536.33 : 532.517 : 621.181.016

An Investigation of Radiant Heat Exchange in Boiler*

By Tatzuo HIROSE** and Akiharu MITUNAGA***

There has been no satisfactory method to obtain the distribution of radiant heat on water-cooled surfaces of a boiler furnace. This distribution may depend heavily upon the flow pattern of burning gas. From this point of view, the present paper relates to a method of defining the distribution of gas temperature and radiant heat by estimating the flow pattern in a furnace, leaving aside a number of unknown factors of combustion and thermal characters of burned gas. Result of the present calculation for a gas burning boiler whose distribution of radiant heat was measured by Mumford and Corey gives a good qualitative agreement with their measurement.

1. Introduction

The radiant heat is very important for the form of a furnace and arrangement of superheaters. But in spite of its importance, it is difficult to treat radiant heat exchange in a furnace theoretically.

As for the study of absorption coefficients of gases, there was a contribution by Hottel⁽¹⁾. Concerning radiant heat exchange of boiler, there were studies by Münzinger, Ishigai⁽³⁾, Ledinegg⁽⁴⁾, Yamasaki⁽⁵⁾, and experiments on real boilers have been made by the ASME Committee since 1945⁽⁶⁾⁽⁷⁾.

In many of studies for radiant heat exchange in a boiler furnace, burned gases and flames were not treated separately, and it seems inevitable to treat so because of their complexity.

The distribution of absorbed radiant heat on furnace walls of the experiments by the ASME Committee, was complex, and the Committee did not solve analytically the distribution.

Yamasaki obtained the frequency tables of absorbed heat quantity on the furnace walls, based on the experiments by the ASME Committee, and found that these forms of distribution were similar in spite of differences of forms of combustion chambers or different operations, and achieved the excellent results as to the total heat absorbed with a model of combustion chamber that had a flame similar in shape to

the chamber. But using this method, differences of absorbed heat between walls etc. can not be measured.

Hottel and Cohen discussed a radiant heat exchange from gas whose temperature was not uniform, but calculation was not carried out about a real combustion chamber, because of impossibility of determining a gas flow⁽⁸⁾.

To express a state of combustion chamber there are such difficulties that absorption coefficients of luminous flame can not be satisfactorily predicted even by the comprehensive study of international committee⁽²⁾. But, chemical reaction also takes place along a flow, and the most important defect in preceding studies seems to come from the fact that a flow cannot be calculated.

From the point of view above mentioned, we at first studied a flow of ordinary temperature without chemical reactions. In recent years, together with the development of fluid amplifier, flows that take place when a jet emerges into an enclosed space have been actively studied⁽⁹⁾. We obtain the principal part of a flow using several useful theories in fluidics. Among the experiments by the ASME Committee, there was an experiment of a natural gas fired boiler. Getting rid of uncertainty of luminous flame etc., we used the above mentioned boiler as a model, and under the conditions similar to its scale, we calculated the principal part of a flow of a rectangular shaped combustion chamber, and on the basis of this flow we calculated the distribution of radiant heat exchange. As for the theory of radiant heat exchange, the method of Hottel and Cohen was used.

* Received 14th November, 1969.

** Professor, Faculty of Engineering, Osaka University, Toyonaka.

*** Assistant, Faculty of Engineering, Osaka University.

The results of calculations were remarkably improved for the distribution of radiant heat (not for the total heat absorbed), and proved to resemble the distribution in an experiment of the real gas fired boiler, compared with the result by a model of uniform gas and with the result by a model in which the chamber includes a high temperature flame similar in shape to the chamber at its center. This is such a very complicated problem, that several rough assumptions used here are permissible.

2. A flow in a combustion chamber

We consider a combustion chamber as two-dimensional and gases at normal temperature. There are several theories that are applied to fluid amplifiers about reattachment of a free jet, and we can show here that the same theories can be applied to our case where a jet is at right angles to a plate. It is assumed that gases which issue from a slit have the same velocity profile as that of a free jet, and equations which R.A. Sawyer used are employed⁽¹⁰⁾. As for a free jet, many studies have been made, but none of them is complete⁽¹¹⁾⁽¹²⁾.

2.1 The velocity profile of a free jet, the rate of entrainment of a curved jet

A jet is divided into two regions, initial and main. Let coordinates, velocity, a slit be such as in Fig. 1. When the velocity is uniform at the slit, the following holds:

$$\frac{u}{U} = \frac{1}{\sqrt{\pi}} \int_{-\infty}^{\xi} e^{-\xi^2} d\xi \quad (1)$$

$$\xi = 12.0 \frac{Y}{X} \quad (2)$$

and on a line parallel to the symmetric axis at a corner of the slit, $\xi = 0.33$. In the main region,

Fig. 1

Fig. 2

when coordinates are such as in Fig. 2, the following hold:⁽¹³⁾

$$\frac{u}{u_{\max}} = \text{sech}^2 \eta \quad (3)$$

$$u_{\max} = \frac{\text{const}}{\sqrt{x}} \quad (4)$$

$$\eta = 7.67 \frac{y}{x} \quad (5)$$

At the joint of initial and main parts the velocities must be approximately continuous. The distance X_c from the slit to the joint is given by Eq. (6), and the distance x_0 from the origin of the main part to the joint is given by Eq. (7).

$$X_c = 6.97 t \quad (6)$$

$$x_0 = 5.90 t \quad (7)$$

Then the maximum velocity on the joint becomes

$$\frac{u_{\max}}{U} = 0.954. \quad (8)$$

The rate of the entrainment and the rate of entrainment of each side of a jet are defined as Eq. (9), Eq. (10).

$$E = \frac{1}{u_{\max}} \frac{d}{dx} \int_{-\infty}^{\infty} u dy = \frac{1}{\sigma} \quad (9)$$

$$E_1, E_2 = \frac{1}{u_{\max}} \frac{d}{dx} \left(\text{volume of entrainment from one side along } dx \right) \quad (10)$$

When the line which means the boundary between the fluid issued from the slit and the entrained fluid is defined $\eta = \eta_R$, and $\tanh \eta_R$ is written as T_R , and if a jet curves with a radius of curvature R , next three equations are satisfied.

$$\int_{y_R}^{\infty} u dy = \int_{x_0}^x E_1 u_{\max} dx + E x_0 u_{\max 0} (1 - T_0) \quad (11)$$

$$\frac{x_0}{x} = \left(\frac{2E_1/E + T - 1}{2E_1/E + T_0 - 1} \right)^{1/2} \quad (12)$$

$$E_1 = \frac{E}{2} \left\{ 1 + 2.627 \frac{E}{R} \left(\frac{x_0 + x}{2} \right) \right\} \quad (13)$$

A free jet shown by Sawyer is such as above mentioned.

2.2 A symmetric line of velocity profile of a jet in a combustion chamber

When the line AB in Fig. 3 which separates the fluid issuing from the slit from the fluid sucked in attaches the plate at B, among the momentum J_1, J_2, J_3 (see Fig. 3) the next relation

Fig. 3

is satisfied, and from the geometrical relation Eq. (15) and Eq. (16) are obtained, where x_1 is the length of the symmetrical line from the origin of the main region to BB.

$$J_3 - J_2 = J_1 \sin \theta \quad (14)$$

$$(R + y_R) \sin \theta = H \quad (15)$$

$$x_1 = R\theta - 1.07t \quad (16)$$

J_1, J_2, J_3 are given as follows.

$$J_1 = \frac{4}{3} \text{Exp} u_{\max}^2 \quad (17)$$

$$J_2 = 0.5J_1(1 - 1.5T_R + 0.5T_R^3) \quad (18)$$

Fig. 4

Fig. 5

$$J_3 = 0.5J_1(1 + 1.5T_R - 0.5T_R^3) \quad (19)$$

If R satisfies all the above equations, R is the solution. But it is difficult to obtain R analytically, and a trial and error method must be used.

2.3 Calculation of flow in a combustion chamber

If the radius of curvature is known, the flow in the jet can be determined. About the parts that have never been discussed, two or three assumptions are made. Near BB in Fig. 4 where the flow curves remarkably, it is assumed that at a point where $D'F' = E'F'$, $D'F'$ in Fig. 4 corresponds to $G'F'$ and $D'F'$ in Fig. 5, and the jet is divided into two and each flows along the plate. When the jet proceeds farther, it spreads abruptly at lines IJ and I'J', and we assumed them appropriately. These two points are very rough assumptions, but compared with the assumption made by Abramovich, our assumed

Fig. 7

Fig. 6

Fig. 8

definition would be permissible. In the rest of the region discussed, we make the assumption that the flow is a potential flow, and solving the Laplace equation by relaxation method, the value of stream function is obtained.

2.4 Calculated example and experimental example

Figure 6 shows a flow when air issues from a slit of 1.1 cm width into an enclosed space, and (a) shows a calculated value, (b) showing an experimental result. The arc is shown both in (a) and (b). And the result of calculation agrees fairly well with the experimental result.

2.5 The flow in the furnace of the gas fired boiler

The boiler was that of Louisiana Power and Light Co, and is illustrated in Fig. 7. As for burners an experiment at half load was made, and in this case burners of down stage were used. We use a slit of a 0.6 m breadth as a model of the burners. A flow calculated by the above method is shown in Fig. 8.

3. The distribution of temperature and radiant heat in the combustion chamber

3.1 The method by Hottel and Cohen

For grey gas whose refractive index is 1, the emission rate dE_g from a volume dV is given by

$$dE_g = 4k dV \sigma T_G^4 \quad (20)$$

where k is the absorption coefficient, σ is Stefan-Boltzmann constant, T_G is gas temperature. The emission rate E_g from a cube of edge B is given by

$$E_g = 4kB^3\phi\sigma T_G^4 \quad (21)$$

where ϕ is the escape factor. The one way radiation flux from gas G_i to G_j is given by Eq. (22), and the flux from gas G_i to surface S_j is given by Eq. (23).

$$q_{G_i G_j} = (G_i G_j) \sigma T_{G_i}^4 \quad (22)$$

$$q_{G_i S_j} = (G_i S_j) \sigma T_{G_i}^4 \quad (23)$$

where $(G_i G_j)$, $(G_i S_j)$ are direct interchange areas and have the form $4kB \times (\text{reception factor } f_{G_i G_j} \text{ or } f_{G_i S_j})$. Hottel and Cohen gave graphically the above mentioned factors with the abscissa of kB as the special positions of gas and surface. In the limiting cases of large distance, reception factors are given in the form of equation. Using these exchange areas, if radiant heat fluxes, enthalpy fluxes, convective heat fluxes, heat accumulation, heat generated are shown with unknown temperatures, and added or subtracted, necessary equations are obtained and they showed a method to solve the equation which we did not

use.

3.2 Application of Hottel and Cohen's theory to the boiler

We divide the model of the furnace of the gas fired boiler into 72 cubes. Without exact data, we assume that gas is CH_4 , excess air is 10%, k is 0.294, total heat generated is 6.8×10^7 kcal/hr and comes from cubes 8, 14, 26, 32 equally, the volume of burnt gas is 9.9×10^4 Nm^3/hr , and specific heat of burnt gas is 0.4 kcal/ $\text{Nm}^3^\circ\text{C}$. In Fig. 9 the numbers marked on cubes are shown, and in Fig. 10 heat capacities in 10^3 kcal/hr $^\circ\text{C}$ calculated from the flow pattern of normal temperature which enter or leave cubes are shown.

For a cube in which chemical reaction occurs, an equilibrium equation is induced in the following form:

$$\begin{aligned} & -4(kB)^3(\phi - f_{G_i G_i})\sigma T_{G_i}^4 \\ & + \sum_{\substack{j=1 \\ j \neq i}}^{36} (f_{G_j G_i} + f_{G_j \bar{G}_i})\sigma T_{G_j}^4 \\ & - c_i T_i + \sum_{j=m} c_j T_j + Q_i = 0 \quad \dots\dots\dots(24) \end{aligned}$$

Fig. 9

Fig. 10

region is shown in Fig. 14. Figure 15 shows the method to obtain the distribution with Eq. (32).

$$\varepsilon = \sum \frac{(1 - e^{-kl}) \cos \alpha \cos \beta}{\pi r^2} \Delta A \quad \dots\dots\dots (32)$$

In Fig. 13 and Fig. 14, the unit is dimensionless and is the emissivity of gas radiating to surface element. Experimental results of gas fired

Fig. 12

Emissivity of gas to surface element, dimensionless
Fig. 13 Emissivity of gas to surface element, dimensionless
Fig. 14

boiler are shown in Fig. 16. The values are the differences between the surface temperature of gas side and the temperature of inner liquid.

Comparing the figures, it is found that the results from our model resemble the experimental results. The agreeable points are as follows.

i) On the upper part of the front wall, heat absorbed is small.

ii) On the side surface, the region of large heat flux runs along the flame and extends radially.

iii) On the rear wall, there is a part of heavy heat absorption at the part where burnt gas impinges the wall.

A large difference exists between the calculated values and the experimental values for heat flux at the ceiling. This difference may occur because of the phenomena that the heated gas was accelerated by buoyancy, impinged on the ceiling and caused large contact heat transfer which we neglected.

5. Closing remark

The phenomena which we have assumed will be made clear, and more correct results will be

Fig. 15

Difference between surface temperature of gas side
and inner liquid °F

Fig. 16

obtained, but great many studies will be required for the purpose. Finally we thank Emerit. Prof. Uematu, Osaka Univ., who helped us, and Prof. Ishigai who gave us many hints.

References

- (1) McAdams, W.H., *Heat Transmission*, (1954), p. 82, McGraw-Hill.
- (2) The International Flame Research Committee, *Jour. Inst. Fuel*, 29, (1956), P. 23.
- (3) Ishigai, S., *Theory of Boiler* (in Japanese), (1961), p. 223, Sankaido.
- (4) Ledinegg, M., *Dampfzeugungen Dampfkessel, Feuerungen*, (1966), p. 194, Springer.
- (5) Yamasaki, Y., *Report of The Technical Research Institute* (in Japanese), (1961-3), The Central Research Institute of Electric Power.
- (6) Wheeler, R.I. and Howard, M.H., *Trans. ASME*, 72 (1950-10), p. 893.
- (7) Munford, A.R. and Corey, R.C., *Trans. ASME*, 74 (1952-10), p. 1191.
- (8) Hottel, H.C. and Cohen, E.S., *A. I. Ch. E. J.*, Vol. 4, No. 1 (1958), p. 3.
- (9) Osaki, S., Hara, Y., *Introduction of Fluidics* (in Japanese), (1967), p. 99, The Nikkan Kogyo.
- (10) Sawyer, R.A., *Jour. Fluid Mech.*, Vol. 17, No. 4, (1963-2), p. 481.
- (11) Niitu, Y., and Kato, T., *J.S. Domestic and Sanitary. E.* (in Japanese), 32, (1958-10), p. 587.
- (12) Trapani, R.D., *Fluerics*, 22, Harry Diamond Laboratories, 1966.
- (13) Schlichting, H., *Boundary Layer Theory*, (1962), p. 590, McGraw-Hill.
- (14) Abramovich, G.N., *The Theory of Turbulent Jet*.
- (15) Jung, R., *Mitt. V.G.B.*, Vol. 90 (1964-6), p. 143.

Discussion

S. ISHIGAI (Osaka University):

(1) It is interesting that the calculated results of this report agree with the experimental results on the whole. The authors guess that the large heat flux at the ceiling in the experiment by the ASME Committee is due to the influence of heat transfer by convection, but such a large heat flux seems to be caused by heat radiation. The assumption in author's calculation that heat is equally generated in cubes 8, 14, 26, and 32 is controversial. It is felt that heat is actually generated in some more spaces. It would be more reasonable if the similar calculations were made for several conditions of heat-generating space and absorption coefficient, and then the condition which agrees best with the experimental distribution were assumed to represent the actual state.

E. NISHIKAWA (Osaka University):

(2) The physical reason why a jet must be bent is lacking in the report. Show its explanation.

(3) The constant 7.67 in Eq. (5) is obtained from the value of a free jet, and this decides the rate of entrainment. It seems that in the theory of fluid amplifiers this value is recognized to be more flexible, is there any influence of this value?

(4) In the part in which a flow is regarded as a jet, large increase of volume exists in a real boiler. How do the authors consider about the point above mentioned?

K. AKAGAWA (Kobe University):

(5) Stream lines of the combustion gas have been obtained with the assumption of two dimensional jet. Can jet issued from real gas burners be approximated well by a two-dimen-

sional jet? (If it cannot be approximated), is it meaningful to investigate so strictly the curvature of the jet?

(6) There are large differences between the distances that a two-dimensional jet, a circular jet, and an annular jet can reach as shown in Append-Fig. 1^(*). How do you consider about this point concerning a flow pattern in a combustion chamber? Are not flow patterns in Fig. 16, Fig. 18 affected by the flow quantity?

Y. YAMASAKI (Technical Lab., Central Reserch Inst. of Electric Power Industry):

(7) What do you think of the effect of mutual interference of the jet from each burner and the interference between a jet and a wall?

(8) Did you consider the influence of

Append.-Fig. 1

(*1) Taga, M., and Akagawa, K., *Trans. Japan. Soc. Mech. Engrs.* (in Japanese), Vol. 31, No. 221 (1965-1), p. 105.

refractory walls in the calculation of radiation?

(9) The largest heat flux on the ceiling (experimental value), which was assumed to be attributable to convection in this study, seems so large that it cannot be explained only by the convective heat transfer of burnt gases. Please show the foundation of the assumption.

Authors' closure

(1) We ourselves thought that better results would be obtained by the method which you pointed out, but in the paper (7), it was written as follows. "Very intense combustion starts at the burner and is completed 9 to 11 feet from the burner" (7), p. 1208. We thought the assumption which differed from this description could not represent the fact, and reported the results of calculations, leaving aside the unreasonable point. As the distribution of degree of combustion is an unsolved problem, the method including the distribution of absorption coefficient which you pointed out must be correct as a rule.

(2) It is when there are two stable situations of a jet that jet curve is a big problem. Even in that case, the reason why a jet must curve is not clarified. We also used the same premise. Pressure difference exists between the two sides.

(3) 7.67 is an experimental value, and it must be determined by many experiments, now 7.67 agrees fairly well with the experiments.

(4) The influence of temperature on a flow pattern is an interesting problem, but we do not think there is any strict solution. As the value of the velocity is not involved in Eq. (14), when a flow becomes hot, is accelerated and expanded, it seems that the curvature is not much influenced. For the reason above mentioned and the fact mentioned in the literature (7), we do not think our assumption is so far from the fact.

(5) Concerning the question to what degree a two-dimensional model can represent the real flow, a general conclusion can not be obtained easily. In our report, the method to calculate curvature etc. is given, because a thorough investigation even for a two dimensional

model has never been done.

(6) There are several problems in applying the study of Taga and Akagawa to our case.

a) Because detailed descriptions of burners were not in the report of ASME, we had to assume the dimension of burner, etc.

b) When annular jets are arranged in a row, many points, for example the interference of jets, cannot be calculated. The flow patterns in Fig. 6, Fig. 8 are not affected by flow rate. The reason is the same as for a free jet. It is impossible, we think, to consider the influence of solid walls, because the apparent turbulent viscosity is a complex function of locations near solid walls.

(7) Ignoring the interference, we treat the flow as two-dimensional. The interference between two-dimensional jets was reported recently^(*2), but concerning the interference between a jet and a solid wall, data seem insufficient.

(8) There were the parts where refractory walls were exposed in the furnace, but we ignored them and treated the walls as black. When the reflection at walls is considered, the solution can be also obtained theoretically [literature (8), and (*4)]. But to do so, unknowns in simultaneous equations increase, and for each coefficient of equations it is necessary to solve a matrix of a large order, and actually this calculation is impossible.

(9) The assumption that the rate of heat flux at the ceiling is large because of convection has not any satisfactory foundation, and this as well as the problem of the influence of buoyancy to a jet is now being studied. The coefficient of heat transfer in a similar flow was studied^(*3), but it is uncertain whether it is correct or not to apply its results to the case where the turbulence is large and the wall is made of pipes.

(*2) Tanaka, Y., *Trans. Japan. Soc. Mech. Engrs.* (in Japanese), Vol. 35, No. 274 (1969-6), p. 1257.

(*3) Kumada, M., and Mabuchi, Y., *Trans. Japan. Soc. Mech. Engrs.* (in Japanese), Vol. 35, No. 273 (1969-5), p. 1053.

(*4) The Research Committee of Heat Transfer by Radiation, *The Recent Development Concerning Heat Transfer by Radiation* (in Japanese), (1966), Japan. Soc. Mech. Engrs.