in computing. Table No. 6 summarizes the instantaneous calorific intensities as read from the curves for different conditions. Tables 6 and 8, Appendix X, give the data used in the computations of Table No. 5, both being abstracted from *Les Sources de L'Énergie Calorifique* by Damour, Carnot and Rengade.

TABLE 1

TEMPERATURE ON CENTER LINE OF CHECKER WALL LAID UP IN STRAIGHT PASSES IN PER CENT VALUE OF THE SURFACE TEMPERATURE OF THE BRICK. ALLOWING FOR THE CORNER EFFECT

Heating Period in Minutes, Wall Thickness		15	30	45	60
Inches	Millimeters	Temperature on Center Line in Per Cent of Surface Temperature			
2.0	50	99.0	99.9	99.99	....
2.5	63	94.0	99.5	99.9	....
3.0	75	91.0	99.0	99.5	....
3.5	90	80.0	97.0	98.5	99.8
4.0	100	70.0	93.0	98.0	99.5
4.5	113	59.5	87.0	96.0	99.0

TABLE 2

TIME IN MINUTES REQUIRED FOR THE CENTER LINE OF A WALL TO REACH GIVEN PERCENTAGE VALUES OF THE SURFACE TEMPERATURE, ALLOWING FOR THE CORNER EFFECT

Center Line Temperature in Per Cent of Surface Temperature Wall Thickness		95	90	80	70	60
Inches	Millimeters	Time in Minutes				
2.0	50	32	26	18	14	12
2.5	63	54	42	30	24	20
3.0	75	76	56	44	32	27
3.5	90	106	82	62	50	38
4.0	100	130	104	76	62	48
4.5	113	170	134	98	78	60

TABLE 3

RELATIONSHIP BETWEEN THE WALL VOLUME AND AREA AND THE FREE VOLUME AND AREA WITHIN THE CHECKER WORK, FOR VARIOUS RATIOS OF WALL THICKNESS AND CHECKER OPENING, BASED UPON SQUARE CHECKER PASSES

Ratio of Wall Thickness to Checker Opening Diameter of Square Pass				Brick Area or Volume per Unit of Area or Space	Free Area or Volume per Unit of Area or Space
Wall	Pass	Wall	Pass		
1.00	0.33	3.00	1.00	93.75	6.25
1.00	0.40	2.50	1.00	91.84	8.16
1.00	0.50	2.00	1.00	88.88	11.12
1.00	0.67	1.50	1.00	84.00	16.00
1.00	1.00	1.00	1.00	75.00	25.00
1.00	1.50	0.67	1.00	64.00	36.00
1.00	2.00	0.50	1.00	55.56	44.44
1.00	2.50	0.40	1.00	48.98	51.02
1.00	3.00	0.33	1.00	43.75	56.25
1.00	3.50	0.28	1.00	39.51	60.49
1.00	4.00	0.25	1.00	36.00	64.00
1.00	4.50	0.22	1.00	33.06	66.94
1.00	5.00	0.20	1.00	30.56	69.44
1.00	6.00	0.17	1.00	26.53	73.47

The values for the last two columns of this table were obtained from a table contained in Mr. Boynton's paper.

TABLE 4

DATA ABSTRACTED FROM "BLAST FURNACE ADVANCEMENT," BY A. E. MACCOUN, PRESENTED BEFORE THE AMERICAN IRON AND STEEL INSTITUTE MEETING OF MAY 28, 1915

Test of a 22′ 0″×100′ 0″—Two-pass Cowper Hot-blast Stove

Heating surface of checkers	51,192 ft²	4764 m²
Total of stove	55,866 ft²	5202 m²
Combustion chamber, area	40.8 ft²	3 m² 79
Height to burner	10.62 ft	3 m 50
Above burner	76.75 ft	23 m 10
Total	87.38 ft	26 m 60
Volume, below burner	468 ft³	13 m³ 26
Above burner	3092 ft³	87 m³ 54
Total	3560 ft³	100 m³ 80
Dome volume	1381 ft³	39 m³ 10
Checker, Number of passes	329	
Dimensions of passes	6″×6″	150×150 mm
Total area of passes	82.25 ft²	7 m² 64
Perimeter of one pass	2.0 ft	0 m 60
329 passes	658 ft	197 m 40
Height of	77.75 ft	24 m 00
Heating surface = 197.40×24.0 =	51,160 ft²	4738 m²

(Difference due to use of approximate metric value)

Wall thickness	0.25 ft	75 mm
Area per pass = (150×75)² =	......	0 m² 050625
Of pass = 150²	......	0 m² 02250
Brickwork per pass (difference)	......	0 m² 028125
Percentage of in brick (Table 3)	......	55.56
In pass (Table 3)	......	44.44
Total Checker = 0.050625×329 =	......	16 m² 655
Passes = 0.022500×329 =	......	7 m² 403

(Difference due to use of approximate metric value)

Brick = 16.655 − 7.403 =	......	9 m² 252
Volume, Total brick and passes 16.655×24 =	......	400 m³ 30
Passes = 7.403×24 =	......	177 m³ 64
Brick	......	222 m³ 66

(Difference due to use of approximate metric value)

Weight (as given) 251.85×1800 =	......	453,350 kg
(metric) 222.66×1800 =	......	400,800 kg

TABLE 4—*Continued*

	Blast	Air Supply for Combustion of Gas	
		40 Per Cent Excess	80 Per Cent Excess
Volumes at 0° 760 mm per second	16 m³ 78	2 m³ 26	2 m³ 26
0° 1 atmosphere	8 m³ 39		
Temperatures, maximum	650°	1220°	1220°
Minimum	70°	280°	280°
Average	360°	750°	750°
Change	+580°	−940°	−940°
Gas factors for average temperature	2.32	3.75	3.75
Air supply, per cent of gas volume	....	95.0	123.0
Volume 0°−760 mm	......	2 m³ 15	2 m³ 78
Products of combustion per cent gas	......	182.0	210.0
Volume 0°−760 mm	......	4 m³ 31	4 m³ 75
Volumes at average temperature and pressure	19 m³ 46	16 m³ 16	17 m³ 81
Time, average, in seconds to pass through			
Combustion chamber	4.50	5.42	4.94
Dome	2.02	3.45	2.21
Checker	9.14	11.00	9.98
Total	15.66	19.87	17.13
Temperature change, average per second			
Based on total volume	37.1°	47.3°	54.9°
Based on checker only	63.5°	85.5°	94.2°
Velocity, average in checker	2 m 626	2 m 182	2 m 405
Heat capacities per 100 molecular volumes from Curves. Calories			
Gases at 1220°	......	1950	2220
Blast at 650°	465		
Gases at 280°	......	375	433
Blast at 70°	50		
Difference	415	1575	1787
Calories released or absorbed per second			
(2.26×44.80÷100)×1575 =	......	1595	
1787 =	......	......	1809
(16.78×44.80÷100)× 415 =	3120		
Calories absorbed in one hour on blast = 3120×3600 =	11,232,000		
Calories released from gases in 2.8 hour = 10,080 seconds			
1575×10,080 =	......	16,078,000	
1809×10,080 =	......	......	18,232,000
Assuming specific heat of brick = 0.250 at 1° change in its temperature will = 400,800×0.250 = 100,200 calories			

TABLE 4—*Continued*

	Blast	Air Supply for Combustion of Gas	
		40 Per Cent Excess	80 Per Cent Excess
The total average temperature change in brickwork of checker will be on above basis.	112.3°	160.7°	182.2°

The difference between the heat in the blast of the gases will be

16,078,000 − 11,232,000 = 4,846,000 = 30.4% loss
18,232,000 − 11,232,000 = 7,000,000 = 38.8% loss

This covers merely the interchange loss in stove checker. If the total heat released by the gases is considered, down to 0°, the following will govern:

Calories released by gases: Per second			
$(2.26 \times 44.80 \div 100) \times 1950 =$	. . . . . .	1974	
$2220 =$	. . . . . .	. . . . . .	2248
Per cycle: $1974 \times 10{,}080 =$	. . . . . .	19,520,000	
$2248 \times 10{,}080 =$	. . . . . .	. . . . . .	22,485,000

19,520,000 − 11,232,000 = 8,288,000 = 42.46% loss
22,485,000 = 11,232,000 = 11,253,000 = 50.04% loss

TABLE 5

Combustion of Blast Furnace Gas—Computation Sheet

Composition of Gas		Calories	Products of Combustion			
			O_2	CO_2	H_2O	
H_2	3.92×58.2 =	228	1.96	.....	3.92	
CO	23.95×68.2 =	1633	11.98	23.95	.....	
O_2	0.39	....	.....	.....	.....	
CO_2	12.96	....	.....	12.96	.....	
H_2O	1.65	....	.....	.....	1.65	
N_2	57.13	....	.....	.....	.....	
		1861	13.94	36.91	5.57	11
			0.39	.....	.....	
			13.55	36.91	5.57	11

Air Supply		Excess Air	Products of Combustion in Molecular Vol					
Per Cent	Mol. Vols.		O_2	CO_2	H_2O	N_2	Excess N_2	
100	67.75	0.0	0.0	36.91	5.57	111.33	0.0	
120	81.30	13.55	2.71	36.91	5.57	122.17	10.84	
140	94.85	27.10	5.42	36.91	5.57	133.01	21.68	
180	121.95	54.20	10.84	36.91	5.57	154.69	43.36	
Expressed upon a Percentage Basis these Values are								
100	....	...	0.0	24.00	3.62	72.38	.....	
120	....	...	1.62	22.05	3.33	73.00	....	
140	....	...	2.99	20.40	3.08	73.53	....	
180	....	...	5.21	17.74	2.68	74.37	....	

TABLE 5—*Continued*

COMBUSTION OF BLAST FURNACE GAS—COMPUTATION SHEET 2

Points for heat capacity curves. Products of combustion, air and gas volumes from Appendix X are multiplied by values in Table 8, Appendix X.

	Vol.	200°	400°	600°	800°	1000°	1200°	1400°	1600°	1800°
N_2	111.33	155	314	480	648	827	1008	1195	1386	1582
CO_2	36.91	68	147	238	445	458	574	708	853	1004
H_2O	5.57	10	20	33	46	61	77	95	113	133
Σ 100% air		233	481	751	1139	1346	1659	1998	2352	2719
+20%	13.55	19	38	58	79	101	123	145	169	193
Σ 120% air		252	519	809	1218	1447	1782	2143	2521	2912
C+40%	27.10	38	76	116	158	202	246	290	338	386
Σ 140% air		271	557	867	1297	1548	1905	2288	2690	3105
+80%	54.20	76	152	232	316	404	492	580	676	772
Σ 180% air		309	633	983	1455	1750	2151	2578	3028	3491

Heat Capacity Air Supply

100%	67.75	94	191	292	394
120%	81.30	113	229	350	473
140%	94.85	132	267	409	552
180%	121.95	170	344	526	710

Heat Capacity of the Gas

N_2+	85.39	117	238	364	491
CO_2	12.96	24	52	83	118
H_2O	1.65	3	6	9	13
Σ	.....	144	296	456	622

N_2 57.13+O_2 0.39+H_2 3.92+CO 23.95=85.39

TABLE 5—*Continued*

COMBUSTION OF BLAST FURNACE GAS—COMPUTATION SHEET 3

Computation for curves showing the effects of preheat upon calorific intensity. Values from Sheets 1 and 2.

Temperature of Preheat	0°	200°	400°	600°	800°
Calories released by gas at 0° .....(A)	1861				
Added to gas by preheat........(B)	....	144	296	456	622
Preheated gas ($A+B$)........... Σ	....	2005	2157	2317	2483
Added by air preheat 100%......(C)	....	94	191	292	394
Preheated air ($A+C$)...........(D)	....	1955	2052	2153	2257
Preheat gas and air ($D+B$)...... Σ	....	2099	2348	2609	2879
Added by air preheat 120%......(E)	....	113	229	350	473
Preheated air ($A+E$)...........(F)	....	1974	2090	2211	2334
Preheated gas and air ($F+B$).... Σ	....	2118	2384	2667	2956
Added by air preheat 140%......(G)	....	132	267	409	552
Preheated air ($A+G$)...........(H)	....	1993	2128	2270	2413
Preheated air and gas ($H+B$).... Σ	....	2137	2424	2726	3035
Added by air preheat 180%.....(K)	....	170	344	526	710
Preheated air ($A+K$)...........(L)	....	2031	2205	2387	2571
Preheated air and gas ($L+B$).... Σ	....	2175	2501	2843	3193

The values marked Σ, D, F, H, L are spotted upon the heat capacity curves for their corresponding air supplies, and curves are drawn through these points which enable certain approximations to be made in regard to other air supplies and temperatures of preheat.

TABLE 6

INSTANTANEOUS CALORIFIC INTENSITY OF BLAST FURNACE GAS WITH VARIOUS AIR SUPPLIES AND SHOWING THE EFFECT OF PREHEAT OF BOTH THE AIR AND THE GAS (APPROXIMATE). (READ FROM CURVE, FIG. 181.)

Temperature of Preheat		Air Supply in Per Cent of Theoretical Requirements			
Gas	Air	100%	120%	140%	180%
0°	0°	1330°	1240°	1175°	1050°
0°	200°	1380°	1310°	1250°	1140°
200°	0°	1425°	1330°	1250°	1170°
200°	200°	1475°	1390°	1325°	1210°
0°	400°	1440°	1370°	1310°	1225°
400°	0°	1500°	1410°	1330°	1200°
400°	400°	1610°	1530°	1470°	1350°
0°	600°	1490°	1435°	1390°	1300°
600°	0°	1590°	1500°	1420°	1280°
600°	600°	1740°	1675°	1625°	1525°
0°	800°	1550°	1500°	1460°	1390°
800°	0°	1675°	1580°	1490°	1340°
800°	800°	1880°	1825°	1730°	1650°

APPENDIX IX

COMBUSTION AND BOILER SETTINGS

A. D. WILLIAMS

When coal or other commercial solid combustibles are burned directly in the firebox it is difficult and in some cases impossible to obtain a temperature around 1600° C., though the theoretical combustion temperature of a good grade of coal in an athermal enclosure is about 2050° C. This temperature limitation for a long time hampered the development of many lines of metallurgical and industrial work as the cost of obtaining the required temperatures was too great for commercial use. The limitation arises from the fact that it is practically impossible to force sufficient air for complete combustion through the incandescent bed of fuel; in other words, the bed of burning fuel acts as a producer of combustible gases which in turn must be burned by an additional air supply. The rate at which the air is forced through the fuel bed directly affects the rate at which the fuel is burned, but the mass of air per unit mass of combustible is practically constant under all conditions.

A very interesting series of tests on "Combustion in the Fuel Bed of Hand-fired Furnaces" (Bureau of Mines, Tech. Paper 137) has been made by Kreisinger, Ovitz and Augustine. One of the features of this paper that attracted the writer's attention was the statement that the mass of air per unit of combustible, computed from the analysis of the gases in the upper layers of the fuel bed was, in all cases, less than the mass of air per unit of combustible, as measured by an orifice meter, introduced into the ash pit. The quantity of air forced into the ash pit was also less than the theoretical air supply required for the fuel.

In endeavoring to obtain an explanation of this difference it was found that the computed combustion temperatures derived from the heat capacity of the gases in the fuel bed agreed very closely with the observed temperatures. This would appear to indicate that a portion of the air supply, instead of passing up through the fuel, passed up the walls of the firebox. This conclusion was borne out by the writer's recollections of fires which seemed to burn faster close to the wall than they did in the center of the grate and of clinkers occurring along the walls, and fused to them, when the central portion of the fuel bed was free from clinkers. This last would seem to indicate a sort of an air-blast or oxidizing-zone effect close to the walls, strong enough to solidify the viscous ashes.

Another indication of the possibility of the air passing up between the wall and the fuel bed lies in the fact that the CO_2 in a number of the tests had a curve with two peaks. One was a high peak with low CO, after which came a steady decrease with an increase in CO, this valley being followed by an increase in CO_2 (and a decrease in CO) to a low peak; then there was another drop in the CO_2. The last drop in CO_2 only showed in a few cases, and might be an error as it was very slight.

The tabulations on p. 322, of the results with a 12-inch fuel bed summarize the combustion results attained:

The total weight of air required for the burning of 1 lb of combustible will vary from 11.5 to 15 lb and will have a volume at ordinary temperatures of from 144 to 187 cu ft. But the amount of air that can be forced up through the fuel bed is much less. The specific gravity of coal will average about 1.20 and a cubic foot of coal as fired will weigh from 40 to 50 lb. From this it will be seen that the volume of the voids will be between 33 and 45 per cent of the mass and this will also give the proportion of the fuel bed area through which the air must be forced. The temperature of the fuel bed and of the gases leaving it will be about 1400° C. and the gases will have an impressed velocity. A formula might be devised to give the ash pit pressure required for burning a given weight of fuel per unit of area, but there are so many variables that it would be of comparatively little use. The limit of the ash pit pressure is reached when the ascending volume of air forced through the fuel is sufficient to keep it floating. Such an intense draft pressure, however, would tend to carry unburned fuel away from the grate.

	Pittsburgh Coal, 7 Tests			Anthracite Coal, 7 Tests			Coke, 7 Tests		
	Av.	Min.	Max.	Av.	Min.	Max.	Av.	Min.	Max.
Carbon in ash, per cent of total carbon	7.98	3.95	17.90	10.76	2.96	24.80	7.32	3.80	9.55
Carbon burned, per cent of total carbon	92.02	82.10	96.05	89.24	75.20	97.04	92.68	90.45	96.20
Per cent of carbon burned to CO_2	17.00	11.00	25.00	16.95	7.90	31.50	22.00	14.90	29.00
burned to CO	76.00	65.00	85.00	82.53	68.00	91.50	77.70	71.00	84.90
burned to CH_4	4.00	1.00	10.00	0.52	0.30	1.00	0.30	0.10	0.60
Pressure in ash pit, inches of water	.....	0.13	1.33	.....	0.20	2.36	.....	0.05	1.67
Pounds of coal per sq ft per hr	.....	21.00	131.00	.....	20.00	99.50	.....	15.00	118.60
Pounds of air per lb of fuel	.....	4.10	6.10	.....	5.70	7.50	.....	6.10	7.30
Pounds of air per lb of combustible	.....	5.00	8.20	.....	7.30	10.70	.....	8.10	10.60
Pounds of air per lb of combustible from gases	.....	5.20	7.20	.....	5.90	6.60	.....	6.60	7.60

When the burning gases leave the fuel bed they have an ascensional force, due to their temperature alone, of between 30 and 80 ft per second and are at least 1000° C. hotter than the heating surface of the boiler. As a result the ascending stream of hot gas will be very much smaller than the area of the pass and the firebox, the balance of this area being occupied by cooler eddies of gases chilled by their contact with the heating surface of the boiler and the walls of the settings.

A very interesting illustration of the way gas may flow through a setting was shown on page 709 of *Power*, Nov. 17, 1914. Mr. Morgan B. Smith, however, has his arrows indicating the course of the gases pointing in the wrong direction. Fig. 169 shows the correct pointing of the arrows. The hot gases from the fire rise straight up to the coking arch, and just below this arch the gases are forced to flow horizontally to the end of the arch where their velocity carries them through the bank of tubes and the hole in the baffle. In fact it is very probable that the failure of the baffle at this point was due to the continual impinging upon it of the hot gases. The chilled current of gas flowing down around the tubes of the boiler would also tend to force the hot gases, flowing out from below the arch, against the baffle. After passing through the break in the baffle a portion of the hot gases will tend to rise into the upper portion of the first and second pass, while the balance will be carried down by the current of cooler gases flowing down the second bank of tubes and will pass under the second baffle. They will then immediately rise to the outlet, passing up the space between the second baffle and the third bank of tubes, then crossing the tube bank near the top, under the rear drum.

The explanation of this action is very simple. Anyone who has sat by a window on a cold day knows that the air chilled by contact with the window drops to the floor, and that the velocity of the air current is appreciable, although the difference in temperature between the inside of the room and outside of the building is comparatively small, say, from 20° to 40° C. The same phenomenon, except that the current of air rises, may be witnessed in the vicinity of any radiator in a steam-heated building, while the hot-air furnace installed in many homes depends upon convection currents. If currents can be established at such low temperature heads in a room at ordinary living temperature, it is only logical that similar currents will be established where temperature differences of several hundreds of degrees exist.

A phenomenon similar to the deflection of the current of heated gases flowing through the broken baffle by cooler downward currents can be observed by using an electric fan to break up and distribute the current of hot air rising from a radiator. The mere fact that a certain amount of gas is being drawn off at one point of the setting and a corresponding quantity of gas or air admitted at another part will not affect the local convection currents any more than the drawing off and admission of air into a room affects the current of cold air at the windows and of the heated air at the radiators.

Unfortunately we have all been hypontized by the idea that any current of gas passing through a series of enclosures sweeps uniformly through the full cross-section of all the passages or flues open before it, entirely forgetful of the fact that in gases, by reason of their high coefficient of expansion due to temperature changes, great variations of density occur. At a temperature of 1638° C. a cubic meter of air weighs 0.171 kg, while at 0° C. the same volume of air will weigh 1.293 kg or 7.5 times as much. The relative densities may be compared by likening the high-temperature air to water, weighing 62.5 lb per cubic foot, while

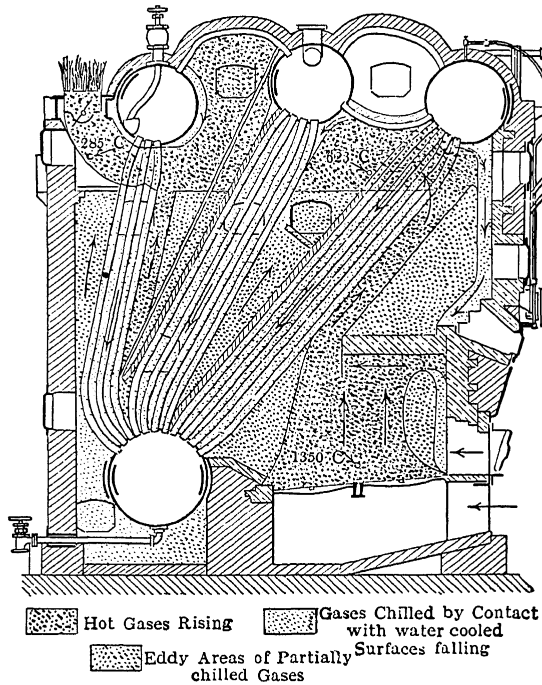


FIG. 168.—Diagram of Gas Flow in Stirling Type Boiler.

the low-temperature air may be likened to molten iron, which weighs 430 lb per cubic foot or nearly 7 times the weight of water. One thing that has tended to obscure this great difference in density that occurs in gases is the fact that most engineering works, in considering the gases passing through a boiler, supply the information as *weight of gases*, allowing the investigator to lose sight of the fact that a unit weight of gas in the firebox will occupy approximately three times the volume it does when it reaches the smoke flue connection. The relative difference in the density of the gases at these two points is as great as the difference in density of wood and granite, and it is a matter of common knowledge that granite will sink in water while wood will float. In the same manner the hottest gases will tend to float above cooler gases. Convection currents and conduction will naturally

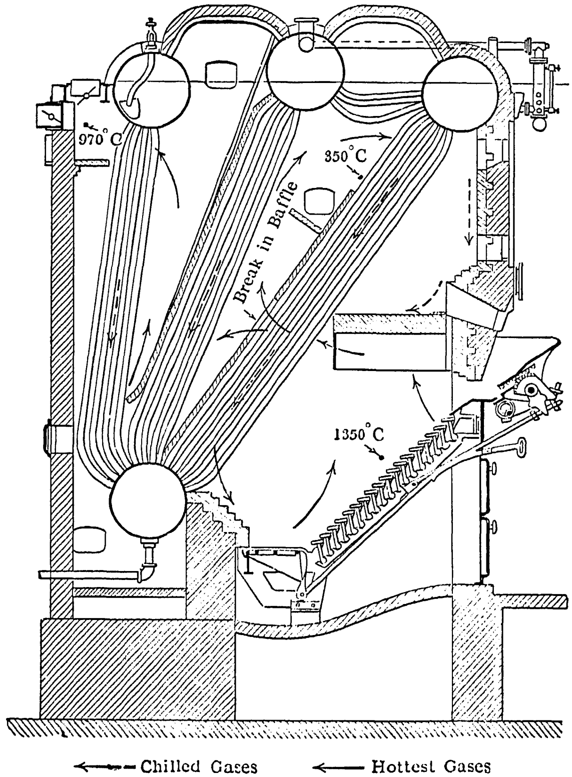


FIG. 169.—Diagram of Gas Flow in Stirling Boiler with Broken Baffle.

tend to equalize the extreme temperature differences, if the supply of heated gases is not maintained, but equalization in this manner would be comparatively slow.

Fig. 168 shows the type of boiler with the broken baffle, baffled according to the builder's catalogue with arrows to show the circulation of the gases. The heated gases rising from the fuel bed impinge upon the arch where the direction of flow changes to horizontal. These gases when they reach the end of the arch will have a certain velocity of flow which will carry them beyond the end of the arch, possibly a portion of them into the first bank of tubes, before they start rising again. The upper portion of the first and second pass acts as an inverted pocket into which the hottest gases flow, the section under the front drum being much hotter than that under the second drum. The tube banks act as refrigerating surface for the gases and heating surface for the water.

The current of hot gases in front of the first bank of tubes is at least 700° C. hotter than the tubes. Their ascensional velocity, due to temperature, is sufficient to carry them the height of the setting in a very small fraction of a second. A part of the gases will be chilled below the ignition temperature and below the temperature which would permit them to float over the top of the first baffle. These gases will circulate down in the spaces between the tubes, depositing soot on them, until they reach the back of the firebox, where they absorb heat and tend to rise again.

The gases in the top portion of the second pass will have a temperature in excess of 600° C. which is about 350° C. hotter than the second bank of tubes, a temperature sufficiently low to permit 75 per cent of the CO present to be dissociate into CO_2 with an accompanying deposit of soot. The refrigerating surface of the second bank of tubes will chill these gases still further so that they will tend to drop into the lower part of the setting where a portion of them will pass under the second baffle. This portion of the gases will be hot enough to float up to the top of the third pass after mixing with some of the chilled gases that have flowed down the third bank of tubes. Some of the gas will be carried downward by the cooling action of the third bank of tubes but the temperature difference between the water-cooled refrigerating surface and the gases is not very great at this point, and the hottest portion of these gases will tend to rise further into the smoke flue.

A number of attempts have been made to distribute the draft loss between the ash pit and the stack damper, on a percentage basis, between the different passes. All of these are apparently based upon the static pressure. In the boiler shown in Fig. 168 there are three vertical columns of gases. These columns exert a pressure against the top of the setting proportional to their vertical height and their average temperature, in the same manner as a chimney. The gases in the first pass, being lighter than those in the second pass, exert more pressure and therefore tend to force the gases in the second pass down and under the second baffle. In the third pass this uplift tends to accelerate the movement of the gases to the smoke flue. The greater the height of the setting the greater the upward pressure. These pressures will naturally affect the reading of any draft gage attached to the boiler. The dynamic pressure due to velocity of flow may have an additive or subtractive effect on the reading of the draft gage, depending upon its location or, more correctly, upon the location of the connections to the setting.

In connection with baffles designed ostensibly to increase the velocity of flow of the off gases of combustion the following table gives the volumetric air supply and the resulting off gases from an exceptionally good grade of coal having a very low ash content.

	Volume at 0° C. and 760 mm Barometer			
	Air Supply CubicMeter per Kilogram Coal	Waste Gas Cubic Meter per Kilogram Coal	Air Supply Cubic Feet per Pound Coal	Waste Gas Cubic Feet per Pound Coal
Air supply 100 per cent.	8.71	9.04	139	145
" 110 per cent.	9.58	9.91	153	159
" 120 per cent.	10.45	10.78	168	174
" 140 per cent.	12.19	12.52	195	201
" 150 per cent.	13.06	13.39	209	215
" 175 per cent.	15.24	15.57	244	250
" 200 per cent.	17.42	17.75	278	284

As the composition of the coal varies, these volumes will vary. The coal consumption per boiler horsepower will vary from two pounds per hour upward and the total amount of coal burned on a grate will depend upon the size of the boiler and the amount it is forced above its nominal rating. As large sizes are the most interesting an 875 horsepower Sterling type boiler operated at several loads will be considered. The volume inside the setting is about 5100 cu ft; about 30 per cent of this space will be occupied by tubes, etc., so that the net volume will be about 3570 cu ft, or a trifle over 100 cu m. A boiler of this size should operate on a coal consumption of about 2.5 lb per horsepower with 20 per cent excess air. The gases of combustion will have the following volume:

	Volume of Waste Gases per Second			
	120 Per Cent Air Supply		140 Per Cent Air Supply	
	Cubic Feet	Cubic Meter	Cubic Feet	Cubic Meter
Normal rating.........	106	3.001	123	3.483
150 per cent of rating...	159	4.501	185	5.225
200 per cent of rating...	212	6.003	246	6.966
300 per cent of rating...	318	9.005	369	10.449

The average temperature of the gases within the setting will be about 800° C., for which the value of the factor $1+0.00367t$ will be 3.936. This gives the average minimum and maximum gas volumes of 417 and 1455 cu ft, from which the average time of the gas inside the setting will be 8.56 seconds for the minimum amount of gas and 2.46 seconds for the maximum amount of gas. This gives a temperature drop of from 117° to 407° C. per second. The distance from the grate to the smoke-flue outlet is about 60 ft, which gives an apparent average velocity of flow ranging from 7 to 24.5 ft per second.

Assuming that the combustion arch is about 6 ft above the grate and the gases leave the fuel at a temperature of 1400° C., they will have an upward pressure against the arch slightly less than 0.10 in of water, and their velocity will be about 42 ft per second. At 200 per cent of rating the volume of gases rising from the fuel bed will be about 212 cu ft per second, which at 1400° C. will give a volume of 1300 cu ft. The firebox of the boiler has a width of 15 ft and the arch will be about 12 ft deep from the front wall, giving an area of 180 sq ft. From Bazin's formula for the flow of water over weirs, Professor Yesmann, of the Polytechnic Institute of Petrograd, has developed a formula which makes it possible to approximate the depth to which the space below this arch will be occupied by the hot gases. In reaching this arch, which forces them to turn horizontally, the gases lose their vertical velocity.

$$h_t = A\sqrt[3]{\frac{Q_t^2}{B^2 \times t}}.$$

Applying this formula to the case in hand, it is found that the depth h_t will be 1.27 m or 4.17 ft. The horizontal velocity of the gases flowing out at the end of this arch will be about 15.25 ft per second and their mean velocity will be about 10.20 ft per second. As soon as they pass beyond the end of the arch the hot gases will have an ascensional component, due to their density, of zero at the end of the arch but increasing so that their rise will be on an arc of a parabola unless interfered with by other forces. In the boiler setting these other forces are a current of colder gases descending among the tubes and the tubes themselves, and while a portion of the gases flowing out from under the arch will penetrate the colder layer of gases and possibly reach the baffle behind the tubes, this action will occur intermittently according as the current of hot or cold gas is stronger in its surges. Naturally these sudden changes in the temperature of the gases at the baffle will cause cracks and ultimately failure of the baffle as shown in Mr. Smith's article. When this occurs the hot gases will pass through the break as indicated in Fig. 169.

At 226 lb pressure absolute, the steam temperature is 200° C. The temperature of the outside of the iron tube should not exceed 220°, allowing a liberal drop between the wet surface and the emulsion of steam and water, and assuming that the tube is clean, and free from both scale and soot. The temperature in the firebox is around 1400° C., and it has been customary to assume that the curve representing the drop in temperature from the fire to the smoke-flue outlet is a regular logarithmic curve, this curve being based upon assumptions which violate the simplest principles of elementary physics. The difference of 1200° between the hot gases and the water tube is nearly five times the difference in temperature between that of the workroom and the temperature of liquid hydrogen, $-259°$. The refrigerating effect of the cold tube on the hot gases is almost equivalent to that of dipping a red-hot bar of steel into liquid hydrogen. The ignition temperature of the gases varies between 350° and 800°; any portion of the unburned gases coming in close contact with the tubes will be chilled below the ignition point. In the case of CO this may result in its decomposition, the formation of carbon and CO_2, which is an exothermic reaction releasing 38.8 calories per molecular volume of CO_2 formed, and the deposition of soot on the tubes. Another source of soot lies in the hydrocarbons. The high firebox temperature has a tendency to break these up, forming soot, CO and hydrogen (H_2). The luminosity of the flame in the firebox is due to the heating of this soot to incandescence, and unless this floating carbon is burned in a properly designed combustion chamber a considerable portion of it will be carried to and deposited on the tubes.

Combustion, like all chemical reactions, follows the law of mass action and the laws of chemical equilibrium. The velocity of the reaction is affected by the temperature of the reacting masses, by the mixing of the combustibles and the comburent and by the dilution with inert gas. The velocity of the reaction increases very rapidly with the temperature. For example, a mixture of hydrogen and oxygen at 200° will require months to combine, while at 2200° the explosion wave indicates that combination occurs in one ten-millionth of a second. The ratio between the two velocities of the reaction is that of 1 to 10^{14}. Increases of pressure increase the velocity of the reaction. This factor, while important in internal combustion motors, does not amount to very much at the slight differences of pressure that exist in furnace fireboxes used in industrial heating. Dilution with inert gases slows down the reaction velocity by reason of the increase of the mass. The mixing has a considerable effect on the velocity of the reaction as may be readily verified by observing the flame of a Bunsen burner while altering the air supply. The slower the reaction velocity the longer the flame length and the distance the gases travel in burning. The temperature will be lower with a long flame than it will be with a short flame.

In a boiler it is absolutely necessary that combustion should be completed before the water-cooled surfaces are reached, as complete combustion will result in the practical elimination of the soot trouble, with a resultant increase in the efficiency of the heating surface. Most boilers and settings, however, are so designed that the incandescent gases are shot up against a refrigerating surface before combustion can be completed. The so-called combustion arches and the baffling being so arranged that this refrigeration will be promoted, instead of combustion.

WATER CIRCULATION IN BOILERS

By A. D. WILLIAMS

Owing to the fact that water may be converted into vapor at rather low temperatures, there is no particular difficulty in constructing a boiler that will produce steam when heat is applied to it. The earliest boilers were modeled upon the familiar kitchen kettle. To-day there are so many different designs of boilers that it would be difficult to enumerate them. A few years ago boilers were operated and rated at the nominal figure of 10 sq ft of heating surface per boiler horsepower. To-day the nominal rating remains the same, but in practice the boiler is operated at from 150 to 300 per cent of rating. The water-tube boiler was devised to increase boiler efficiency by securing improved water circulation and breaking the circulating water up into multiple streams surrounded by hot gases. In this it has met a certain degree of success, as is proved by the modern method of boiler forcing.

The most widely used design of the water-tube boiler consists of one or more steam drums and an assemblage of inclined tubes, the pitch or slope varying from 8 per cent to 30 per cent. The tubes are generally arranged in multiple, but there are designs that employ series operation of a number of multiple banks of tubes. Circulation generally depends upon the thermal syphon principle, that is, upon the hydrostatic head developed by the difference in weight of two columns of water of slightly different temperatures. Additional circulating velocity is supposed to be due to the fact that one of these columns of water is solid and the other partially displaced by steam bubbles.

The accompanying table shows the weights of water and steam at different temperatures, by 20° C. increments, together with the corresponding pressures. At the boiling temperature, 100° C., the ratio between the weights of equal volumes of steam and water is about 1 to 1600, which is sufficient to supply an ascensional velocity to the steam of about 177 m (580 ft) per second. At 200° C. the ratio drops to 1 to 110 and the ascensional velocity to 46.5 m (153 ft) per second. The coefficient of friction between the bubble of steam and the surrounding water is unknown, but even if an extremely liberal allowance is made for this

Weight of Water and Steam at Various Temperatures

Temperature, Degrees C.	Steam Pressure: Kilograms per Square Centimeters	Steam Pressure: Pounds per Square Inch	Weights of: Water Pounds per Cubic Feet.	Weights of: Water, Kilograms per Cubic Meter. Ounces per Cubic Feet	Weights of: Steam, Kilograms per Cubic Meter	Weights of: Steam, Pounds per Cubic Feet	Ratio of Weights, $W \div S$.
0	0.00623	0.0886	62.492	999.87	0.00485	0.000303	206,158
4	....	....	62.500	1000.00			
20	0.0238	0.338	62.389	998.23	0.0173	0.00108	57,701
40	0.0749	1.066	62.016	992.25	0.0511	0.00319	19,517
60	0.2028	2.885	61.453	983.24	0.1305	0.00815	7,534
80	0.4828	6.867	60.739	971.83	0.2938	0.0183	3,307
100	1.0330	14.690	59.899	958.38	0.5980	0.0373	1,602
120	2.0240	28.790	58.963	943.40	1.1220	0.0700	841
140	3.6840	52.390	57.900	926.40	1.9680	0.123	471
160	6.3000	89.590	56.719	907.50	3.2650	0.204	278
180	10.2160	145.30	55.413	886.60	5.1500	0.322	173
200	15.8400	225.20	53.925	862.80	7.8400	0.489	110

coefficient and for the viscosity of the hot water the steam bubble will rise above any moving stream of water circulating through a tube before it can be carried any great distance horizontally. A simple experiment will permit the verification of this. A glass test-tube is nearly filled with water and a smaller tube is used to blow air so it will bubble up through the water. A cork with a vent on one side and a hole for the smaller tube permits the test-tube to be inclined at various degrees. The ratio between the weights of equal volumes of air and water is 1 to 774. (When the tube is inclined the vent should be at the upper side, otherwise the apparatus will " backfire.") Upon blowing through the small tube it will be found that at the vertical position and at slight inclinations from the vertical, the air will rise in bubbles. This action is utilized in the air lift used in raising water from deep driven wells. When the degree of inclination approaches the horizontal there will be a tendency for the stream of bubbles to follow the high side of the tube, and a stream of air can be formed. The less the pitch of the test-tube the easier it is to maintain this stream of air. This would lead to the conclusion that in the inclined-tube design of water-tube boiler the lowest portion of the tube would be filled with water for its full area while at its highest portion the tube would have the lower part of its area filled with water and the upper part filled with steam.

In the boiler, however, the action is complicated by the baffling and the portion of the tube exposed to the hottest gases. With the ordinary method of baffling, used with the B. & W. type of boiler, the high end of the tubes is exposed to the hottest gases and this end, therefore, will be filled partly with steam and partly with water. When the lowest portion of the tube is exposed to the hottest gases there is a tendency for a portion of the steam generated to pass from the tube to the rear header and up to the steam drum. This action may be avoided by bushing down the rear end of the tube and in this manner increasing the velocity of the water entering to an extent that will enable it to carry the steam with it to the front header.

An experimental single-tube boiler may be extemporized from a gage glass, a wooden clamp arranged to hold the tube at various degrees of inclinations, two pieces of rubber tubing fitted to the ends of the gage glass and connecting with nozzles on the bottom of a tin pail, and a Bunsen burner or a blow torch. The pail should be partially filled with water and supported so that there is space below it to permit the gage glass to be brought nearly vertical. The heat can be applied to different parts of the water tube and held there until steam commences to form. The evidence of convection currents may be obtained by dropping a few crystals of potassium permanganate into the water near the down leg. As these crystals slowly dissolve a stream of pink will show the direction of the current. If the water is colored a light pink, it will, in the steaming tests, permit the steam bubbles to have a higher degree of visibility. When the heat is applied at the lower end of the slightly inclined tube it will be found that a portion of the steam goes up the down leg instead of the up leg. As the inclination of the tube is increased this tendency decreases until all of the steam tends to flow to the up leg, and this greatly increases the velocity of circulation. (Note: A neat modification of this one-tube boiler has glass up legs and down legs of various lengths, with only short rubber connections between the tubes and to the pail.) A number of very interesting experiments may be conducted with this simple apparatus, and in this way a better idea of the conditions governing water circulation will be obtained.

The specific heat of water and its heat capacity are unity. The heat capacity of steam has a fractional value, but in contact with water, under the conditions existing in a boiler tube, it has reached the limits of its heat-absorbing capacity for the existing temperature; that is, any additional heat imparted to the steam in the tube will be immediately absorbed by the vaporization of a corresponding amount of the water present. When the water is converted into steam its volume increases from 110 to 1600 times and it displaces from 109 to 1599 additional volumes of water.

The 4-in or 100-mm boiler tube has a sectional area of 78 cm^2 54 (12.56 sq in). Its length is about 5 m 50 (18 ft) and its heating surface 1 m^2 73 (18.6 sq ft). When a boiler is driven at 200 per cent rating it is evaporating water at the average rate of 8.1 grams per square meter per second. On this basis each tube will evaporate 14 grams of water per second on the average. However, it is not unreasonable to suppose that the lower tubes, directly exposed to the hot gases and radiant heat from the fuel, will have an evaporation rate of, say, ten times the average, or 140 grams of water per tube per second. At a steam temperature of 200° C. this would give a volume of 15,400 cm^3 of steam per second. The internal volume of the boiler tube is 43,200 cm^3. This would mean that about 35 per cent of the internal volume of the tube was occupied by steam. At lower steam temperatures the volume of the steam released will be much greater and fill a larger percentage of the tube volume. It is probable that the lower end of the tube will be occupied by solid water, and if this is the case and the steam is generated uniformly the full length of the tube, 70 per cent of the area of the highest end, will be occupied by steam. Assuming that the pitch of the tube is 30 per cent, this is sufficient to give a head that will impress a velocity of 5 m 69 per second on the steam. Friction against the water surface may be either negative or positive, depending upon their relative velocities. Friction against the tube surface will reduce the steam velocity.

The steam in the upper section of the tube will not be able to absorb heat as rapidly as the water in the lower section, but as long as a portion of the perimeter of the tube is covered with water there is no danger of the tube becoming overheated. The hydrostatic head causing the flow of water into the tube will be due to any difference of temperature between the front and rear tube headers and the height of the water column above the tube. Some experiments have indicated that this velocity decreases as the boiler is forced above rating. Should the flow of water be interrupted it will only require a few seconds for the tube to become full of dry steam, it would then rapidly heat until it burst. This would throw full boiler pressure against any slight obstruction and immediately remove all evidence of the cause of the trouble. With tube pitches of less than that assumed, the liability to interruption in the water circulation increases. The slight differential head that exists through a tube might readily be sufficient to hold a piece of loose scale over the end of a tube and be insufficient to break it; that is, a very slight obstacle would suffice to stop the flow of water or so reduce it that a considerable portion of the tube surface would become dry. Another possible cause of burst tubes is the forcing of the boiler to such an extent that the water inflow into a tube becomes insufficient to provide for the evaporation taking place. This last condition would be more likely to occur in a clean tube, free from scale internally and from dust externally, than in a dirty tube.

When the boiler tubes are vertical or nearly vertical the water circulation is enhanced by the "air-lift effect" of the steam bubbles. Like the air lift, this is a problem that involves so many uncontrollable variables that it is doubtful whether any rational expression for this circulation will ever be worked out. The investigation of this circulation, however, will throw considerable light upon the rational design of steam boilers. It is very possible that a modification of the design of the Niclausse boiler, having the field tubes set vertically with the manifold at the top, will offer almost unlimited forcing possibilities, greatly exceeding the evaporative capacity of existing designs. To secure increased circulation in any design it is rather important that the course of the water and the steam bubbles should be arranged in such a manner that they do not impede each other. Theoretically, there should be no limitation to the amount of forcing which a vertical-tube boiler can stand except the heat-absorption capacity of its heating surface. With inclined tubes the boiler can be forced only to the extent of turning a sufficient amount of water into steam to occupy the full area of the hottest tubes at their highest point. Any further forcing with this type of boiler will cause it to destroy itself.

Exactly what the last word in boiler design will be is hard to say. Present designs leave much to be desired, not only in the circulation of the water, but in the manner in which the hot gases pass through the setting and come in contact with the heating surface. The ruling temperature of boilers is very low. Convection currents in water are set up with very low-temperature differences, and as the temperature of the mass increases the convection head caused by one degree difference in temperature increases. These physical principles do not cause design difficulties and it is probably for this reason that boiler engineering has settled into a rut and the mechanical details of the connections entirely overshadow the importance of increasing the heat-absorbing capacity.

SUGGESTIONS FOR CORRECT BOILER BAFFLING

By A. D. Williams

Heating or the application of heat to industrial and domestic processes is an engineering problem of importance. A study of existing installations reveals many interesting discrepancies in practice. In writing about heating furnace a great deal has been said regarding the importance of permitting the heat to soak in. It is well known that a time factor is necessary. The velocity of gas travel is very low in many furnaces. But when boiler settings are considered, a great deal is said regarding the importance of high gas velocities as contributing to efficient operation.

The ideal condition is found between these two extremes. In boiler practice the problem is to pass the gases by the recipient surface at a velocity which will permit them to give up or transfer all of the heat they carry between two temperature limits. Their initial or high temperature limit is fixed by the firebox or combustion temperature. Their final or low temperature limit is fixed by the ruling temperature, or temperature of régime, of the boiler plus the temperature required to supply a temperature head necessary for the transfer of heat. This final temperature of the gases should be somewhere between 200° and 300° C. Frequently it is much higher.

Another point to be considered in connection with gas velocities arises from the fact that the carrying power of the flowing stream varies with the sixth power of its velocity. The heavier particles

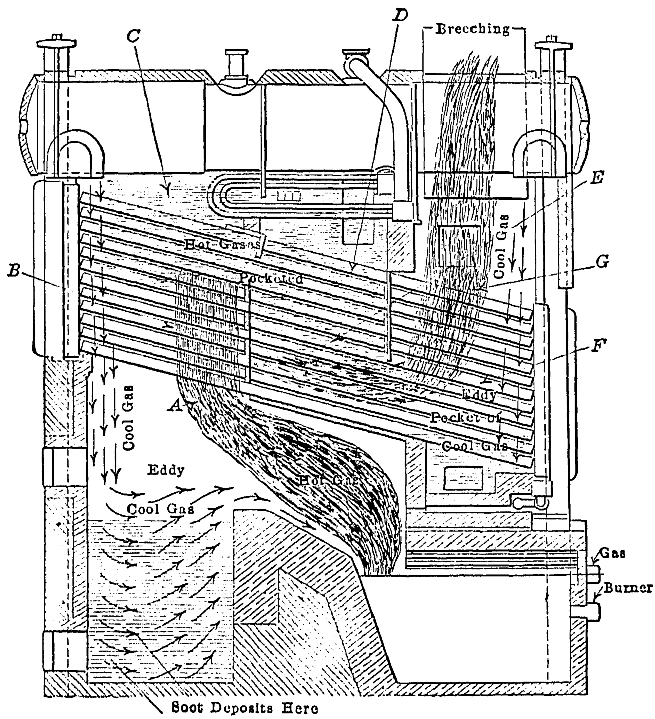


FIG. 170.

Diagram showing the flow of gases in a B. & W. type boiler with special setting. This boiler is gas fired. Temperature observations made in this setting by the Bureau of Mines are shown in Figs. 171, 172 and 173. The observations of temperatures made by the bureau indicate that the flow of the gases follows the laws of hydraulics. The hot gases behaving in the same manner as an inverted stream of water. By looking at this cut from the top instead of the bottom of the page this analogy may be clearly seen.

are more readily thrown out of the stream into the eddies where changes of direction occur. Soot blowers and the steam lance are employed to rid the heating surface of this dirt, soot, etc., instead of attacking the root of the trouble and thereby eliminating it, or at least reducing it to a minimum. The usual methods of setting and baffling boilers coupled with the design of the boilers themselves is well calculated to intensify combustion and soot difficulties. At the same time the troubles with existing plant

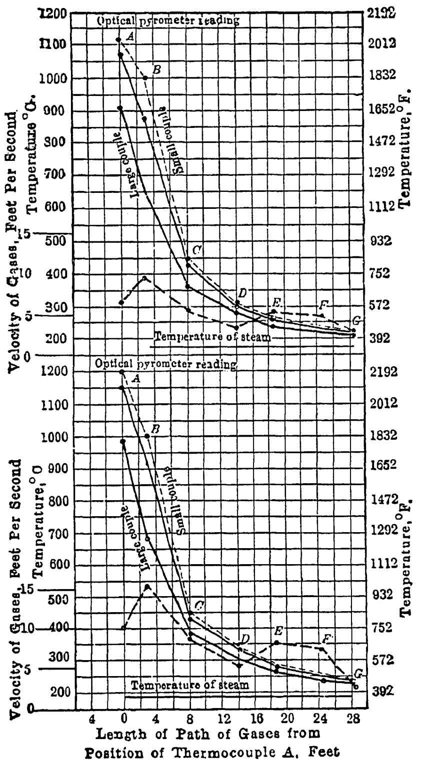


FIG. 171.

Curve showing temperature of the gases flowing through the B. & W. type boiler setting shown in Fig. 170. The letters on the curve refer to pyrometer locations as indicated in Fig. 170. Solid curves show temperatures indicated by pyrometer couples. Dotted curve shows the computed temperatures from experimental data. Dash curve shows computed velocity of gases. This curve is probably based on the assumption that the flowing gases entirely fill the full area of the passage. As the basic assumption does not agree with the action of the gases in this matter, the true velocity curve will be much higher at certain points. Upper and lower diagrams show similar data for two different initial temperatures.

may be considerably reduced by consideration of the laws governing the flow of hot gases.

In April, 1919, the Bureau of Mines issued their Bulletin No. 145 on "Measuring the Temperature of Gases in Boiler

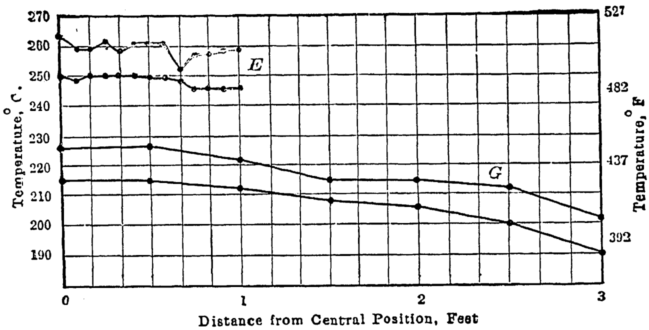


FIG. 172.

Curve showing variation in temperature at points *E* and *G* at different distances from the center line of the boiler shown in Fig. 170. Upper pair of curves shows the temperatures indicated by moving twin couple *E* 1-in steps away from center line. Lower pair of curves show indications of twin couple *G* when moved by 6-in steps away from the center.

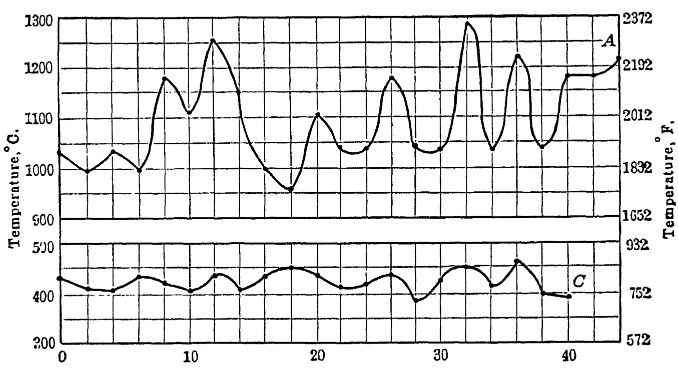


FIG. 173.—Curves Showing the Fluctuations of Temperature at Points *A* and *C* in the Boiler Shown in Fig. 170.

The fluctuations were obtained by reading the small couple at two-second intervals. These fluctuations are a beautiful illustration of the analogy between the flow of water and the flow of hot gases. Couple *C* is located near the bottom of the steam drum where it is immersed in a comparatively quiet body of gas. Couple *A* is located immediately below the tubes at a point where it would show the effects of all the surges in the stream of flowing gas. Anyone who has watched water flowing, from the top of a dam or the spout of a pump, has noticed that the flow is broken up by numerous small surges. This pyrometer shows the effect of the gas surges. Another factor contributing to these surges is the refrigerating action of the water-cooled tubes on the gases. Chilled gases will be held among the tubes in the same manner as a ball is supported by a stream of water or air, until the weight of chilled gases becomes sufficient to permit it to break its way down through the up-flowing current of hot gases. Conditions similar to this exist with all the usual forms of boiler baffling, as they are well arranged to promote this condition. One of the effects of this condition is the formation of soot and CO_2 by the dissociation of CO instead of its combustion. Considerable heat loss results.

Settings." The temperature curves plotted in this bulletin confirm in a very interesting manner the hydraulic laws governing the flow of hot gases. The illustrations in this article are taken from this bulletin and the only additional feature is that in three of the figures lines have been added to show the flow of the hot gases and soot pockets and the flow of the chilled gas in Figs. 170, 174 and 177 of this article (Figs. 2, 4 and 6 of Bulletin 145).

Fig. 170 shows a gas fired type boiler installed at the Carnegie

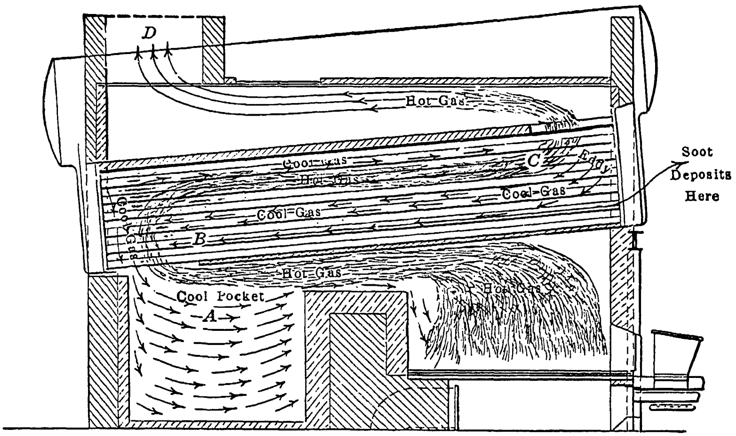


FIG. 174.

Diagram showing flow of gases in single pass horizontally baffled boiler. Underfeed stoker coal firing. Temperature measurements made in this boiler setting are shown in curves, Figs. 175 and 176. By looking at this figure upside down the analogy to the flow of water will be clearly observed. This type of baffling is particularly well adapted for the formation of cool gas and soot pockets.

Institute of Technology. By turning this figure upside down it will appear that the flow from the firebox falls on the inclined baffle on the lower tubes and cascades over the end of the baffle into and among the tubes, flowing under the top of the baffle. This flow continues until the last baffle is overflowed and a cascade is formed to the breeching. The hot gases being very much lighter than the air have a tendency to rise. Relatively cooler gases will have tendency to fall below hotter gases or stratify above colder air or gas. Portions of the setting have been indicated as forming cold or cool gas pockets while other portions are indicated as the location of eddies. Arrows indicate approximately the flow of the cold and the hot gases. The pockets of relatively cold, chilled gases form banks of soot and dirt. Letters on this figure designate the points where the pyrometer couples were located.

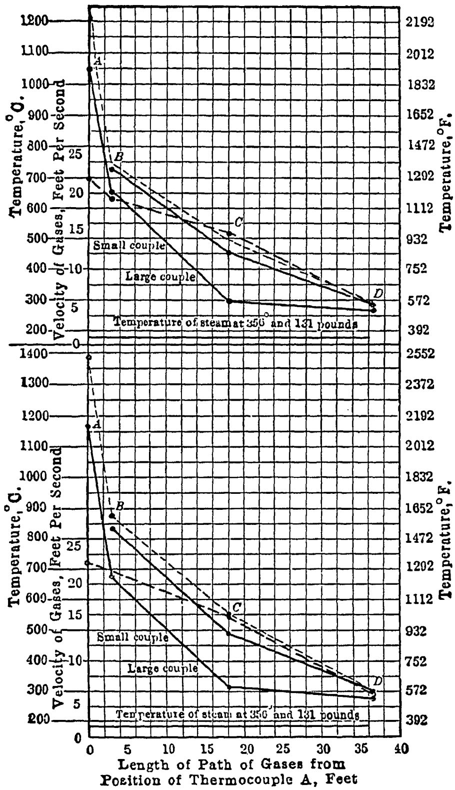


FIG. 175.

Curve showing temperature of gases flowing through single pass horizontally baffled boiler. Positions of thermo-couple are indicated by letters in Fig. 174. Solid lines curve show temperatures indicated by couples. Dotted curve shows approximate computed true temperatures, dash curve computed velocity of gases at different points. The velocity curve, being probably based on the sectional area of the pass, is erroneous. Upper and ower diagrams show similar data for two different initial temperatures.

Fig. 171 shows the curve of temperature drop through the setting. One of the interesting features of this curve is the difference between the indications of the large and the small couples. It is rather important that the pyrometer couple be suited to the work. The large couples are better suited to obtain the average temperature as they have an inertia that prevents their responding to sudden differences due to the continual surging of the gas currents. This surging is very much the same as that of water falling over a dam. Fig. 173 illustrates these surges at points A and C. The surges at C have an amplitude of slightly over 50°, while those at A have an amplitude of about 300°. Anyone

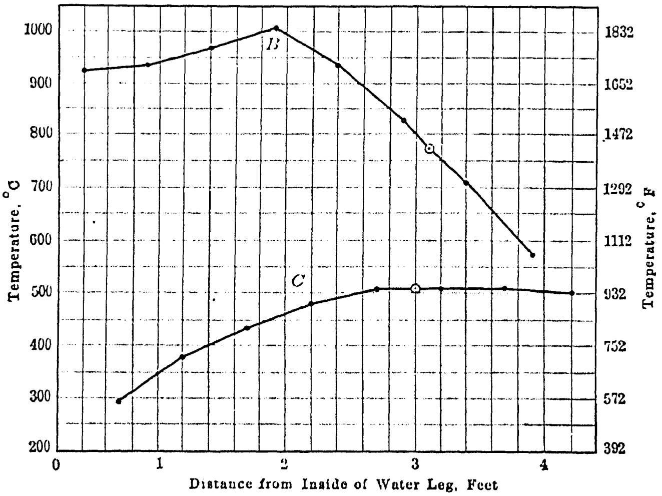


FIG. 176.

Curve showing variation in temperature at different distances from the inside of the water leg of a single pass horizontally baffled boiler, solid fuel, underfeed stoker. These curves were obtained by moving the couples located at points B and C by 6-in steps from a point 6 in inside the water leg to a point 48 in from the water leg. These curves illustrate the "hydraulic" flow of the hot gases forming a stream of high temperature gases below the lower surface of each baffle and cascading over the end of the baffle to the next higher level. The location of the high temperature points and the drop in these curves indicates the trajectory of the stream of gases due to its horizontal velocity in flowing off the end of the baffles.

who has watched a bonfire located where the flames were sheltered from the wind, has noticed that there were continual flame surges. In fact a fairly good idea of flame surges may be obtained by watching a match burn when the flame is sheltered from all drafts. Another illustration of the surging of flames is the "singing flame" mentioned in all works on elementary physics. Any flame burning in an enclosure will be subjected to certain periodic vibration.

The slight amplitude of the temperature surges at C is due to the fact that this couple is located at a point where a pocket of hot gas occurs. This would hold the temperature within rather close limits and it is fairly probable that these surges are due, in part, to the infiltration of cold air, as these couples were located fairly close to the center line of the boiler. This location would bring them between the two drums at a point where cold air might drop on them from leakage. The air leaking in at this point would be heated by the brickwork. Stratification would not be likely

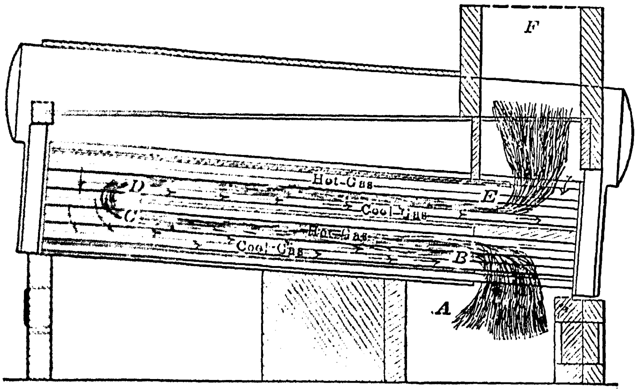


FIG. 177.

Diagram showing the flow of gases in two pass horizontally baffled boiler. Temperature measurements made in this setting are shown in curves, Figs. 178 and 179. This boiler is the same size and make as the single pass shown in Fig. 174. Presumably the steam was generated at about the same rate in both boilers. The location of cool gas pockets is indicated in the same manner as in Figs. 170 and 179.

at this location as the tubes have a tendency to mix the gases very completely as they pass upward.

The large amplitude of the temperature surges at A is probably due to the fact that this couple was at times immersed in the jet of ascending hot gases and at other times in the eddy of comparatively cool gases at this point. Another factor that is mentioned in the caption of the curve is the formation of balls of chilled gases among the tubes. These balls of cool gas will have a tendency to drop into the lower portion of the setting, but will be sustained by the up flowing current of hot gases. An analogy is found in the submerged bubbles of air caused by a stream of water falling into a body of water.

Another fact that is shown in Fig. 173 is that the temperatures at points A and C are those between which the dissociation of 2 CO to C and CO_2 will occur as shown in the curve in Fig. 180.

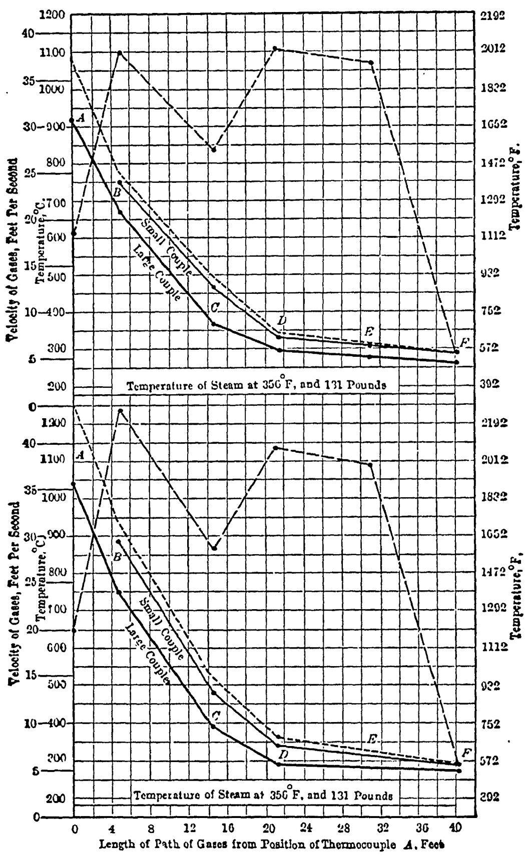


FIG. 178.

Curve showing the temperature of gases flowing through double pass horizontally baffled boiler in Fig. 177. Positions of thermo couples indicated by letters. Solid line curves show temperatures indicated by pyrometer couples. Dotted line curve shows computed approximate true gas temperatures. Dash curve shows computed gas velocities, probably based on gases occupying full area of pass. Upper and lower diagrams show similar data for two different initial temperatures.

A considerable portion of the soot formed in this pass will drop into the pocket in the setting below the pass, other portions will pass on and out of the stack.

Fig. 174 shows the stream of hot gases passing through a single pass horizontally baffled boiler. The location of pyrometer couples is indicated by letters and the curves in Figs. 175 and 176 have been plotted from the readings made. The point *B* on curve in Fig. 175 is 125° lower than the peak of the curve in Fig. 176. In commenting upon Fig. 176 (Fig. 14, Bulletin 145) the bureau says: "Couple *B* showed a considerable range of temperature variation. The maximum temperature, which was at a point about 2 ft from the inside of the water leg, was 430° C.

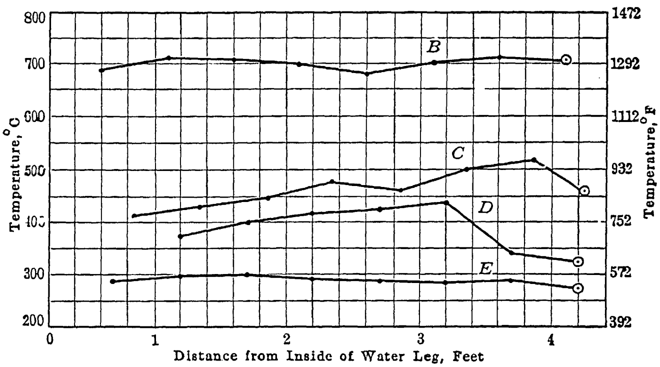


FIG. 179.

Curves showing the temperature at points at different distances inside of the water legs of the boiler as indicated in Fig. 177, showing two pass horizontally baffled boiler. The flatness or lack of variation in curves *B* and *E* indicates that one of these was located in a chilled gas pocket and the other in a stream of hot gases. Curve *C* indicates that the points furthest from the water leg were just on the edge of the stream of hot gas flowing below the baffle. The drop toward the water leg shows an eddy of cooler gas at this point. The drop in *D* furthest from the water leg shows the cool gas pocket formed above the baffle.

higher than that at a point 4 ft from the water leg. This wide variation was undoubtedly caused by the position of the end of the lower baffle and the sudden turn of the gases. *It seems that immediately above the lower baffle there was a layer of comparatively inert gas which had a temperature much lower than that of the stream of moving gases.*" The italics are the writer's. It will be noted that the bureau attributes this correctly as a layer of cool gas is indicated in Fig. 174 immediately above the lower baffle. The drop in the curve *C* (Fig. 176) is likewise due to this eddy of cooler gases. In this boiler (Fig. 174) the slope of the upper baffle is such that it will tend to increase the velocity of the stream of hot gases flowing immediately below it, thus decreasing the thickness of the stream and increasing the thickness of the eddy of cooler gases below it.

It is to be regretted that the bureau did not make surge observations in this boiler setting as they would have been particularly interesting. This particular setting is an extremely good example of what should not be done. In his book, *Fours à Flamme*, Professor Groume-Grjimailo makes the following comment upon boiler settings: "The builders of boilers very rarely pay any

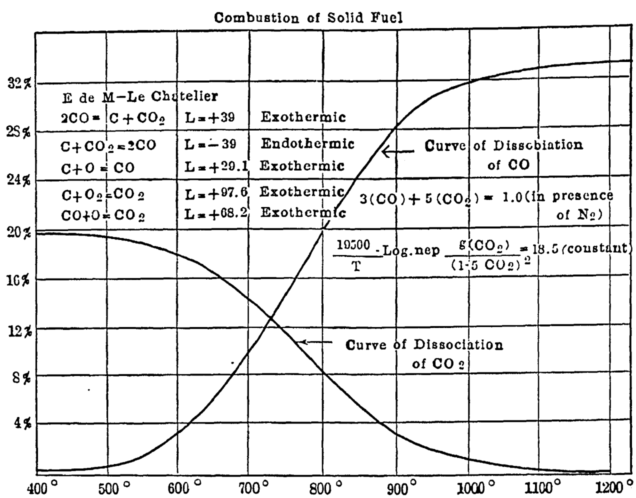


FIG. 180.—Equilibrium Curves of CO and CO_2 in the Presence of Nitrogen. In the combustion of solid fuel the reaction occurs with a rising temperature $CO_2+C=2CO$ when the temperature drops, particularly in the presence of iron oxide the reaction reverses, $2CO=CO_2+C$.

attention whatsoever to the rational distribution of the hot gases. These defects are extremely common in the most recent designs of water tube boilers; this is the explanation of the numerous systems of bafflings and obstructions of the gas passages which are intended to distribute the hot gases in such a manner that they will bathe regularly and uniformly the tubes of the boiler. In reality it is not necessary to have any baffling or distributing walls. The hot gases will rise of their own accord and the heat will be regularly and uniformly distributed to all parts of the boiler." The setting designed by A. Bement is shown in Bureau of Mines

Bulletin No. 40, Fig. 3, page 15. This setting comes the nearest to being in agreement with the flow conditions existing in a boiler set as recommended by the builder. Mr. Bement, however, has altogether too great an area in his first and last pass.

Fig. 177 shows the gas flow in a two pass horizontally baffled boiler. This design of baffling is better than the single pass. The gas velocities computed by the Bureau of Mines are much higher (see curve, Fig. 178) than those of the single-pass boiler. In reality the gain in this boiler is due to the longer path of the gases in contact with the tubes, but it is extremely doubtful whether the gas velocity is any higher than in the single-pass boiler. Fig. 179 shows curves of temperatures made by locating the thermo couples at different distances from the water legs. The drops in curves *D* and *C* indicate the same pockets of cooler gas which occurred in the single-pass setting. While the regularity of the curve *E* would seem to show that this couple had not at any time come in contact with the stream of hot gases.

The location of the last thermo couple in all of these tests seems to the writer to cause some doubt as to whether it really showed the temperature of the stream of hot gases leaving the boiler. This stream of ascending gas would have a fairly high velocity due entirely to its temperature and it is extremely unlikely that it filled the full area of the gas uptake, being surrounded by a layer of relatively cooler gases.

APPENDIX X

HEAT CAPACITY AND CALORIFIC INTENSITY CURVES

By A. D. WILLIAMS

HEAT is probably the energy of the moving molecules of a substance. Its intensity is measured by the thermometric scale, and its quantity by the Calorie or the British thermal unit. Heat transfer takes place by conduction, convection or radiation and will occur whenever a temperature differential exists. Fuel is merely latent heat energy. The quantity of heat stored in a unit of fuel may be determined by the calorimeter or computed from the chemical composition of the fuel. But the amount of heat stored up within the fuel does not supply any index of the temperature differential which may be established by its combustion. For instance, blue water gas at 300 B.t.u per cubic foot will produce a higher temperature than natural gas of three times this thermal value or producer gas of one-half its thermal value, while all of these fuels may be used in a suitably designed furnace for the melting of steel or for other high-temperature applications.

Combustion is the oxidation of fuel, releasing the latent heat and establishing a temperature differential which will permit utilization of the heat. The losses which occur in combustion are small and comparatively unimportant in comparison with the enormous losses which occur in the application and utilization of the heat released. At the same time fuel costs are continually advancing. This leads to a consideration of the possibility of substituting for one fuel another, or of introducing methods of operation and apparatus, which will accomplish the result at a lower cost.

Comparison of the relative values of different fuels for any given purpose involves many factors, any one of which may have a determinative effect. There are many tables giving the thermal values of fuels, as determined by the calorimeter or by computation, but the temperature obtainable by the use of any particular source of heat energy depends upon so many variables that it cannot be reduced to tabular form. Experiment and practice have developed a certain amount of knowledge regarding the temperatures obtainable, but this is an uncertain guide regarding the temperature effects which may be expected under different conditions.

The temperature obtainable by the oxidation of a combustible depends upon:

(*a*) The quantity of heat released by the oxidation of a unit of combustible;
(*b*) The oxygen supply or the amount of air used;
(*c*) The temperature of the fuel and air supply at the point of combination;
(*d*) The heat capacity of the products of combustion;
(*e*) The velocity of the reaction of combustion;
(*f*) The combustion chamber;
(*g*) The completeness of the reaction.

Each one of these seven important variables has a considerable effect upon the temperature produced. *a* is readily obtainable. *b* and *c* are susceptible of control and in turn bring *d* under control. The problem may be solved by making certain assumptions regarding the other variables. These assumptions are:

1. That complete, instantaneous combustion occurs;
2. That the total amount of heat released is contained in the products of combustion, *i.e.*, that combustion occurs in an athermal chamber which does not absorb or give off heat.

It is impossible to realize either of these assumptions in practice, but they permit the computation and plotting of curves which supply a large amount of "*relative*" information regarding the temperature possibilities of a combustible under a variety of conditions. True, the values are only approximate, only relative, the assumed conditions impossible, but at the same time these curves permit a very close approximation of the results obtained in practice and of the effect of variations in operating conditions upon the temperature realized. There is a time element in the combustion of any substance. This time element varies with the temperature of the combining bodies according to an exponential law. High-velocity combustion may be extremely desirable in artillery practice, or in a very high-speed gas engine, but it would be of doubtful utility in a boiler setting or an industrial furnace. The velocity of the reaction is dependent upon mixing as well as temperature; that is, the same amount of gas burned per unit of time will give a short flame when jetted through numerous small holes and a long flame when burned in one jet.

The facility with which gaseous fuels may be mixed with their air supply is one of their inherent advantages over liquid and solid fuels. The liquid fuels fall into an intermediate class, while liquids which have a low evaporative temperature are intermediate between those of high boiling point and the gases.

The investigations of Mallard and Le Chatelier determined the heat capacity and specific heats of several gases, the heat capacity of any substance being the amount of heat which will be absorbed by or given up by that substance when heating or cooling between the absolute temperatures of T_0 and T_1. The specific heat of this substance at any given temperature may be derived from its heat capacity, or the amount of heat that will be absorbed in increasing its temperature from T_0 to T_1. In all heating and cooling operations, the heat capacities of the substances involved are much more important than their specific heats, as the function of the operation is heat interchange. The heat capacities fix the amount of heat that may be given up by one substance as well as the amount of heat absorbed by the other substance, between the temperature limits involved.

Therefore these curves, in addition to showing approximately the temperature that may be realized from the combustion of the particular combustible, give the amount of heat that will be available or given off by the products of combustion in cooling from a high temperature to a lower one.

The utilization of this heat will depend upon:

(*a*) The ruling temperature of the operation, that is, the upper and lower temperatures required;
(*b*) The design of the heating chamber;
(*c*) The heat capacity of the substance to which the heat is imparted;

(*d*) The thermal receptivity and mass of the substance to which heat is imparted;
(*e*) The temperature differential established;
(*f*) The velocity of flow of the gases;
(*g*) The pressure in the combustion chamber.

The accompanying curves show the possibilities, as regards heat transfer, of the respective combustibles when burned with various amounts of excess air and various amounts of preheat for the air and the fuel, when this latter does not contain hydrocarbons dissociated by preheating. The chemical composition of these combustibles is given in Table 6. In the case of the liquid and pulverized fuels the amount of air or steam required for atomizing purposes is given in the caption of the curve. One of the interesting facts revealed is that the use of steam instead of air for atomizing oil results in reducing the temperature obtainable by 170° C. (306° F.). This is due to the fact that the same amount of air is required in both cases, and the steam merely increases the volume and weight of the products of combustion. At the same time it may be that it is much more economical to use the steam directly in the atomizing instead of indirectly in compressing the air for atomizing. A steam jet is notoriously inefficient. In a marine installation where space and weight have to be considered, a Great Lakes boat would use a steam jet floating in her boiler feed supply. On salt water, where boiler make-up is obtained by evaporation, an air compressor would exhaust to the condenser and evaporator capacity would be saved or the air blast fans would be run by motors. Steam jets would increase evaporator capacity and heat consumption. *Mechanical atomizing will avoid fans or compressors but will necessitate the use of oil heaters and the pumps necessary to create the oil pressure required for the spray nozzles.* Assuming a 20 per cent excess air supply, the comparison on p. 352 can be made from the curves. While the difference in favor of air atomizing is only 1 per cent of the heat released it is available in producing steam at boiler efficiency, from 1 to 9 per cent of the steam generated will be used in atomizing. The oil will evaporate from 11 to 14 times its weight of water. With air atomizing the temperature in the firebox will approximate 1885° C. (3280° F.) and with steam 1710° C. (3110° F.). The actual temperature realized will be about 100° C. or 200°

	Steam Atomizing		Air Atomizing	
	Calories per Kg.	B.t.u. per Lb.	Calories per Kg.	B.t.u. per Lb.
Heat released by oil................	10,400	18,720	10,400	18,720
Heat carried to stack at 400° C. (752° F.)	1,960	3,528	1,850	3,330
Difference available in boiler.......	8,440	15,192	8,550	15,390
Difference in favor of air atomizing..	....	....	110	198

F. lower, due to firebox construction. In fact, much lower temperatures may be obtained under unfavorable conditions. Similar methods to the above make these curves and the volumetric data in Tables 11 to 21 extremely useful in designing furnaces for a particular fuel.

Several systems are in use for preheating the air for boilers on ships and on land. The temperature drop of the waste gases ranges from 70° C. (126° F.) to 120° C. (216° F.). The temperature rise in the air ranges from 60° C. (108° F.) to 105° C. (189° F.). These curves may be utilized to approximate the effect of such preheat:

	Steam or Air Atomizing	
	Calories per Kg.	B.t.u. per Lb.
Heat in 120 per cent air supply at 120° C. (184° F.)..........................	420	756

This will result in an increase in calorific intensity of about 50° C. (90° F.). This extra heat will be utilized at boiler efficiency in producing steam. Preheating of the oil supply to secure fluidity results in a slight addition to the heat available.

The ruling temperature of a heating operation is the temperature to which the material being heated must be raised in order to effect a given result. In order to produce this rise in temperature a heat flow must be established by a temperature differential. In many cases it is necessary that the products of combustion leave the heating chamber at a temperature higher than the ruling temperature. This is the case in the open-hearth furnace, the steam boiler, etc. One of these is such a high-temperature operation that it is necessary to preheat the air and frequently the fuel supply in order to attain the high temperature at which the waste gases leave the chamber, while waste heat boilers can be advantageously employed to recover heat leaving the regenerator chambers. The steam boiler is a low-temperature heat application and for that reason permits a greater heat utilization than is possible in a high-temperature process; at the same time it is frequently found desirable to pass the gases leaving the boiler through an appliance which recovers heat and returns it to the boiler in the shape of hot feed-water or preheated air supply. The foregoing methods of preheating the air and fuel supply or the waste heat boiler are indirect methods of heat recovery, while the heating of the feed water or the preheating of material as it is moved from a low temperature to a higher temperature zone in the same furnace are direct methods of heat recovery. The indirect methods involve a greater inherent heat loss than the direct methods. High-temperature operations and processes cannot be accomplished without considerable loss of heat. The lower the temperature required for an operation or process the greater the possibility of reducing the heat loss to a minimum by utilizing the maximum amount of the heat released in the operation itself. At the same time the very ease with which the low-temperature operation may be accomplished leads to a very superficial study of its possibilities. Therefore such operations are notorious for their wasteful utilization of the heat and fuel.

It is hardly necessary to state that intermediate conditions between those plotted on the curve may be obtained by proportion. In the same manner other values for preheat may be established by the use of dividers and graphical addition; that is, the curves permit the making of numerous approximations, giving a relative idea of what may be expected from any fuel, a base for the comparison of the results which may be obtained by the substitution of one fuel for another. However, it is very important to bear in mind that the heat appliance which works successfully and economically with one form of fuel may not be adapted in any way to the use of another form of fuel, unless extensive alterations are made. There are great differences, not only in the amount of heat released by different fuels, but in the temperatures obtained by the different fuels. For instance, Table 22 compares several gases, arranged in the order of the temperatures obtained through their combustion with the theoretical air supply. It is interesting to note that blue water gas comes first with a thermal value of less than one-third that of natural gas, which comes third in calorific intensity. Another interesting fact is that with 80 per cent excess air, natural gas and a good producer gas are very nearly on par. This tabulation also shows the number of volumes of air supply, products of combustion and combustible mixture. The gas cannot burn unless it has an opportunity to combine with oxygen and it is the thoroughness of the mixing which governs the length of the flame and its temperature. With too long a flame and without good mixing, the temperature realized may be very much lower than the potential temperature of the gas.

The working base necessary for the computation of any of these curves is a complete analysis of the fuel, either by weight or by volume. When the proportions are given by weight, each unit per cent is taken as 10 grams or 10 ounces and divided by the molecular weight to obtain the molecular proportions. Volumetric values may be considered as gram molecules or ounce molecules or as cubic feet or cubic meters. As the chemical equation of any substance fixes the relative weights of the substances and the relative volume when in the gaseous state as well as the amount of heat released or absorbed in a reaction, basing the computations on the molecular composition simplifies the work of computation by eliminating a number of clumsy conversion factors.

A further advantage of using molecular units arises from the following facts:

When the weight is in grams the volume is.......... 22.32 liters
When the weight is in kilograms the volume is....... 22.32 m^3
When the weight is in ounces the volume is......... 22.32 cu ft

Another short cut is the assumption that the air consists of 1 volume of oxygen and 4 volumes of nitrogen, making 5 volumes of air. This assumption is close enough for the purpose and saves much laborious calculation. In addition it compensates to some extent for the fact that the air naturally contains a certain amount of water vapor and that all combustible gases carry water vapor, being generally saturated at ordinary temperatures.

Table 6 gives the usual combustible elements of fuels, their molecular weights and the amount of heat released in calories per gram molecule at constant pressure, water assumed to remain a vapor. While this last assumption gives lower thermal values it agrees with practice, inasmuch as water vapor never condenses within the furnace. The reaction formulas given in this table do not contain the nitrogen. Table 7 is merely an extension of this table, giving oxygen required for the hydrogen and carbon and the total oxygen, the volume of nitrogen and the air volume, as well as the volume of the products of combustion. This table is based on the assumption that the air consists of 1 volume of oxygen and 4 of nitrogen.

The first step in the plotting of these curves is the computation on Table 1, in which the volumetric composition of the gas is considered as giving the number of gram molecules. These are multiplied by the calories released per molecule at constant pressure, and the summation of these values gives the total heat released by the combustion of 100-g molecules of the gas. In the case of solid or liquid fuels it is necessary to divide the weight of each element multiplied by 10 by the molecular weight of the substance or use weight values for the heat released. The products of combustion and the amount of oxygen required are tabulated for the combustible portions of the fuel, and summed with the inert portions of the fuel. When the fuel contains oxygen the amount of this oxygen is deducted from the total of the oxygen column and four times this value is deducted from the total of the nitrogen column. The summation of these four columns gives the total amount of oxygen required and the products of complete combustion in air. If the weight of the fuel is desired the number of molecules may be multiplied by the molecular weights. This total weight may be readily converted to the specific weight of a unit volume.

Should it be desired to note the effect of reducing the air supply, deficiencies of 20 and 40 per cent are generally assumed in order to get three points. Hydrocarbons are assumed to dissociate. It is likewise assumed that 90 and 80 per cent of the hydrogen is consumed and the oxygen remaining is combined with the carbon and carbon monoxide. These assumptions may not be correct, but they supply a base for the purpose desired and data covering the case have not been found. Several other assumptions were made and tried, but the one given seemed to average up with the others within 4 or 5 per cent; therefore it was used. The amount of heat released and the products of combustion were computed in the same manner as for complete combustion, except that the unconsumed hydrogen and carbon monoxide were carried into the products of combustion as shown in the computations on Table 1.

Table 2 is a tabulation of the air supply and the products of combustion for various percentages of the theoretical air supply ranging from 60 to 500 per cent. These values are selected with a view to facility in computation; that is, the volume of excess air or deficiency in air, in the second and fourth lines of the table, is the same as the volume of oxygen for complete combustion, and the other values in this column are simple multiples of this value. The deficiency or excess of oxygen is one-fifth of the excess or deficiency in the air supply. The excess of nitrogen is the difference between the excess air and the excess oxygen. The total nitrogen is obtained by adding the excess nitrogen to the volume of nitrogen in the products of combustion with 100 per cent air supply. The last column, total products of combustion, is obtained by adding the excess air to the total volume of products of combustion with 100 per cent air supply.

Table 3 is the computation of the points for the heat capacity curves of the products of combustion resulting from 100-g molecules of the gas. For the case in hand the computation has been carried to ridiculous limits in order to determine the limiting combustion conditions both with excess air and with a deficiency of air. This point is of some importance in connection with the use of producer and blast furnace gases, and in certain applications of these gases it has been found necessary to use an auxiliary coal fire in order to insure ignition. It is well known that with both of these gases the flame may be extinguished by an excess of air cooling the products of combustion and the heating chamber below the ignition temperature of the gases. Similar difficulties are experienced in starting up regenerative furnaces where the operator has been unfamiliar with producer gas.

Table 8 gives the heat capacities of the usual gases found in the products of combustion as determined by the formulas developed by Mallard and Le Chatelier. These formulas, given in Table 10, were used in computing the heat capacities of the gases at intervals of 200° C. (360° F.) between 0° C. (32° F.) and 2600° C. (4712° F.) which are given in Table 8.

At the bottom of Table 1 there have been collected and totalized the numerical values used in the second column of Table 3 designated by the symbols N_2+ and N_2. These gases have the same heat capacity per gram molecule as given in the second column of Table 8. This procedure simplifies the computation. The other numerical values in the second column of Table 3 were brought forward from Tables 1 and 2. These numerical values, designating the number of gram molecules in the products of combustion, were then multiplied by the heat capacity values given in Table 8 and the products for steps of 400° C. (720° F.) were tabulated in columns three to nine, inclusive, of Table 3. The points for the curves were obtained by the summation of these values for the appropriate combustion conditions. These points were then plotted and connected by curved lines giving a number of diverging parabolas indicated by the solid black lines, each of which is marked by a percentage value designating the air supply conditions which influenced the formation of the products of combustion.

Table 4 shows a similar computation made to obtain the heat capacity of the gas itself and the air supplies.

The heat capacity of the gas is plotted in dash lines and that of the air supplies in dot-dash lines, these last being marked with the percentage values of the air supply.

The next step in the plotting of these curves is the spotting of the points denoting the amount of heat released by the combustion of the gas, with 60, 80 and 100 per cent air supply. The values marked I, F and A in Table 5 give the summations of the number of calories released by the gas as given in Table 1, where they are marked with the same reference letters. From this point on, the plotting may be done graphically or by computing the points as in Table 5. When the points I, F and A have been spotted, a line parallel to the temperature scale is drawn downward crossing all the heat capacity curves, over 100 per cent air supply, the assumption being made that the total heat released is absorbed in raising the temperature of the products of combustion. This curve has a point of flexure at *A* on the curve of the heat capacity of the products of combustion formed with 100 per cent or theoretical air supply.

With this line as a base the points for similar curves may be located by setting off with a pair of dividers the additional heat capacities added by the preheating of the gas and the air supply to any temperature. These heat capacities are given in Table 4 and are plotted on the chart. The dividers may be set from the air or gas heat capacity curves on the charts and measuring the quantity of heat added at the temperature of preheat, and this distance added to the base line *A* to fix a point upon a curve showing the effect of preheat. Care should be used in stepping these distances off to set the dividers parallel with B.t.u. or calorie scale.

The writer frequently prefers to sum up the preheat points analytically, as shown, and use reference letters to connect the corresponding values shown.

TABLE 1

PRODUCER GAS NO. 3, CLEAN COLD GAS. TAR RETURNED TO PRODUCER

Liscum, A. C. Ford Motor, Co., *Power*, Aug. 31, 1920

Theoretical Air Supply (1) (2) (3)	Calories (4)	O_2 (5)	Products of Combustion CO_2 (6)	H_2O (7)	N_2 (8)
H_2 16.60× 58.2=	966	8.30	.....	16.60	33.20
CH_4 5.50×195.2=	1074	11.00	5.50	11.00	44.00
CO 27.10× 68.2=	1848	13.55	27.10	.....	54.20
CO_2 3.90	......	......	3.90		
N_2 46.30	......	......	.....	.....	46.30
H_2O 0.60	......	......	.....	0.60	
I	3888	32.85	36.50	28.20	177.70

80 per cent air supply. Assumed that CH_4 dissociates, that 90 per cent will be burned. O_2 available $=32.85\times0.8=26.28$.

80% Air Supply	Calories	Products of Combustion					
		O_2	H_2	CO	CO_2	H_2O	N_2
H_2 $16.60\times0.9=14.94\times58.2=$	870	7.47	1.66	.....	.....	14.94	29.88
CH_3 5.50 Dissociates							
H_2 $11.0\times0.9=9.90\times58.2=$	576	4.95	1.10	.....	.....	9.90	19.80
C 5.50 $\times97.6=$	537	5.50	....	.....	5.50	.....	22.00
$26.28-17.92=8.36$							
		17.92					
CO 27.10 $16.72\times68.2=$	1141	8.36	....	10.38	16.72	.....	33.44
CO_2 3.90	.....	.....	....	.....	3.90		
N_2 46.30	.....	.....	....	.....	.....	.....	46.30
H_2O 0.60	.....	.....	....	.....	.....	0.60	
F	3124	26.28	2.76	10.38	26.12	25.34	151.42

60 per cent air supply. Assumed that CH_3 dissociates, 80 per cent of H_2 will be burned. O_2 available $=32.85\times0.6=19.71$.

60% Air Supply	Calories	Products of Combustion					
		O_2	H_2	CO	CO_2	H_2O	N_2
H_2 $16.60\times0.8=13.28\times58.2=$	773	6.64	3.32	.....	.....	13.28	26.56
CH_4 5.50 Dissociates							
H_2 $11.00\times0.8=8.80\times58.2=$	512	4.40	2.20	.....	.....	8.80	17.60
C 5.50 $\times97.6=$	537	5.50	....	.....	5.50	.....	22.00
CO 27.10							
$19.71-16.54=3.17$	.....	16.54					
$6.34\times68.20=$	432	3.17	....	20.76	6.34	.....	12.68
					3.90	0.60	46.30
A	2254	19.71	5.52	20.76	15.74	22.68	125.14

100 per cent air supply, products of combustion: $N_2=177.70$

80 per cent air, products of combustion:

$$N_2+CO+H_2=151.42+10.38+2.76=164.56.$$

60 per cent air, products of combustion:

$$N_2+CO+H_2=125.14+20.76+5.52=151.42.$$

The assumption is made that air $=1$ volume O_2 and 4 volumes of N_2; that 5 volumes of air are required to supply 1 volume O_2. This allows for moisture in fuel and air supply.

TABLE 2

VOLUMETRIC AIR SUPPLY AND VOLUME OF PRODUCTS OF COMBUSTION PER 100 VOLUMES OF GAS BURNED. COMPUTED FROM DATA IN TABLE 1

Per Cent	Air Supply		Products of Combustion per Unit Volume of Gas Burned							
	Vols.	Excess Vols.	Excess O_2	CO	H_2	CO_2	H_2O	Excess N_2	Total N_2	Total Vols.
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)
60	98.51	−65.70	−13.14	20.76	5.52	15.74	22.68	......	125.14	189.84
80	131.40	−32.85	− 6.57	10.38	2.76	26.12	25.34	......	151.42	216.02
100	164.25	0.0	0.0	.....	.....	36.50	28.20	0.0	177.70	242.40
120	197.10	+32.85	+ 6.57	.....	.....	36.50	28.20	26.28	203.98	275.25
140	229.95	65.70	13.14	.....	.....	36.50	28.20	52.56	230.26	308.10
180	295.65	131.40	26.28	.....	.....	36.50	28.20	105.12	282.82	373.80
260	427.05	262.80	52.56	.....	.....	36.50	28.20	210.24	387.94	505.20
340	558.45	394.20	78.84	.....	.....	36.50	28.20	315.36	493.06	636.60
420	689.85	525.60	105.12	.....	.....	36.50	28.20	420.48	598.18	768.00
500	821.25	657.00	131.40	.....	.....	36.50	28.20	525.60	703.30	899.40

TABLE 3

Points for Heat Capacity Curves of Products of Combustion Computations Made on Basis of 100 Volumes of Gas Burned

Values of Heat Capacity per gram molecule from *Les Sources de L'Energie Calorifique* by Damour, Carnot and Rengade, based on the work of Mallard and Le Chatelier. Refer to Table 7.

Heat Capacity of Products of Combustion. (Note Below)

(1) Temperatures	(2)	(3) 400°	(4) 800°	(5) 1200°	(6) 1600°	(7) 2000°	(8) 2400°	(9) 2800°	
N_2+	151.42	427	881	1370	1,887	2,431	3,005	3,607	
CO_2	15.74	63	143	245	364	501	657	832	
H_2O	22.68	84	187	315	462	630	819	1,035	60% air
$\Sigma=$	......	574	1211	1930	2,713	3,562	4,481	5,474	Curve points
N_2+	164.56	464	958	1489	2,050	2,641	3,265	3,920	
CO_2	26.12	104	237	406	603	832	1,091	1,380	
H_2O	25.34	94	209	351	516	703	915	1,156	80% air
$\Sigma=$	......	662	1404	2246	3,169	4,176	5,271	6,456	Curve points
N_2	177.70	501	1034	1608	2,213	2,852	3,526	4,232	
CO_2	36.50	146	331	568	843	1,162	1,524	1,929	
H_2O	28.20	104	232	391	574	783	1,048	1,287	100% air
$\Sigma A=$	......	751	1597	2567	3,730	4,797	6,098	7,448	Curve points
$B=20\%$	32.85	93	192	298	410	526	652	783	120% air
$\Sigma A+B=$	......	844	1789	2865	4,140	5,323	6,750	8,231	Curve points
$C=40\%$	65.70	185	383	595	819	1,055	1,304	1,565	140% air
$\Sigma A+C=$	......	936	2172	3460	4,549	5,852	7,402	9,013	Curve points
$D=80\%$	131.40	371	765	1189	1,637	2,109	2,607	3,130	180% air
$\Sigma A+D=$	......	1122	2362	3756	5,367	6,906	8,705	10,578	Curve points
$E=160\%$	262.80	742	1530	2378	3,274	4,218	5,214	6,160	260% air
$\Sigma A+E=$	......	1493	3127	4945	7,004	9,015	11,212	13,608	Curve points
$F=240\%$	394.20	1113	2295	3567	4,911	6,327	7,821	....	340% air
$\Sigma A+F=$	......	1864	3892	6134	8,641	11,124	13,919	....	Curve points
$G=320\%$	525.60	1484	3060	4756	6,548	8,436	....	....	420% air
$\Sigma A+G=$	......	2235	4657	7223	10,278	13,233	....	....	Curve points
$H=400\%$	657.00	1853	3824	5946	8,187	....	....	....	500% air
$\Sigma A+H=$	......	2609	5421	8513	11,917	....	....	....	Curve points

This computation has been carried out further than necessary in order to approximate the limits in regard to combustion conditions, both with excess and deficient air supply.

TABLE 4

HEAT CAPACITY OF AIR SUPPLIES AND GAS. POINTS FOR CURVES AND TO BE USED IN SPOTTING THE INTERSECTIONS INDICATING THE EFFECTS OF PREHEAT

This particular producer gas is rather high in CH_4 which will have a tendency to dissociate at high temperatures. This has not been considered in order to find the limiting combustion conditions due to air supply.

HEAT CAPACITY OF AIR SUPPLY

Tempertures, Per Cent.	Vol.	200°	400°	600°	800°	1000°	1200°	Reference Letters Table 5
Air 60	98.55	*137*	278	425	573	732	893	*D*
80	131.40	183	371	566	765	976	1189	*H*
100	164.25	228	463	708	956	1220	1486	*K*
120	197.10	274	556	850	1147	1464	1784	*L*
140	229.95	320	648	991	1338	1708	2081	*M*
180	295.65	411	834	1274	1721	2197	2676	*N*
260	427.05	594	1204	1840	2485	3173	3865	*P*
340	558.45	776	1575	2407	3251	4149	....	*O*
420	689.85	959	1945	2973	4015	5126	....	*R*
500	*821.25*	*1140*	*2321*	3535	4774	6094	....	*S*

HEAT CAPACITY OF GAS

N_2 +	90.00	125	263	368	525	669	814	
CO_2	3.90	7	16	25	35	48	61	
H_2O	0.60	1	2	4	5	7	8	
CH_4	5.50	12	27	44	63	87	112	
Σ	....	145	308	441	623	811	995	*B*

TABLE 5

Computation for Points on Heat Capacity Curves Showing the Effects of Preheating Gas, Air Supply and Both the Gas and the Air Supply

The numerical values used are obtained from Tables 3 and 4, and are combined as indicated.

Temperature of Preheat	0°	200°	400°	600°	800°	1000°	1200°	
Calories Released Gas (*A*)	2254							
(*B*)	....	*145*	*308*	*441*	*623*	*811*	*995*	60% air
A+*B* Σ(*C*)	....	2399	2562	2695	2877	3065	3249	points
(*D*)	....	*137*	*278*	*425*	*573*	*732*	*893*	
A+*D*=*E* Σ	....	2391	2532	2679	2827	2986	3147	points
C+D Σ	....	2536	2840	3120	3450	3797	4140	points
F	3124							80% air
B+*F*=*G* Σ	....	3179	3432	3565	3747	3935	4109	points
H	....	*183*	*371*	*566*	*765*	*976*	*1189*	
F+*H* Σ	....	3307	3495	3690	3889	4100	4213	points
G+*H* Σ	....	3362	3803	4131	4612	4911	5298	points
I	3888							100% air
J Σ	....	4033	4196	4329	4511	4699	4883	points
K	....	*228*	*463*	*708*	*956*	*1220*	*1486*	
I+*K* Σ	....	4116	4351	4596	4844	5108	5374	points
J+*K* Σ	....	4261	4659	5037	5467	5919	6363	points
L	....	*274*	*556*	*850*	*1147*	*1464*	*1784*	120% air
I=*L* Σ	....	4162	4442	4738	5035	5352	5672	points
J+*L* Σ	....	4307	4752	5179	5658	6163	6667	points
M	....	*320*	*648*	*991*	*1338*	*1708*	*2081*	140% air
N+*I* Σ	....	4208	4536	4879	5226	5596	5969	points
J+*M* Σ	....	4353	4844	5320	5849	6407	6952	points
N	....							
	....	*411*	*834*	*1274*	*1721*	*2197*	*2676*	180% air
N+*I* Σ	....	4299	4722	5162	5609	6085	6564	points
N+*J* Σ	....	4444	5030	5603	6232	6896	7559	points
P	....	*594*	*1204*	*1840*	*2485*	*3173*	*3865*	260% air
P+*I* Σ	....	4482	5092	5728	6373	7061	7753	points
P+*J* Σ	....	4627	5400	6169	6996	7872	8748	points
O	....	*776*	*1575*	*2407*	*3251*	*4149*	....	*340%* air
O+*I* Σ	....	4664	5463	6295	7139	8037	....	points
O+*J* Σ	....	4809	5771	6736	7762	9032	....	points
R	....	*959*	*1945*	*2973*	*4015*	*5126*	....	*420%* air
R+*I* Σ	....	4847	5833	6861	7904	8914	....	points
R+*J* Σ	....	4992	6141	7302	8526	9825	....	0points
S	....	*1140*	*2321*	*3535*	*4774*	*6094*	....	*500%* air
S+*I* Σ	....	5028	6209	7423	8662	10082	....	points
S+*J* Σ	....	5173	6517	8046	9285	10793	....	points

These computations have been carried out to obtain limiting values on the curves.

TABLE 6

Heat Released by the Combustion of the Principal Combustibles at Constant Pressure. Water Assumed to Remain as Vapor

Substance Burned, Unit Weight or Volume	Molecular Formula	Molecular Weight	Reaction of Combustion Formula	B.t.u. Per Pound	B.t.u. Per Cu. Ft.	B.t.u. Per Ounce Molecule	Calories per Gram Molecule
Hydrogen...	H_2	2	$H_2+0.5O_2=H_2O$	52,384	293.2	6,548	58.2
Carbon to CO	C	12	$C+0.5O_2=CO$	4,411	148.1	3,308	29.4
Carbon......	C	12	$C+O_2=CO$	14,640	491.7	10 980	97.6
Carbon monoxide......	CO	28	$CO+0.5O_2=CO_2$	4,366	343.6	7,673	68.2
Methane....	CH_4	16	$CH_4+2O_2=CO+2H_2O$	21,961	983.4	21,961	195.2
Acetylene....	C_2H_2	26	$C_2H_2+2.5O_2=2CO_2+H_2O$	21,108	1536.0	34,300	304.9
Ethylene....	C_2H_4	28	$C_2H_4+3O_2=2CO_2+2H_2O$	20,545	1610.0	35,955	319.6
Ethane......	C_2H_6	30	$C_2H_6+3.5O_2=2CO_2+3H_2O$	20,392	1812.5	38,241	339.9
Benzene (gas)	C_6H_6	78	$C_6H_6+7.5O_2=6CO_2+3H_2O$	17,308	3779.2	84,380	750.2
Sulphur.....	S	32	$S+O_2=SO_2$	3,893	348.6	7,785	69.2
Sulphur.....	S	32	$S+1.5O_2=SO_3$	5,164	462.5	10,328	91.8
Evaporation of water...	H_2O	18	....	1,073	54.06	1,207	10.73

Low values are given for B.t.u. and calories released by different substances. Various authorities differ on these values. The values given are from *Les Sources de L'Energie Calorifique*, by Damour, Carnot and Rengade.

TABLE 7

Air Supply, Oxygen Required and Products of Combustion of the Usual Combustibles

This table is based upon the assumption that air = 1 volume of oxygen, O_2 + 4 volumes of nitrogen, N_2. This assumption is sufficiently accurate for most purposes and greatly simplifies the work of computation. If greater accuracy is desired, use 3.8 volumes for N_2 and 4.8 volumes for air.

The volumes may be considered as cubic feet, cubic meters, ounce molecules or gram molecules as desired for the purpose in hand.

Substance Burned, 1 Volume	Molecular		Oxygen Required,			Nitrogen, N_2	Air Required	Products of Complete Combustion with Air.				
	Formula	Weight	For Hydrogen, H_2	For Carbon, C	Total			Carb. Monoxide, CO	Carb. Dioxide, CO_2	Water, H_2O	Nitrogen, N_2	Total
Hydrogen....	H_2	2	0.5	.....	0.5	2.0	2.5	.....	.....	1.0	2.0	3.0
Carbon* to CO	C	12	.....	0.5	0.5	2.0	2.5	1.0	.....	.....	2.0	3.0
Carbon*......	C	12	.....	1.0	1.0	4.0	5.0	.....	1.0	.....	4.0	5.0
Carbon monoxide......	CO	28	.....	0.5	0.5	2.0	2.5	.....	1.0	.....	2.0	3.0
Methane.....	CH_4	16	1.0	1.0	2.0	8.0	10.0	.....	1.0	2.0	8.0	11.0
Acetylene....	C_2H_2	26	0.5	2.0	2.5	10.0	12.5	.....	2.0	1.0	10.0	13.0
Ethylene.....	C_2H_4	28	1.0	2.0	3.0	12.0	15.0	.....	2.0	2.0	12.0	16.0
Ethane.......	C_2H_6	30	1.5	2.0	3.5	14.0	17.5	.....	2.0	3.0	14.0	19.0
Benzene......	C_6H_6	78	1.5	6.0	7.5	30.0	37.5	.....	6.0	3.0	30.0	39.0
Sulphur......	S	32	.....	.....	1.0	4.0	5.0	SO_2 1.0	.....	.....	4.0	5.0
Sulphur......	S	32	.....	.....	1.5	6.0	7.5	SO_3 1.0	.....	.....	6.0	7.0

* Carbon may be considered as a gas or as a solid.

TABLE 8

Heat Capacity in B.t.u. of Gases Measured from 32° to $t = T - 491$ per Cubic Foot of Gas

Temperature, Degrees F.	$N_2 = 28$, $O_2 = 32$, $H_2 = 2$, $CO = 28$	$H_2O = 18$	$CO_2 = 44$	$CH_4 = 16$
32	0.00	0.00	0.00	0.00
392	7.00	8.71	9.32	11.03
752	14.21	18.59	20.1	24.43
1112	21.71	29.57	32.44	40.4
1472	29.32	41.46	45.7	57.7
1832	37.43	55.31	62.57	79.44
2192	45.59	69.87	78.33	102.6
2552	54.05	85.64	96.5	128.2
2912	62.77	102.5	116.4	156.1
3272	71.58	120.2	137.1	180.6
3632	80.85	139.8	160.4	219.4
3992	90.22	160.3	184.6	254.6
4352	99.94	181.9	210.4	292.3
4712	109.87	204.6	237.6	332.7
5072	120	229.9	266.2	374.9
5432	130.4	255.1	296.5	419.8

Adapted from *Les Sources de L'Energie Calorifique*, by Damour, Carnot and Rengade.

Note.—As CH_4 and other hydrocarbons (C_nH_{xn}) break up at low temperatures around 1450° F., their heat capacity is valueless and such gases cannot be preheated successfully. Some data as to the dissociation of CH_4 and C_nH_m were given on page 1014 of *The Iron Age*, April 24, 1913.

TABLE 9

Heat capacity in calories of gases above 0° per gram molecule (or for 22.32 liters = 0.78822 cubic foot of gas).

Temperature C. $t=T-273$ Degrees	$N_2=28$, $O_2=32$, $H_2=2$, $CO=28$	$H_2O=18$	$CO_2=44$	$CH_4=16$	Molecular weights given in grams
0	0.00	0.00	0.00	0.00	B.t.u. per cubic foot = 5.03794× calories per gram molecule
200	1.39	1.73	1.85	2.19	
400	2.82	3.69	3.99	4.85	
600	4.31	5.87	6.44	8.02	Ounce molecule = 22.32 cubic feet
800	5.82	8.23	9.07	11.46	
1000	7.43	10.98	12.42	15.77	
1200	9.05	13.87	15.55	20.37	B.t.u. per ounce molecule = 112.472077× calories per gram molecule
1400	10.73	17.00	19.18	22.44	
1600	12.46	20.35	23.10	30.99	
1800	14.21	23.86	27.21	35.86	
2000	16.05	27.76	31.84	43.55	Calories per cubic meter = 44.80287× calories per gram molecule
2200	17.01	31.82	36.65	50.54	
2400	19.84	36.10	41.76	58.02	
2600	21.81	40.62	47.16	66.04	

Heat capacity values are based on the formulas developed by Mallard and Le Chatelier.

As hydrocarbon gases dissociate when exposed to high temperatures the heat capacity values for CH_4 have a theoretical value above 800°.

Simmersbach in *Journal of Chemical Industry*, Feb. 28, 1913, page 186, and in *Journal für Gasbeleuchtung*, Dec. 13, 1913, page 1242, gives data covering his experiments in regard to the decrease in heating value of gases containing hydrocarbons.

TABLE 10

P. 25, *Les Sources de l'Energie Calorifique.*

The formulas for the law for the heat capacity of gases at constant pressure have been given by Mallard and Le Chatelier in terms of the absolute temperature, and the total heat capacity from 0 absolute (−273°) to a temperature $T=273+t$ can be expressed by a parabolic formula of two terms:

$$Q=a\frac{T}{1000}+b\frac{T^2}{1000^2},$$

in which a is a constant common to all gases, equal to 6.5 and b is a constant variable with the different gases, the value of which is

For nitrogen, hydrogen, oxygen and carbon monoxide = 0.6;
For water vapor = 2.9;
For carbon dioxide = 3.7;
For methane = 6.0.

From this, the molecular heat capacity for a gram molecule or 22.32 liters of a gas between two temperatures $t_0=T_0-273$ and $t=T-273$ may be expressed by the formula:

$$Q=a\frac{T-T_0}{1000}+b\frac{T^2-T_0{}^2}{1000^2}.$$

By taking the values derived from this formula the specific heat for a particular temperature may be obtained by the formula:

$$\frac{dQ}{dT}=C_T=\frac{a}{1000}+\frac{2b}{1000^2}T.$$

TABLE 11

Graphical Comparison of Fuels

		Low	High
Blast-furnace gas, No. 1.	B.t.u. per cubic foot	= 94	96
	Calories per m^3	=834	852

Weight kg. per m^3 1.295 lb. per cu. ft. 0.0809.

Composition Volumetric		Products of Combustion per 100 Volumes of Gas				Percentage Basis (Wet)			
		Air Supply in Percentage of the Theoretical Requirements							
		100%	120%	140%	180%	100%	120%	140%	180%
H_2	3.92	......	......	......	......	......	......	......	......
CO	23.95	......	......	......	......	......	......	......	......
O_2	0.39	0.00	2.71	5.42	10.84	0.00	1.62	2.99	5.21
CO_2	12.96	36.91	36.91	36.91	36.91	23.99	22.05	20.40	17.74
H_2O	1.65	5.57	5.57	5.57	5.57	3.62	3.33	3.08	2.68
N_2	57.13	111.33	122.17	133.01	154.69	72.39	73.00	73.53	74.37
Total	100.00	153.81	167.36	180.91	208.01				
Air vols.........		67.75	81.30	94.85	121.95				
Weight, kg per m^3		1.413	......	......	1.381				
lb per cu ft.....		0.0882	......	......	0.0862				

	Percentage Basis (Dry)			
	100%	120%	140%	180%
O_2	0.00	1.67	3.09	5.36
CO_2	24.90	22.81	21.05	18.23
N_2	75.10	75.52	75.86	76.41

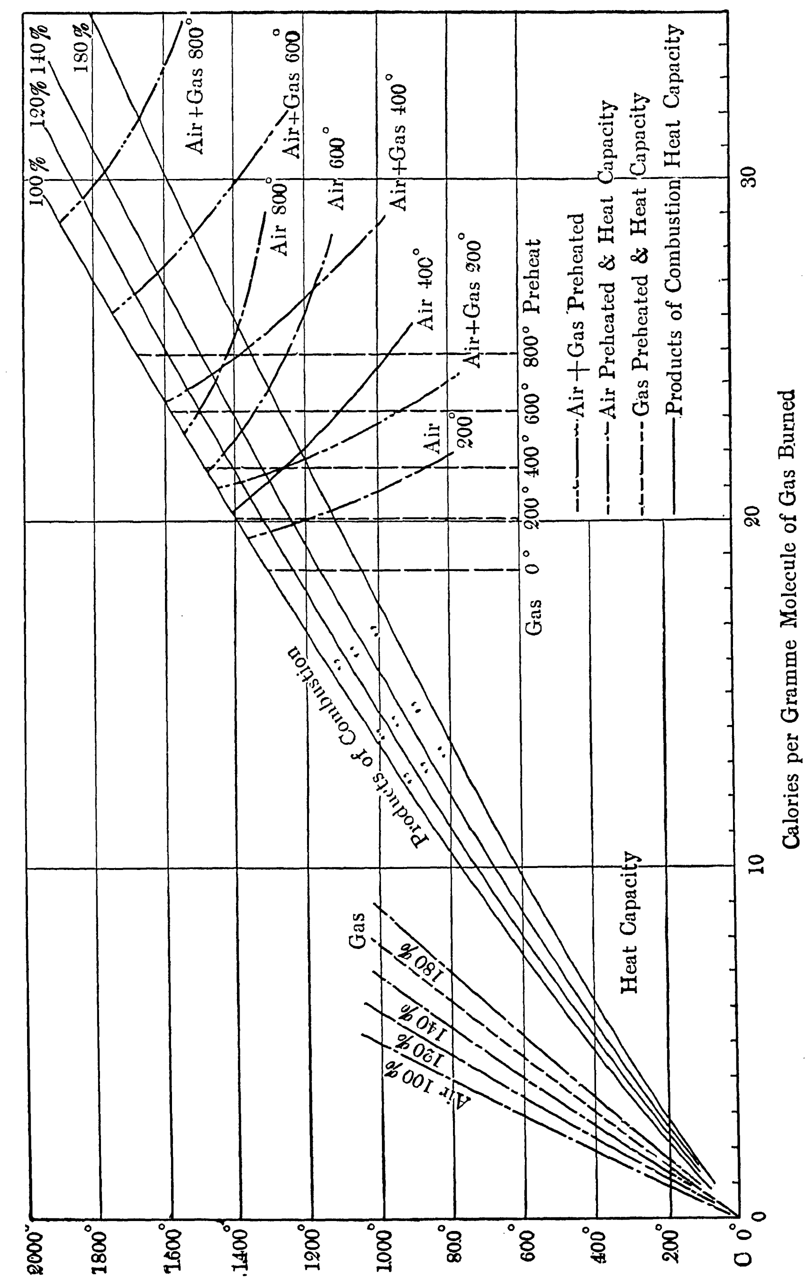


FIG. 181.—Heat Capacity and Calorific Intensity Curve of Blast Furnace Gas No. 1. For composition, volume of products of combustions and their composition refer to Table 11.

TABLE 12

Graphical Comparison of Fuels

		Low	High
Blast-furnace gas, No. 2.	B.t.u. per cubic foot	=105	107
	Calories per m^3	=940	955

Weight, kg per m^3 = 1.295 = 0.0809 lb per cubic foot

Composition Volumetric		Products of Combustion per 100 Volumes of Gas				Percentage Basis (Wet)			
		Air Supply in Percentage of Theoretical Requirements							
		100%	120%	140%	180%	100%	120%	140%	180%
H_2	2.70	......	......	......	......	......	......	......	......
CH_4	0.30	......	......	......	......	......	......	......	......
CO	27.50	......	......	......	......	......	......	......	......
O_2	0.50	0.00	3.04	6.08	12.16	0.00	1.73	3.18	5.48
CO_2	10.00	37.80	37.80	37.80	37.80	23.49	21.46	19.75	17.05
H_2O	......	3.30	3.30	3.30	3.30	2.05	1.87	1.72	1.49
N_2	59.00	119.80	131.96	144.12	168.44	74.46	74.94	75.34	75.98
Total	100.00	160.90	176.10	191.30	221.70				
Air vols.		76.00	91.20	106.40	136.80				
Weight, kg per m^3		1.408	......	......	1.3765				
lb per cubic foot		0.0879	......	......	0.0860				

	Percentage Basis (Dry)			
O_2	0.00	1.76	3.23	5.57
CO_2	23.98	21.87	20.12	17.30
N_2	76.02	76.37	76.62	77.13

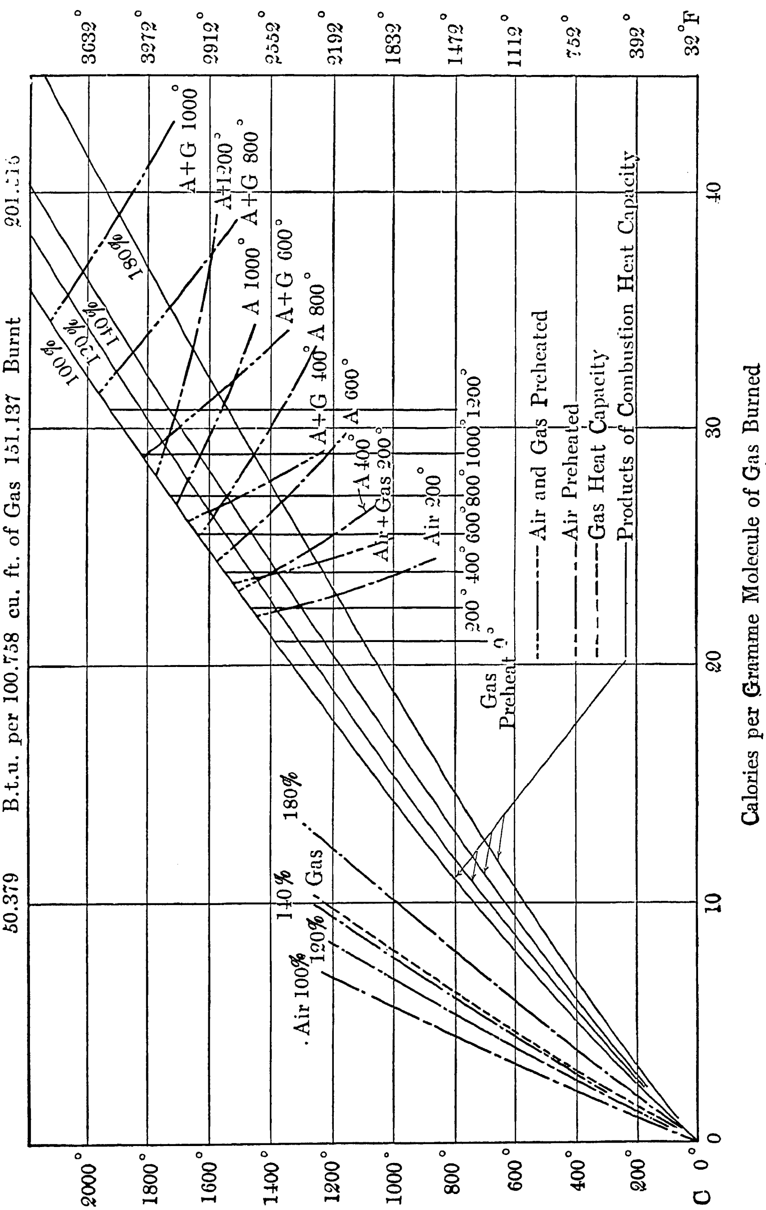


FIG. 182.—Heat Capacity and Calorific Intensity Curve of Blast Furnace Gas No. 2. For composition, volume of products of combustion and their composition refer to Table 12.

TABLE 13

Graphical Comparison of Fuels

		Low	High
Producer gas (mixed) HAW No. 5.	B.t.u. per cubic foot	= 167	176
	Calories per m^3	= 1482	1570

Weight, kg per $m^3 = 1.098 = 0.06862$ lb per cubic foot

Composition Volumetric		Products of Combustion per 100 Volumes of Gas — Air Supply in Percentage of Theoretical Requirements				
		100%	120%	140%	180%	260%
H_2	11.26	....	....	....	....	....
CH_4	3.24	....	....	....	....	....
CO	29.65	....	....	....	....	....
O_2	0.19	0.00	5.35	10.70	21.41	42.82
CO_2	1.25	34.14	34.14	34.14	34.14	34.14
H_2O	3.72	21.47	21.47	21.47	21.47	21.47
N_2	50.69	157.70	179.11	200.52	243.33	328.96
Total	100.00	213.31	240.07	266.83	320.03	427.39
Air vols.........		133.75	160.51	187.27	240.79	347.83
Weight kg per m^3		1.321	....	....	....	1.308
lb per cubic foot		0.0825	....	....	....	0.0817
		Percentage Basis (Wet)				
O_2............		0.00	2.23	4.01	6.68	10.02
CO_2..........		16.00	14.20	12.79	10.66	7.99
H_2O..........		10.07	8.93	8.05	6.70	5.02
N_2............		73.93	74.54	75.15	75.96	76.97
		Percentage Basis (Dry)				
O_2............		0.00	2.45	4.36	7.16	10.55
CO_2..........		17.79	15.61	13.91	11.42	8.41
N_2............		82.21	81.94	81.73	81.42	81.04

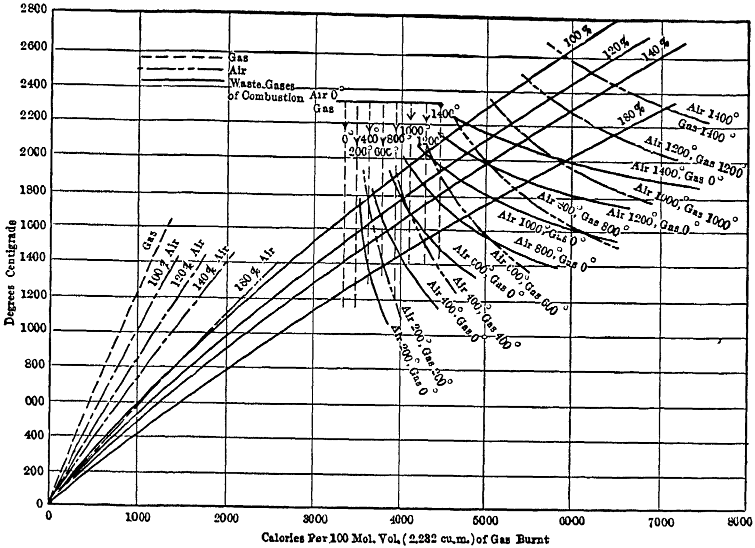


FIG. 183.—Heat Capacity and Calorific Intensity Curve of Mixed Producer Gas *HAW* No. 5. For composition, volume of products of combustion refer to Table No. 13.

TABLE 14

GRAPHICAL COMPARISON OF FUELS

		Low	High
Producer gas (mixed) SC–DSW.	B.t.u. per cubic foot =	142	152
	Calories per m^3 =	1265	1352

Weight = 1 kg 126 per m^3 = 0.0703 lb per cubic foot.

Composition Volumetric		Products of Combustion per 100 Volumes of Gas				Percentage Basis (Wet)			
		Air Supply in Percentage of Theoretical Requirements							
		100%	120%	140%	180%	100%	120%	140%	180%
H_2	12.10	......	......	......	......	......	......	......	......
CH_4	2.60	......	......	......	......	......	......	......	......
C_2H_4	0.40	......	......	......	......	......	......	......	......
CO	21.78	......	......	......	......	......	......	......	......
O_2	0.02	0.00	4.54	9.09	18.18	0.00	2.06	3.75	6.31
CO_2	5.68	30.86	30.86	30.86	30.86	15.64	14.03	12.72	10.71
H_2O	3.82	21.92	21.92	21.92	21.92	11.11	9.97	9.03	7.61
N_2	53.60	144.48	162.66	180.83	217.18	73.25	73.94	74.50	75.37
Total	100.00	197.26	219.98	242.70	288.14				
Air supply.......		113.60	136.32	159.04	204.48				
Weight kg per m^3		1.3168	......	......	1.283				
lb per cubic foot		0.0822	......	......	0.0801				
						Percentage Basis (Dry)			
					O_2	0.00	2.29	4.12	6.83
					CO_2	17.60	15.58	13.97	11.59
					N_2	82.40	82.13	81.91	81.58

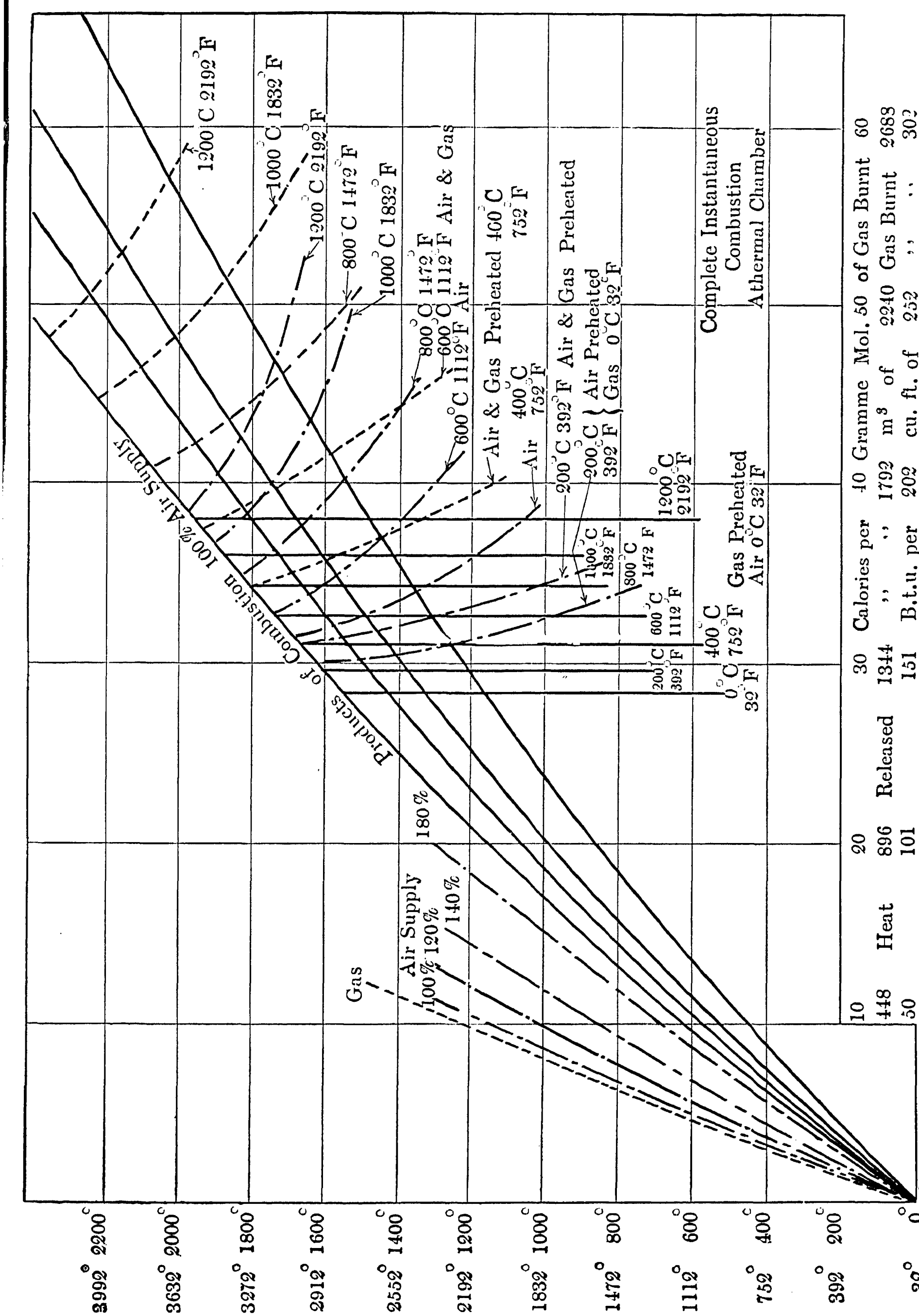


FIG. 184.—Heat Capacity and Calorific Intensity Curve of Mixed Producer Gas, SC–DSW. For composition, volume of products of combustion and their composition refer to Table 14.

TABLE 15

GRAPHICAL COMPARISON OF FUELS

		Low	High
Clean cold producer gas No. 3.		Low	High
Tar returned to producer	B.t.u. per cubic foot =	195	211
	Calories per m^3 =	1742	1875

Weight = 1.057 kg per m^3 = 0.066 lb per cubic foot

Composition, Volumetric		Products of Combustion per 100 Volumes of Gas. Air Supply in Percentage of Theoretical Requirements								
		60%	80%	100%	120%	140%	180%	260%	340%	420%
H_2	16.60	5.52	2.76	......	......	......	......	......	......	......
CH_4	5.50	......	......	......	......	......	......	......	......	......
CO	27.10	20.76	10.38	......	......	......	......	......	......	......
CO_2	3.90	15.74	26.12	36.50	36.50	36.50	36.50	36.50	36.50	36.50
O_2	0.00	0.00	0.00	0.00	6.57	13.14	26.28	52.56	78.84	105.12
H_2O	0.60	22.68	25.44	28.20	28.20	28.20	28.20	28.20	28.20	28.20
N_2	46.30	125.14	151.42	177.70	203.98	230.26	282.82	387.94	493.06	598.18
Tot.	100.00	189.84	216.12	242.40	275.25	308.10	373.80	505.20	636.60	768.00
Air vol....		98.55	131.40	164.25	197.10	229.95	295.65	428.07	559.45	690.85
Weight:										
kg per m^3		1.231	......	......	......	......	......	......	......	1.298
lb per cu ft		0.0769	......	......	......	......	......	......	......	0.0811
		Percentage Basis (Wet)								
H_2		2.91	1.28	......	......	......	......	......	......	......
CO		10.94	4.80	......	......	......	......	......	......	......
O_2		......	......	0.00	2.39	4.27	7.03	10.40	12.39	13.69
CO_2		8.29	12.09	15.06	13.26	11.85	9.77	7.23	5.73	4.75
H_2O		11.94	11.77	11.63	10.24	9.15	7.54	5.58	4.43	3.67
N_2		65.92	70.06	73.31	74.11	74.73	75.66	75.79	77.45	77.89
		Percentage Basis (Dry)								
H_2		3.30	1.45	......	......	......	......	......	......	......
CO		12.42	5.44	......	......	......	......	......	......	......
O_2		......	......	0.00	2.66	4.70	7.61	11.02	12.96	14.21
CO_2		9.42	13.70	17.04	14.77	13.04	10.56	7.65	6.00	4.93
N_2		74.86	79.41	82.96	82.57	82.26	81.83	81.33	81.04	80.86

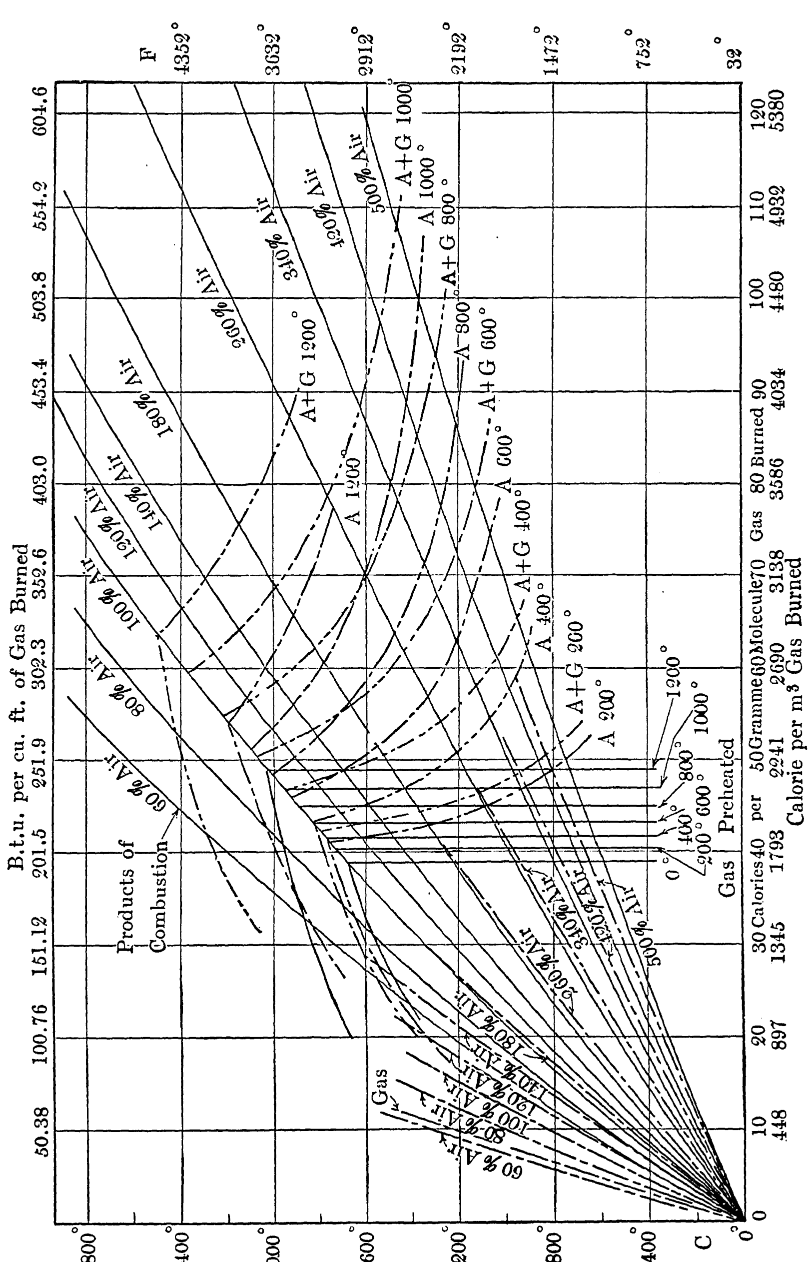


FIG. 185.—Heat Capacity and Calorific Intensity Curve of Mixed Producer Gas No. 3, Cold, Cleaned and Tar Returned to Producer. For composition, volume of products of combustion and their composition refer to Table 15.

TABLE 16

Graphical Comparison of Fuels

		Low	High
Coke oven rich gas, No. 11.	B.t.u. per cubic foot =	467	527
	Calories per m^3 =	4160	4691

Weight = 0 kg 492 per m^3 = 0.03072 lb per cubic foot.

Composition Volumetric		Products of Combustion per 100 Volumes of Gas				Percentage Basis (Wet)			
		Air Supply in Percentage of Theoretical Requirements.							
		100%	120%	140%	180%	100%	120%	140%	180%
H_2	55.00	......	......	......	......	......	......	......	......
CH_4	25.40	......	......	......	......	......	......	......	......
C_2H_4	2.30	......	......	......	......	......	......	......	......
CO	5.70	......	......	......	......	......	......	......	......
O_2	0.20	00.0	17.57	35.14	70.28	0.00	2.94	5.13	8.17
CO_2	2.10	37.80	37.80	37.80	37.80	7.42	6.34	5.52	4.39
H_2O	......	110.40	110.40	110.40	110.40	21.69	18.40	16.13	12.83
N_2	9.30	360.70	430.98	501.26	641.82	70.88	72.22	83.22	74.61
Total	100.00	508.90	596.75	684.60	860.30				
Air vol.	....	439.25	527.10	614.95	790.65				
Weight:									
kg per m^3	.....	1.215	......	......	1.246				
lb per cubic foot		0.0759	......	......	0.0778				
						Percentage Basis (Dry)			
					O	0.00	3.61	6.12	9.37
					CO	9.49	7.77	6.58	5.04
					N	90.51	88.62	87.30	85.59

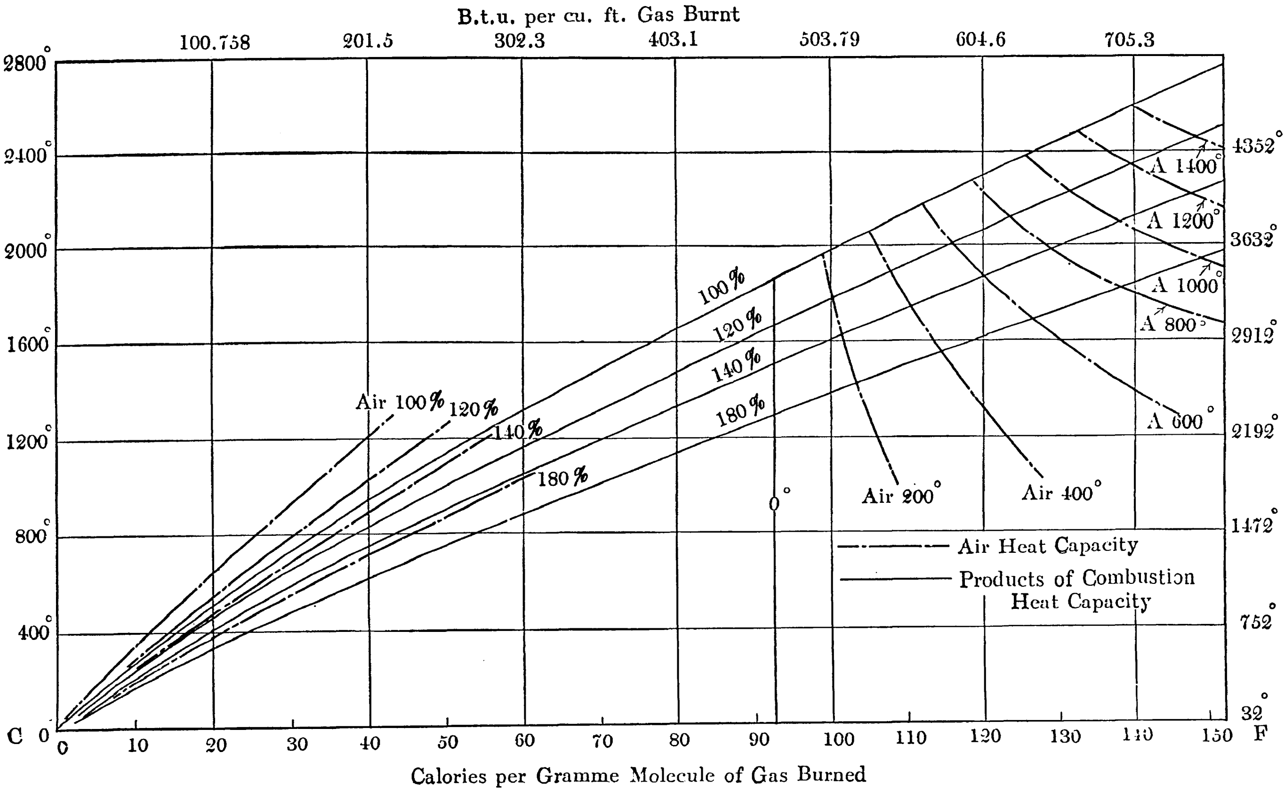


FIG. 186.—Heat Capacity and Calorific Intensity Curve of Coke Oven Rich Gas No. 11. For composition, volume of products of combustion and their composition refer to Table 16.

TABLE 17

Graphical Comparison of Fuels

		Low	High
Blue water gas, No. 9.	B.t.u. per cubic foot	= 290	318
	Calories per m³	=2585	2831

Weight = 0 kg 695 per m³ = 0.0434 lb per cubic foot.

Composition Volumetric		Products of Combustion per 100 Volumes of Gas — Air Supply in Percentage of Theoretical Requirements: 80%	100%	120%	140%	180%	260%
H_2	50.80	5.00	....	....	....	....	....
CH_4	0.20	....	....	....	....	....	....
CO	40.90	13.10	....	....	....	....	....
O_2	0.90	0.00	0.00	9.10	18.20	36.40	72.80
CO_2	3.40	31.40	44.50	44.50	44.50	44.50	44.50
H_2O	0.80	47.00	52.00	52.00	52.00	52.00	52.00
N_2	3.80	149.00	185.40	221.70	258.00	330.60	476.00
Total	100.00	245.50	281.90	327.30	372.70	463.50	645.30
Air supply.......		181.60	227.00	272.40	317.80	408.60	590.20
Weight:							
kg per m³.....		1.245	....	....	....	....	1.291
lb per cubic foot		0.0777	....	....	....	....	0.0806
Percentage Basis (Wet)							
H_2............		2.04	....	....	....	....	....
CO...........		5.34	....	....	....	....	....
O_2...........		0.00	0.00	2.78	4.88	7.85	11.28
CO_2..........		12.79	15.78	13.59	11.94	9.60	6.90
H_2O..........		19.14	18.45	15.89	13.95	11.52	8.06
N_2...........		61.09	65.77	67.74	69.23	71.33	73.76
Percentage Basis (Dry)							
H_2............		2.52	....	....	....	....	....
CO...........		6.60	....	....	....	....	....
O_2...........		....	0.00	3.31	5.68	8.85	12.27
CO_2..........		15.82	19.35	16.16	13.87	10.81	7.50
N_2...........		75.07	80.65	80.53	80.45	80.34	80.23

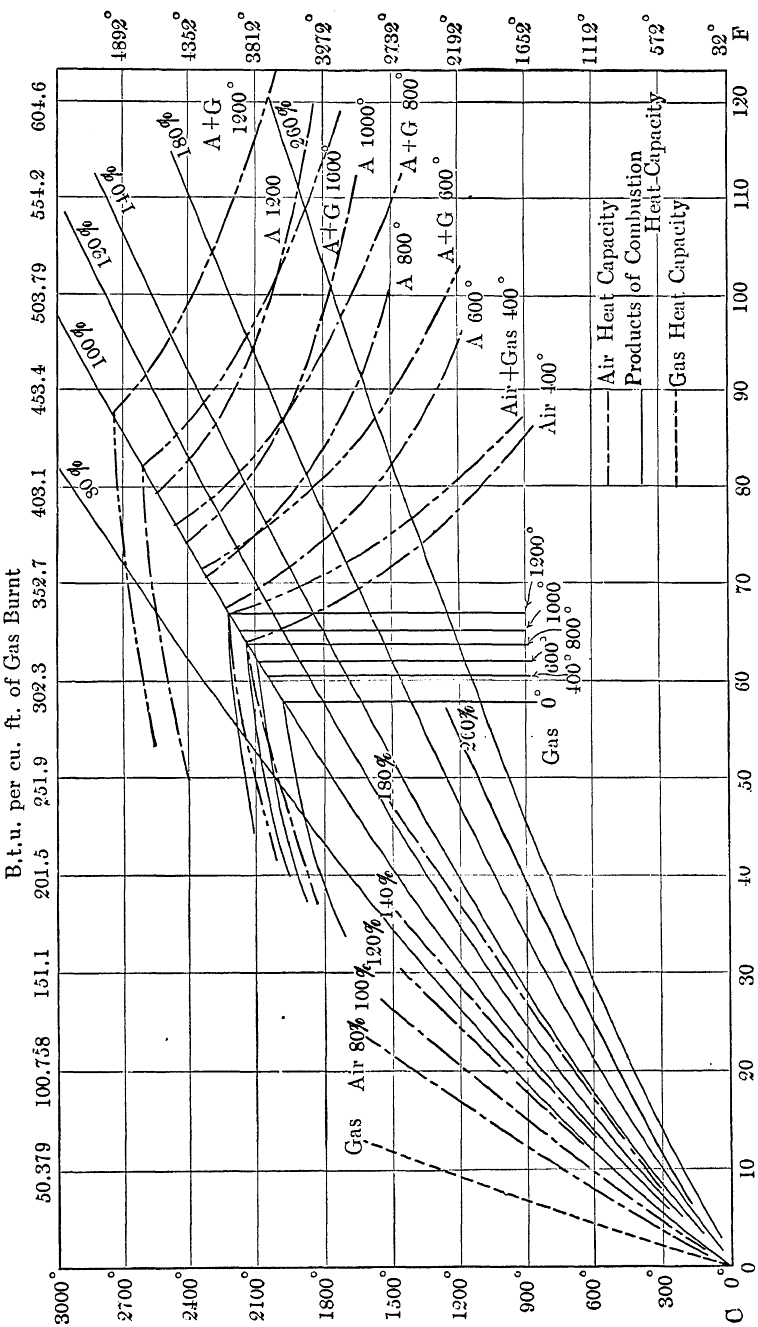


FIG. 187.—Heat Capacity and Calorific Intensity Curve of Blue Water Gas No. 9. For composition, volume of products of combustion and their composition refer to Table 17.

TABLE 18

GRAPHICAL COMPARISON OF FUELS

		Low	High
Natural gas "C," No. 6.	B.t.u. per cubic foot =	924	1025
	Calories per m^3 =	8226	9126

Weight = 0 kg 743 per m^3 = 0.0465 lb per cubic foot.

Composition Volumetric		Products of Combustion per 100 Volumes of Gas				Percentage Basis (Wet)			
		Air Supply in Percentage of Theoretical Requirements							
		100%	120%	140%	180%	100%	120%	140%	180%
H_2	2.10	......	......	......	......	......	......	......	.....
CH_4	92.10	......	......	......	......	......	......	......	.....
C_2H_4	0.50	......	......	......	......	......	......	......	.....
CO	0.40	......	......	......	......	......	......	......	.....
O_2	0.20	0.00	37.35	74.70	149.40	0.00	3.06	5.31	8.40
H_2O	......	187.30	187.30	187.30	187.30	18.14	15.36	13.32	10.52
CO_2	0.50	94.00	94.00	94.00	94.00	9.10	7.71	6.69	5.28
N_2	4.20	751.20	900.60	1050.00	1348.80	72.76	73.87	74.68	75.80
Total	100.00	1032.50	1219.25	1406.00	1779.50				
Air supply...		933.75	1120.50	1307.25	1680.75				
Weight:									
kg per m^3		1.242	....	....	1.268				
lb per cu ft		0.0775	....	....	0.0792				
						Percentage Basis (Dry)			
					O_2	0.00	3.62	6.13	9.39
					CO_2	11.11	9.10	7.72	5.90
					N_2	88.89	87.28	87.15	84.71

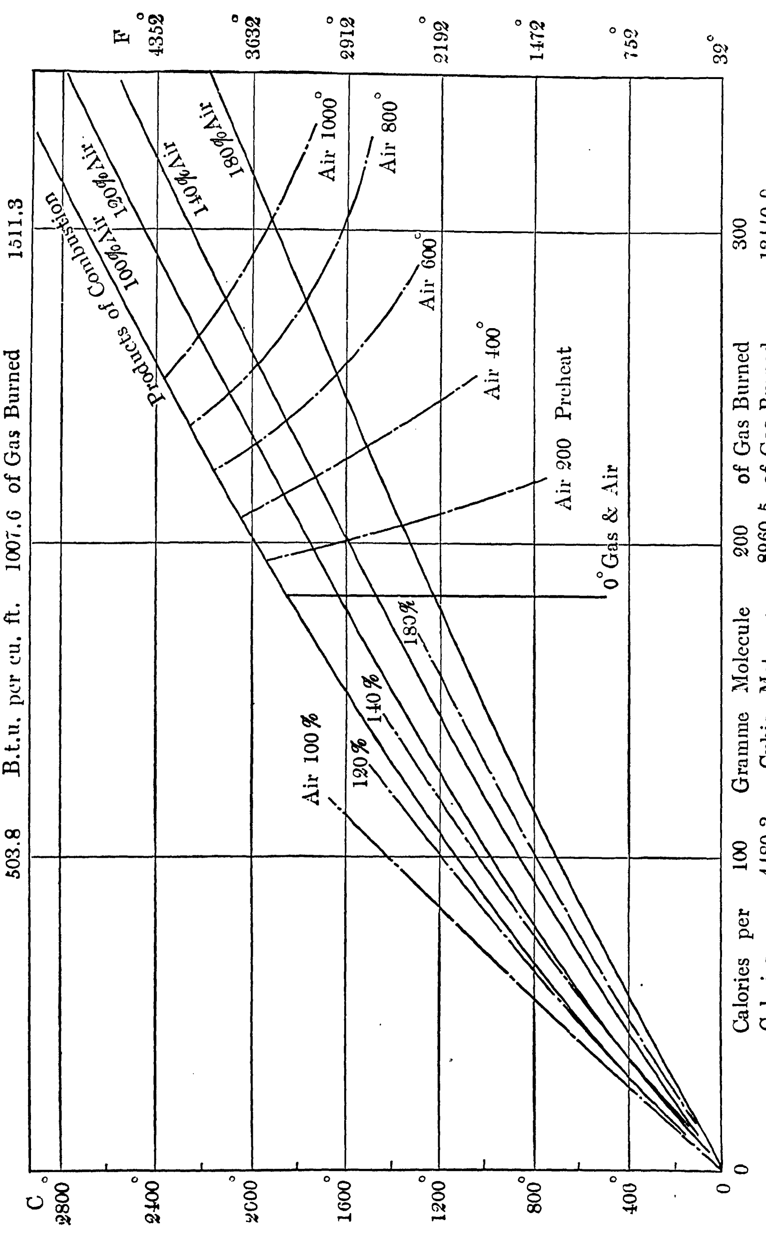


FIG. 188.—Heat Capacity and Calorific Intensity Curve of Natural Gas "C" No. 6. For composition, volume of products of combustion and their composition refer to Table 18.

TABLE 19

Graphical Comparison of Fuels

		Low	High
Fuel oil (No. 17).	B.t.u. per pound =	18,840	20,080
	Calories per kg =	10,467	11,154

Specific Gravity 0.90 = 7.5 lb per gallon.

Composition by Weight	Products of Combustion, Cu M per Kg							
	Air or Mechanical Atomizing				Atomizing Steam = 0.30 Weight of Oil.			
	Air Supply in Percentage of Theoretical Requirements.							
	100%	120%	140%	180%	100%	120%	140%	180%
H_2 12.80	....	....	....	....	....	....	....	....
C 83.00	....	....	....	....	....	....	....	....
O_2 1.40	0.00	0.45	0.90	1.80	0.00	0.45	0.90	1.80
CO_2	1.54	1.54	1.54	1.54	1.54	1.54	1.54	1.54
H_2O 1.50	1.45	1.45	1.45	1.45	1.82	1.82	1.82	1.82
N_2 1.30	9.00	10.80	12.60	16.20	9.00	10.80	12.60	16.20
Total 100.00	11.99	14.24	16.49	20.99	12.36	14.61	16.86	21.36
Air supply..	11.24	13.49	15.74	20.23	11.24	13.49	15.74	20.23
kg per m^3	1.293	....	....	1.291	1.278	....	....	1.274
	Products of Combustion, Cu Ft per Pound							
O_2	0.00	7.21	14.41	28.82	0.00	7.21	14.41	28.82
CO_2	24.73	24.73	24.73	24.73	24.73	24.73	24.73	24.73
H_2O	23.18	23.18	23.18	23.18	29.14	29.14	29.14	29.14
N_2	144.21	173.17	201.82	259.50	144.21	173.17	201.82	259.50
Total....	192.12	218.29	264.14	336.23	198.08	234.25	270.10	342.19
lb per cu ft	0.0807	....	....	0.0806	0.0798	....	....	0.0795
	Percentage Basis (Wet)				Percentage Basis (Wet)			
O_2	0.00	3.16	5.45	8.57	0.00	3.08	5.33	8.42
CO_2	12.87	10.84	9.36	7.35	12.49	10.56	9.15	7.23
H_2O	12.06	10.16	8.77	6.89	14.71	12.45	10.79	8.51
N_2	75.07	75.84	76.42	77.19	72.80	73.91	74.73	75.84
	Percentage Basis (Dry)							
O_2	0.00	3.51	5.98	9.21				
CO_2	14.64	12.07	10.26	7.90				
N_2	85.36	84.42	83.76	82.89				

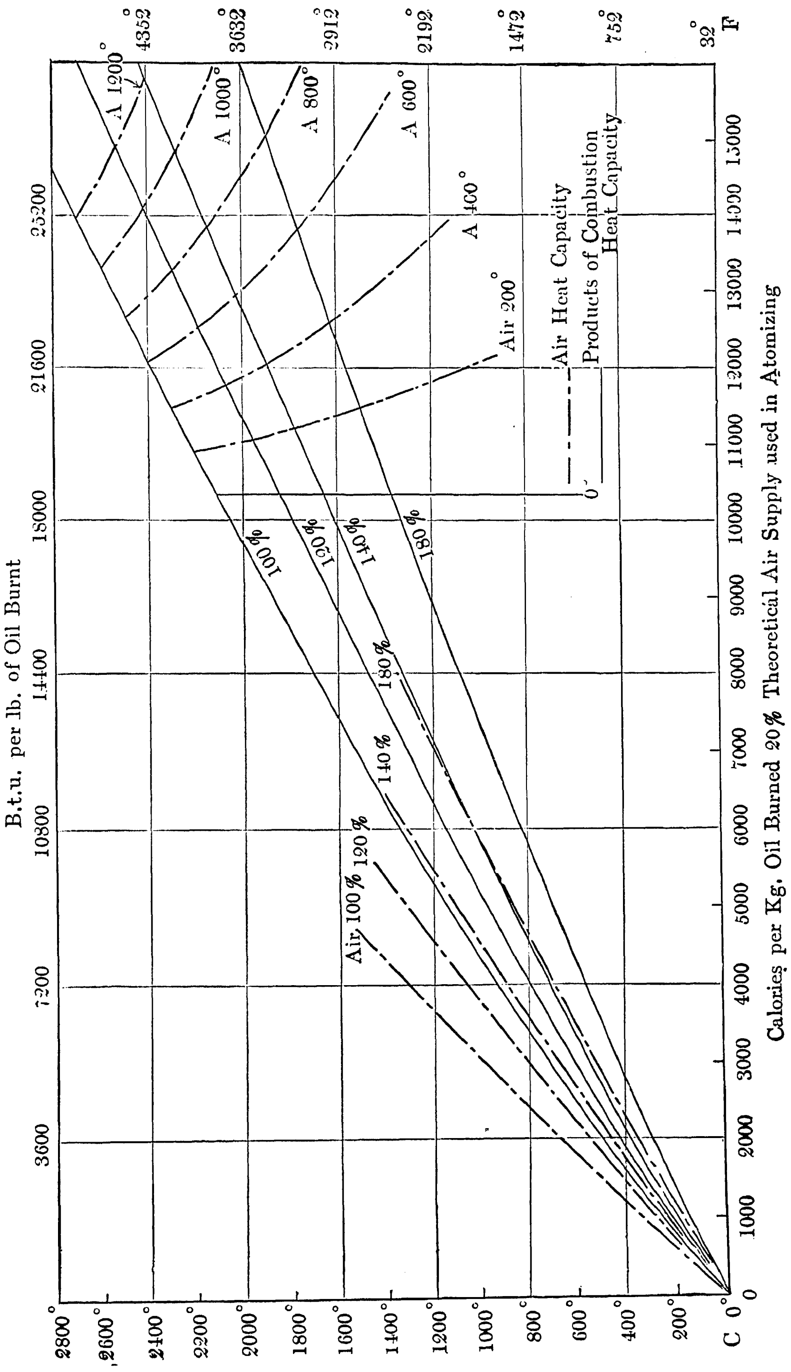


FIG. 189.—Heat Capacity and Calorific Intensity Curve of Fuel Oil No. 17, Air Atomizing. For composition, volume of products of combustion and their composition refer to Table 19.

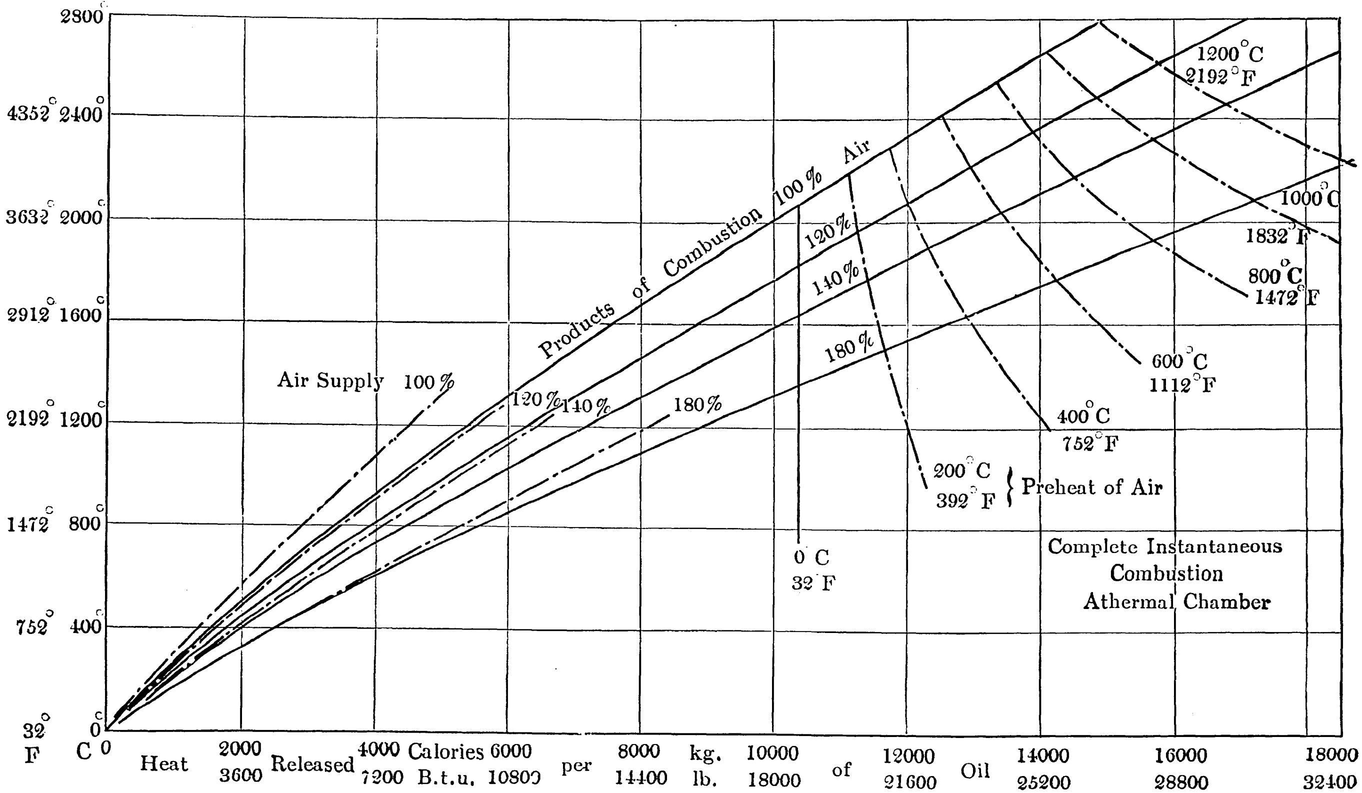


FIG. 190.—Heat Capacity and Calorific Intensity Curve of Fuel Oil No. 17, Mechanical Atomizing. For composition, volume of products of combustion and their composition refer to Table 19.

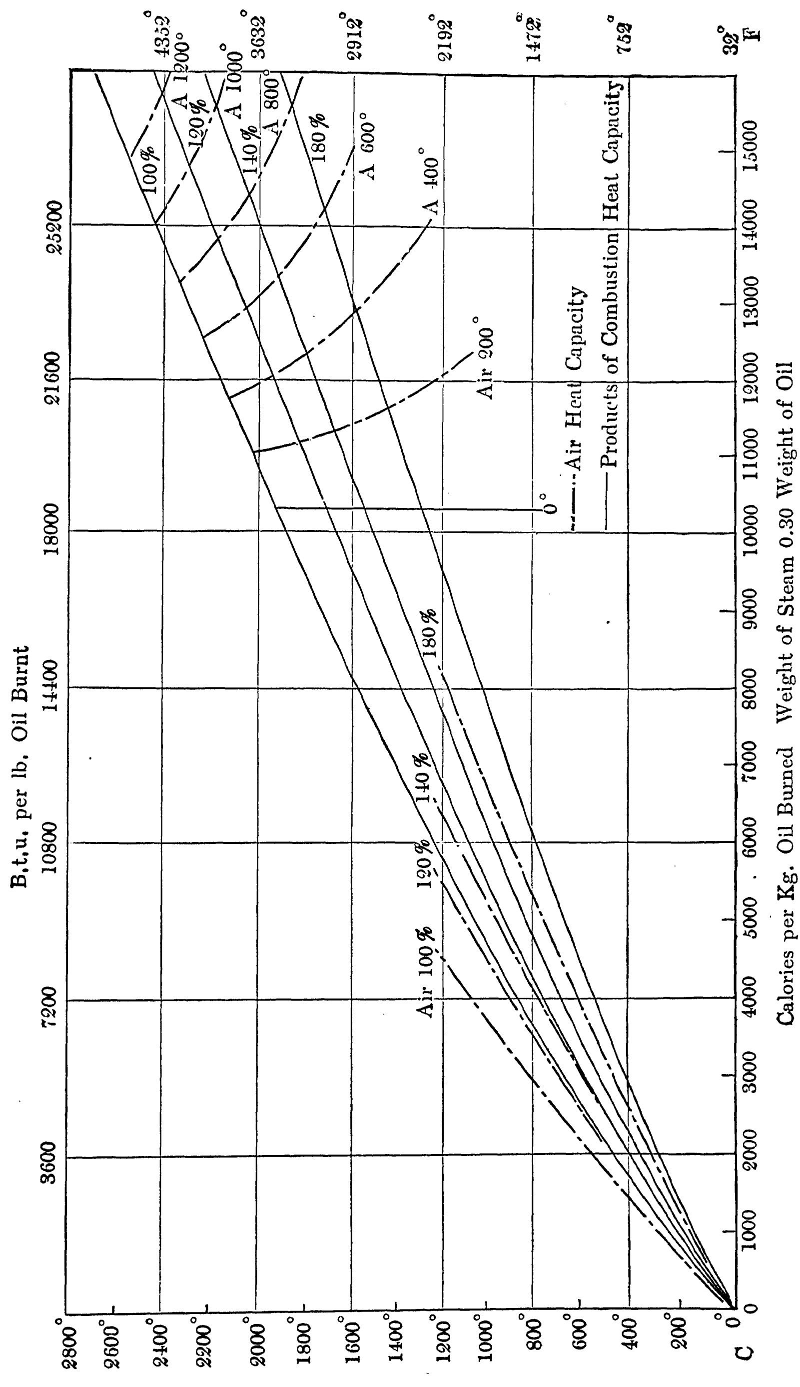


FIG. 191.—Heat Capacity and Calorific Intensity Curve of Fuel Oil No. 17, Steam Atomizing. For composition, volume of products of combustion and their composition refer to Table 19.

TABLE 20

Graphical Comparison of Fuels

		Low	High
Coal tar, by-product oven.	B.t.u. per pound =	16,042	16,076
	Calories per kg =	8,913	8,931

Specific gravity = 1.165 = 9.7 lb per gallon.

Mechanical Atomizing

Composition by Weight		Products of Combustion							
		Cubic Meters per Kg				Cubic Feet per Pound			
		Air Supply in Percentage of Theoretical Requirements							
		100%	120%	140%	180%	100%	120%	140%	180%
H_2	6.00								
C	86.82								
S	0.75								
O_2	3.11	0.00	0.78	1.57	3.13	0.00	12.54	25.07	50.14
CO_2	......	3.26	3.26	3.26	3.26	52.14	52.14	52.14	52.14
H_2O	3.10	1.42	1.42	1.42	1.42	22.76	22.76	22.76	22.76
N_2	0.12	15.66	18.79	21.92	28.18	250.78	300.95	351.10	431.30
Ash	0.10								
Total	100.00	20.34	24.25	28.17	35.99	325.68	388.39	451.07	556.34
Air supply..		19.57	23.48	27.39	35.22	313.43	376.30	438.80	564.08
Weight:									
kg per m^3		1.337			1.317				
lb per cu ft						0.0834			0.0822
		Percentage Basis (Wet)				Percentage Basis (Dry)			
O_2		0.00	3.23	5.56	8.70	0.00	3.43	5.86	9.06
CO_2		15.98	13.42	11.55	9.04	17.20	14.25	12.76	9.41
H_2O		6.99	5.86	5.05	3.95				
N_2		77.03	77.49	77.84	78.31	82.80	82.32	81.38	81.53

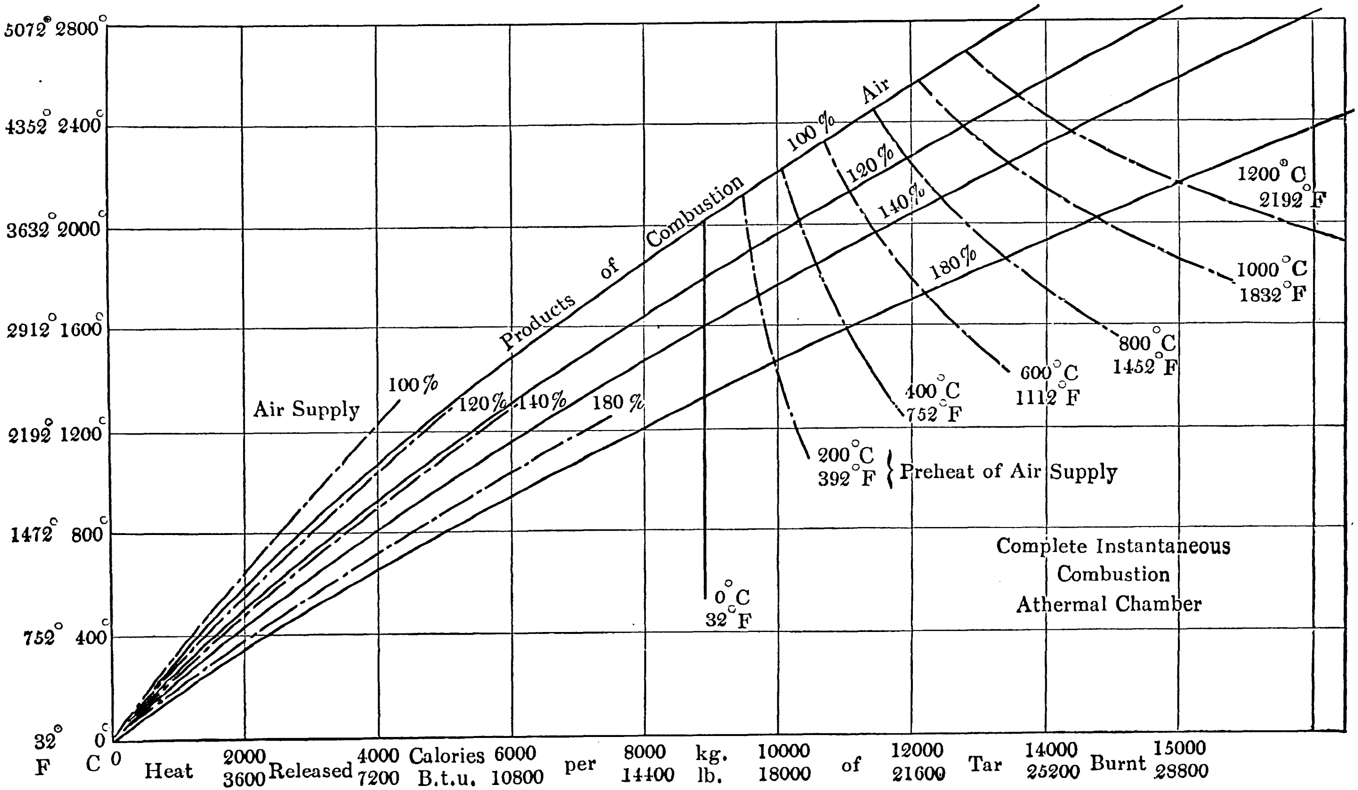


FIG. 192.—Heat Capacity and Calorific Intensity Curve of By-product Coal Tar, Mechanical Atomizing. For composition, volume of products of combustion and their composition refer to Table 20.

TABLE 20—(*continued*)

With steam atomizing, the weight of steam being 0.30 the weight of the tar, the following changes in the waste gases will be produced:

	Cubic Meters per Kiligram				Cubic Feet per Pound			
	Air Supply in Percentage of Theoretical Requirements.							
	100%	120%	140%	180%	100%	120%	140%	180%
H_2O	2.17	2.17	2.17	2.17	34.73	34.73	34.73	34.73
Total....	21.09	25.00	28.92	36.74	337.65	400.36	463.04	568.31
	Percentage Basis (Wet)							
O_2	0.00	3.13	5.42	8.52				
CO_2	15.43	13.00	11.25	8.85				
H_2O	10.29	8.68	7.50	5.90				
N_2	74.28	75.19	75.83	76.73				

When the weight of the atomizing steam is 0.60 time the weight of the tar, the waste gases change as follows:

	Cubic Meters per Kilogram				Cubic Feet per Pound			
	Air Supply in Percentage of Theoretical Requirements							
	100%	120%	140%	180%	100%	120%	140%	180%
H_2O	2.91	2.91	2.91	2.91	46.68	46.68	46.68	46.68
Total....	21.83	25.74	29.66	37.49	349.60	412.31	474.99	580.26
	Percentage Basis (Wet)							
O_2	0.00	3.04	5.28	8.35				
CO_2	14.90	12.64	11.00	8.68				
H_2O	13.35	11.32	9.83	7.78				
N_2	71.75	73.00	73.89	75.19				

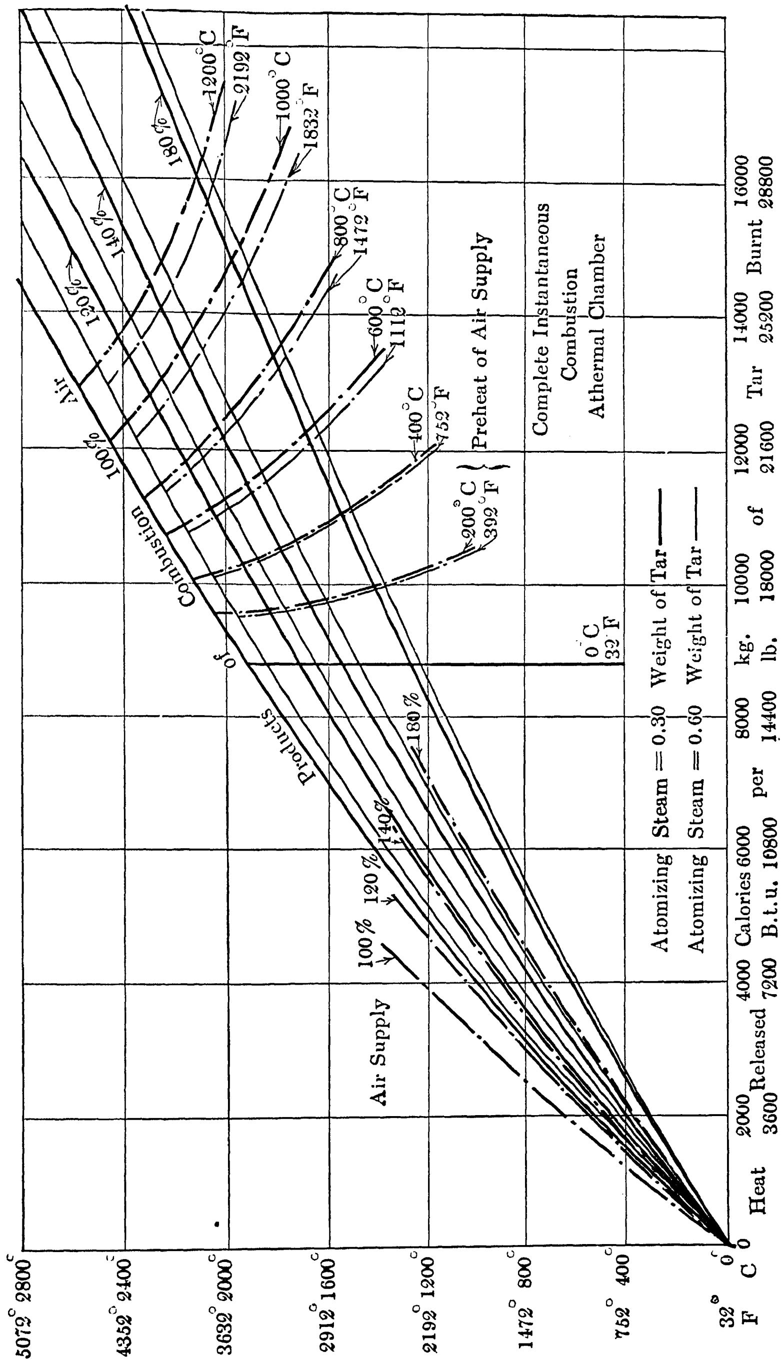


FIG. 193.—Heat Capacity and Calorific Intensity Curve of By-product Coal Tar, Steam Atomizing. For composition, volume of products of combustion and their composition refer to Table 20.

TABLE 21

GRAPHICAL COMPARISON OF FUELS

		Low	High
Pittsburgh coal, pulverized, No. 21.	B.t.u. per pound =	13,336	13,806
	Calories per kg =	7,409	7,671

Composition by Weight as Fired		Products of Combustion — Cubic Meters per Kilogram — Air Supply in Percentage of Theoretical Requirements				
		100%	120%	140%	180%	260%
H_2	4.87	....	....	....	....	....
C	73.28	....	....	....	....	....
S	1.77	....	....	....	....	....
O_2	5.63	0.00	0.31	0.63	1.26	2.51
CO_2	......	1.38	1.38	1.38	1.38	1.38
H_2O	2.28	0.57	0.57	0.57	0.57	0.57
N_2	1.35	6.31	7.56	8.82	11.33	16.35
Ash	10.82	....	....	....	....	....
Total	100.00	8.26	9.82	11.40	14.54	20.81
Air supply......		7.84	9.41	10.98	14.12	20.39
Weight:						
kg per cu m...		1.341	....	....	....	1.341
		Cubic Feet per Pound				
O_2...........		0.00	5.03	10.06	20.11	40.22
CO_2..........		22.03	22.03	22.03	22.03	22.03
H_2O.........		9.08	9.08	9.08	9.08	9.08
N_2...........		101.05	121.12	141.25	181.45	261.90
Total.........		132.16	157.26	182.42	232.67	333.23
Air supply.....		125.66	150.80	175.90	226.20	326.80
		Percentage Basis (Wet)				
O_2...........		0.00	3.19	5.51	8.66	12.07
CO_2..........		16.66	14.03	12.07	9.47	6.61
H_2O.........		6.93	5.82	5.02	3.93	2.75
N_2...........		76.41	76.96	77.40	77.94	78.57
		Percentage Basis (Dry)				
O_2...........		0.00	3.39	5.80	8.99	12.41
CO_2..........		17.91	14.87	12.71	9.86	6.80
N_2...........		82.09	81.74	81.49	81.15	80.79

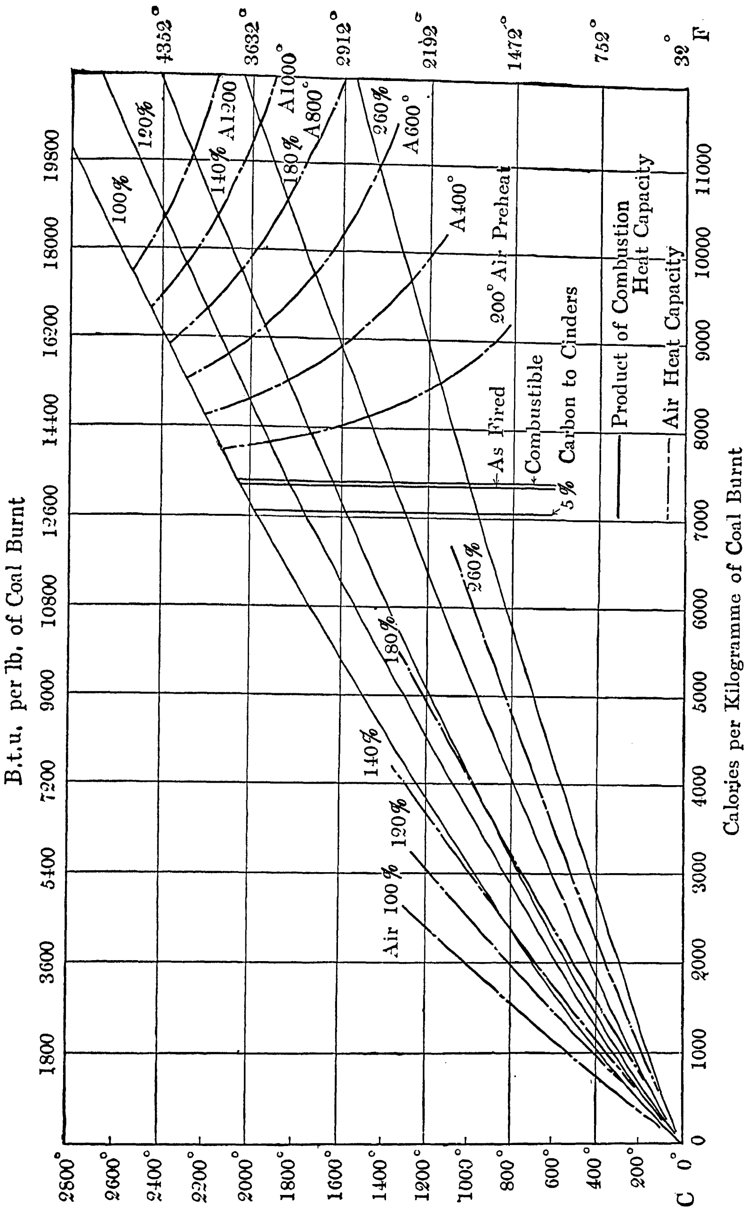


FIG. 194.—Heat Capacity and Calorific Intensity Curve of Pittsburgh Coal No. 21, Pulverized. For composition, volume of products of combustion and their composition refer to Table 21.

TABLE 22

Heat Capacity and Calorific Intensity Curves

Comparison of five gases arranged in the order of their calorific intensity giving their thermal value, thermal value of combustible mixture, the heat contained in their products of combustion, the volume of the air supply, the combustible mixture and the products of complete combustion.

	Blue Water Gas, No. 9	Coke Oven Rich Gas, No. 11	Natural Gas "C," No. 6	Producer Gas "HAW," No. 5	Blast Furnace Gas, No. 1
Calorific intensity:					
100% air supply, C.	1920°	1860°	1840°	1650°	1310°
F.	3488°	3380°	3344°	3002°	2390°
180% air supply, C.	1340°	1295°	1230°	1225°	1050°
F.	2444°	2363°	2246°	2237°	1922°
Calories per m³ of gas	2585	4180	8260	1482	835
B.t.u. per cubic foot of gas	291	468	925	167	93
Combustible mixture, 100% air supply: Calories per m³	917	776	800	636	497
B.t.u. per cubic foot	103	87	90	72	55
Products of combustion, 100% air supply: Calories per m³	1050	823	800	693	546
B.t.u. per cubic foot	119	92	90	78	61
Combustible mixture, 180% air supply: Calories per m³	618	469	464	331	376
B.t.u. per cubic foot	70	53	52	37	42
Products of combustion, 180% air supply: Calories per m³	680	486	464	346	401
B.t.u. per cubic foot	77	55	52	39	45
Volumes per 100 volumes of gas burned completely of:					
Air supply, theoretical 100%	182	439	933	133	68
Combustible mixture, theoretical	282	539	1033	233	168
Products of combustion, theoretical	245	508	1032	214	153
Air supply, 180%	318	791	1681	348	122
Combustible mixture, 180% air supply	418	891	1781	448	222
Products of combustion, 180% air supply	380	860	1780	429	208

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ALPHABETICAL INDEX

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