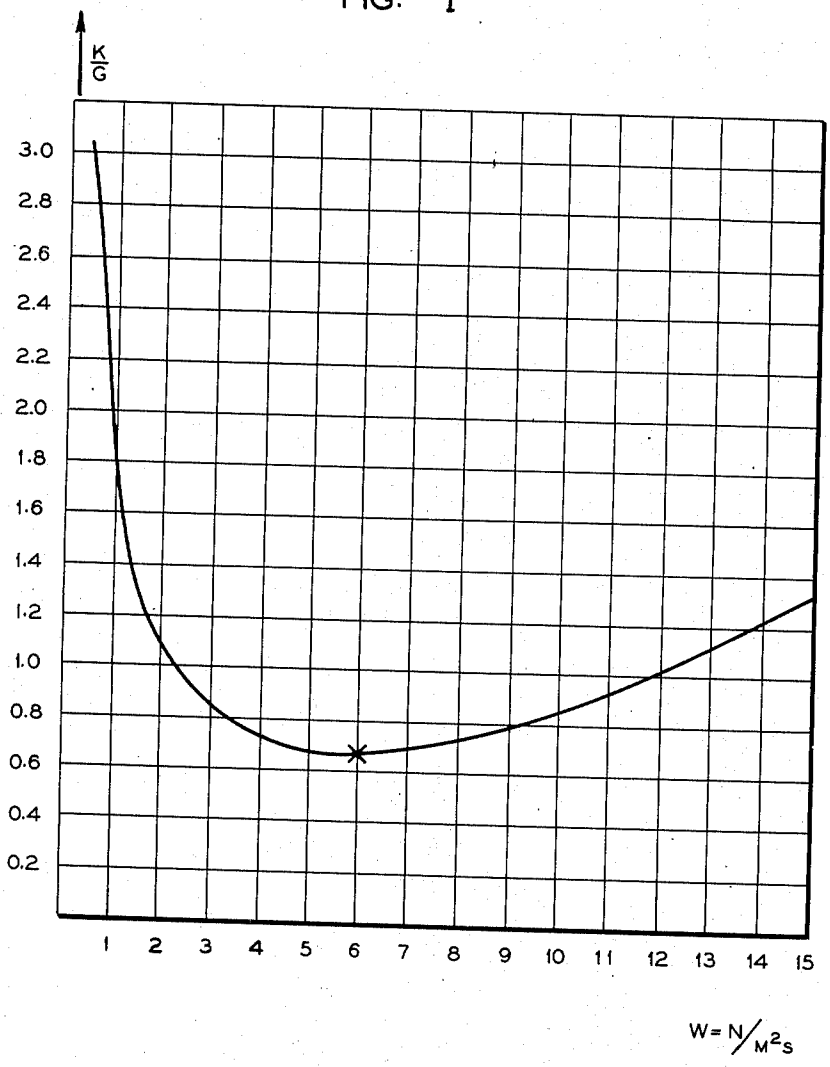
Dec. 2, 1952 J. A. RYDBERG 2,619,942

WATER WALL BOILER WITH UNDULATING FLUE

Filed Feb. 1, 1949

3 Sheets-Sheet 1

FIG. 1



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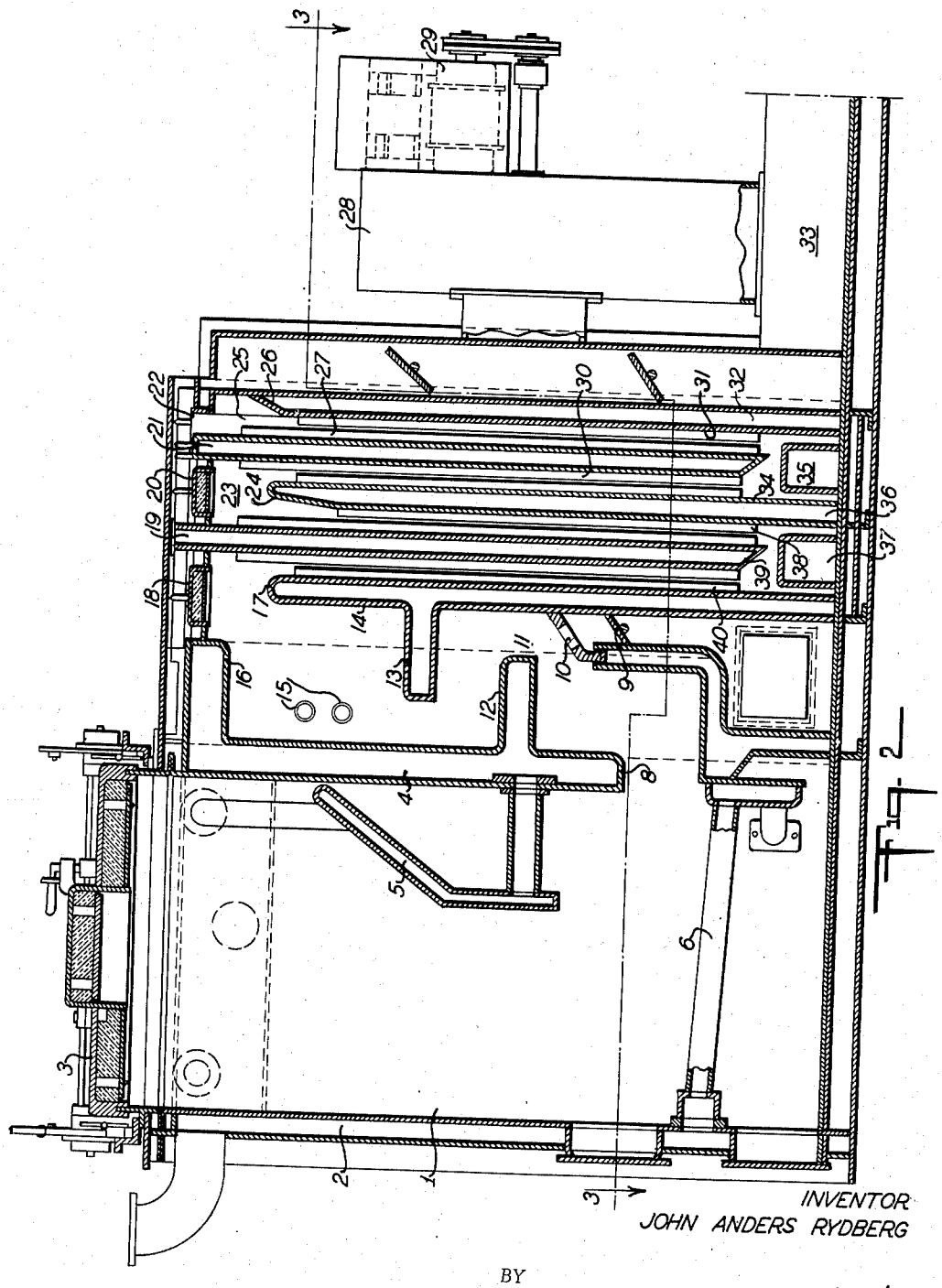


Fig. 2

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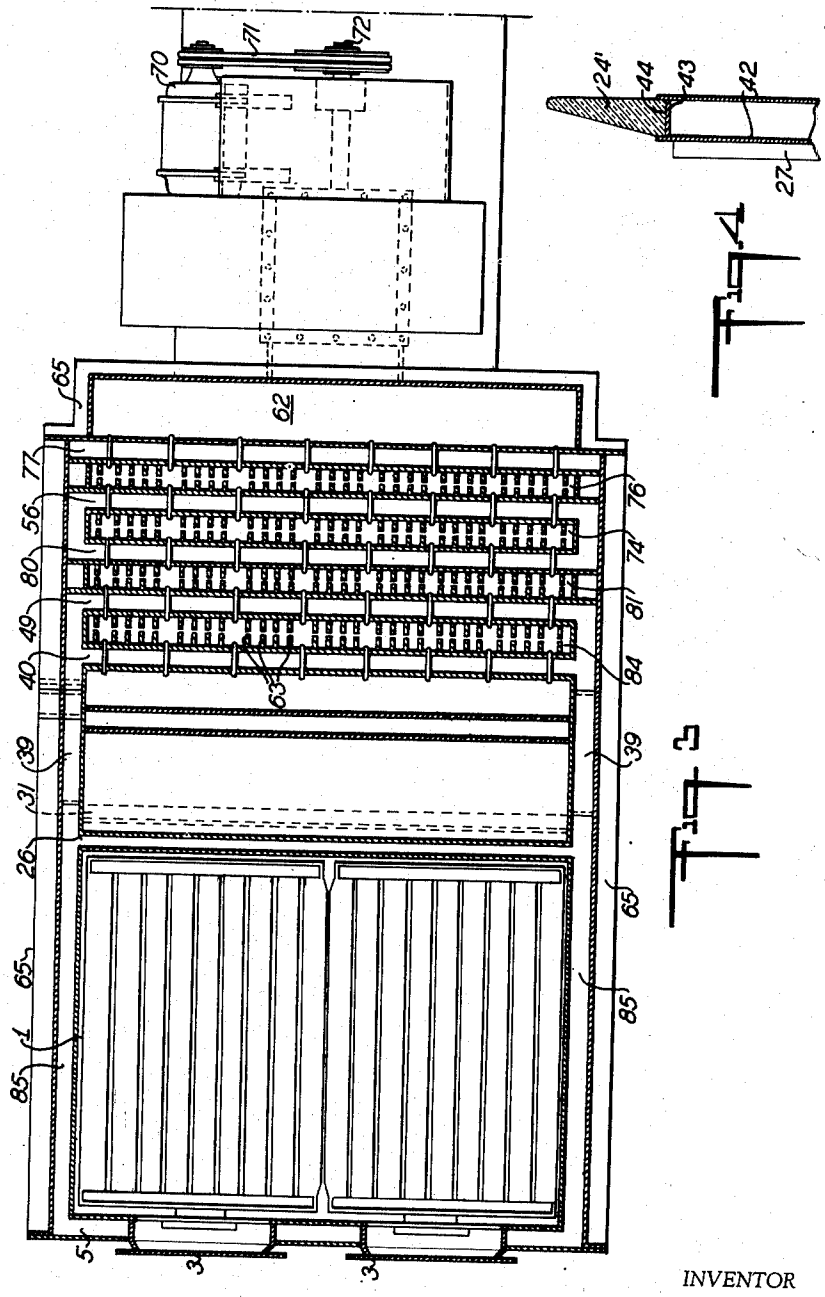


Fig. 4

Fig. 5

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UNITED STATES PATENT OFFICE

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WATER WALL BOILER WITH UNDULATING FLUE

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Application February 1, 1949, Serial No. 73,890
In Sweden January 30, 1948

5 Claims. (Cl. 122—120)

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In known construction of heating boilers, these consist of a combustion chamber and a flue system connected with same, in which water-filled partitions connected with the water space of the heating boiler conduct the flue gases in a zigzag course prior to their discharge into the flue gas exit, for instance a chimney. This invention is related to that disclosed in application Serial No. 73,888, filed February 1, 1949.

The draft required for a heating boiler is usually generated by means of a chimney. In certain cases, however, it has been suggested that the draft should be improved by means of a fan. Such a fan has also been used for overcoming excessive resistance in the grate or fuel layer. Any increase in the flow rate of the flue gas current in the boiler flues was not desired beyond the normal gas speed resulting from a good chimney and other satisfactory draft conditions, which can be stated to be less than $W=1.5N/m.^2s.$ where W is the gas flow rate in cubic meters per square meter of gas channel flow section per second and the normal gas volume N is expressed as cubic meters of gas at 0° C. and 760 mm. of Hg (normal cubic meters).

In heating boilers with a fan it has been found to be economically advantageous to increase the gas flow rate in the flues above the hitherto usual rate. At an increasing gas flow rate it will be found that the heat transfer coefficient of the heating surface increases, and thus the size of heating surface required for a certain purpose decreases. The dimensions and space required for the heating boiler can thus be reduced.

It can be proved, however, that if the gas flow rate increases above a certain limit, the cost of power for the fan will increase faster than the cost of the heating surface decreases. This invention is based on the knowledge that a higher gas flow rate is suitable and that such a rate can be determined as the most economical, i. e. a rate, at which the total of operating cost for generating the draft and of the annual amortization for the heating surface required can be determined as a minimum.

The pressure drop P in the heating boiler flues can be expressed by formula:

$$P=A.W^n \tag{1}$$

The heat transfer coefficient k for the convection heating surface is approximately:

$$k=B.W^u \tag{2}$$

In these expressions, A, B, n and u are constants, and W = the gas flow rate in the boiler flues in $N/m.^2s.$ The formula for obtaining the pressure drop P given above is based on the fact that such pressure drop is dependent on the gas flow rate W in the flues. Experiments have shown that P is proportional to an exponential function

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of the gas flow rate W as expressed by the formula where n is the exponential value by which P depends on W and in which A is a factor of proportionality. As in the formula for k the terms A, B, n and u are numerical values. Those for n and u have been found by experimentation to be 1.8 and 0.8, respectively and for B to be

$$\frac{3.8}{d^{0.25}}$$

where d is the hydraulic diameter of the flue.

The total cost of the power for the fan and of the annual amortization for the convection heating surface can be expressed thus:

$$K=P.G.r.Z+E.a \tag{3}$$

where

G=gas volume per time unit
r=cost of power
E=convection heating surface
a=annual amortization per m.^{2} heating surface
Z=operating time per year.

For cooling down in the boiler flues there is the following relation:

$$\log_e \frac{t_1-t_v}{t_2-t_v}=\frac{k.E}{G.C_p} \tag{4}$$

where

t_1=temperature of flue gases before contact with the convection heating surface
t_2=temperature of flue gases upon leaving the convection heating surface
t_v=temperature of heating surface
C_p=specific heat of flue gases.

If it is desired to cool down the flue gases to a certain value, independently of variations in gas flow rates and in heating surface, this can be expressed by:

$$\frac{k.E}{G}=C \tag{5}$$

where C is a constant.

The above expression for the cost k may now be written by substituting (1), (2), and (5) in (3).

$$K=A.W^n.G.r.Z+\frac{C.G}{B.W^u}.a \tag{6}$$

If cost K is derived with regard to gas flow rate W for a constant value of the gas volume per time unit, the following is obtained:

$$\frac{dK}{dW}=Z.A.r.GnW^{n-1}-\frac{C.a.G.u}{B}.\frac{1}{W^{u+1}} \tag{7}$$

The value of gas flow rate W, which gives a minimum for cost K, is obtained by putting down 0 (zero) for this derivative. Thus the following

expression is obtained for the economical value of W:

$$(8) \qquad W_e = \left[\frac{C.a.u}{A.B.r.n.Z}\right]^{\frac{1}{u+n}}$$

My investigations have shown that one can put, with sufficient safety, 2.6 for $u+n$.

This shows that the variation in boiler amortization and power prices, a and r respectively, and the operation time Z has no considerable effect upon the economical gas flow rate W_e. I have thus succeeded in establishing that there is an economical value of gas flow rate and that this value W_e will vary comparatively slightly in the case of variations in prices and method of operating. My investigations have further shown that W_e normally is in the neighbourhood of $6N/m.^2s.$, and that this value has comparatively general validity. Around the economical value W_e for the gas flow rate, cost K is found to vary very slightly. Owing to my investigations I have found that from a practical and economical point of view it is sufficient, if the gas speed is kept within the limits (3 to 10) $N/m.^2s.$ As an example I show in Fig. 1 of the drawings a diagram of the relation between the cost

$$\frac{K}{G}$$

per year for cooling down the gas flow in N per operating hour and the gas flow rate W in $N/m.^2s.$ in the flue system. The gas flow has been assumed to enter the convection surface at 1100° C. and to leave it cooled down to 200° C., and the number of operating days of 24 hours per year of the heating boiler has been assumed to be 240, all values quite normal. The diagram shows that the cost per year has a minimum at a gas flow rate of $6N/m.^2s.$, and that the curve around this value is rather flat, which means that the gas flow rate W can be varied within the limits (3 to 10) $N/m.^2s.$ without considerably increasing the cost per year.

The invention thus relates to a heating boiler of the type described in the beginning, with a fuel storage compartment and flue system and a fan arranged between the flue system and the flue gas discharge, for instance a chimney. According to this invention, the flue gas system is constructed as a convection surface with a high heat transfer figure by locating the flue gas flow rate within the limits (3 to 10) $N/m.^2s.$ by means of suitable adaptation of the capacity of the flue gas fan in relation to the heating surface of the boiler. This invention relates in particular to a heating boiler with a gas flow rate in the flue system of $6N/m.^2s.$ or in the neighbourhood of this figure. In the case of large heating boiler plants of, for instance, 1 million kilogram calories per hour for larger buildings or similar purposes, the convection heating surface can be reduced to half size with good economy. It is evident that hereby a considerable saving in space and cost is attained. Moreover, it has become possible, owing to this invention, to exceed the upper limit for heat production in one single boiler unit, which formerly had been fixed in practice with regard to the possibility that they might be placed in, for instance, a block of buildings.

In order to render possible such high economical gas flow rates, it is suggested, according to this invention, that the flue or flues of the heating boiler prior to their discharge into the smoke gas exit and where the flow path changes in direction should be designed as diffusers, in order to reduce the gas flow rates prior to the smoke gas discharge or change of direction respectively. Such diffuser action, i. e. conversion of the energy of movement into energy of pressure, can be obtained by having the water-filled walls or partitions made with ends diverging in the direction of the gas flow at the point of the smoke gas discharge or the point of the above mentioned change of direction respectively. The flue system should preferably be made up of flat water walls.

By the arrangement proposed the pressure losses in the flue bends are reduced to acceptable values. Without this arrangement the combined pressure drops in the bends would be excessive which is not desirable, as it is only pressure drops owing to friction along the heating surface that efficiently contribute to an improvement in the transfer of heat.

A heating boiler constructed according to this invention is shown as an example in the drawings. Figure 1 is a diagram showing the relation between the cost K/G per year for cooling down the gas flow in cubic meters per operating hour and the gas flow rate W in meters per second. Fig. 2 is a longitudinal section, Fig. 3 is a sectional view of the heating boiler on the line 3—3 of Fig. 2 and Fig. 4 is a fragmentary sectional view of a modified diffuser structure.

The heating boiler shown has a fuel compartment 1, which forms a shaft for the fuel put in from the top of the boiler. Above the fuel compartment lies a removable cover 3. The vertical walls of fuel compartment 1 form water spaces 2, 4, 41, 41, of which the longitudinal ones 41, 41 continue along the entire length of the boiler, forming a connection with other water spaces of the boiler. Between the transverse water spaces 2, 4 of the fuel compartment is inserted a removable grate 6, consisting of tubes in the lower part of the compartment. This grate 6 is water-filled and is connected with the remaining water spaces of the boiler. In the rear cross wall 4 there is above grate 6 an exit 8 for combustion gases, through which they pass to a subsequent secondary combustion or flame chamber 15. In front of and above the fuel gas exit 8 there is disposed a water-filled supporting bridge 5 for the fuel. This supporting bridge produces a larger separating area 7 if small size fuel is used, which reduces the resistance at the gas exit 8 owing to lower gas flow rate in the separating area.

The secondary combustion chamber 15 is arranged between the rear wall 4 of the fuel compartment and a subsequent, likewise vertical and water-filled partition 14. These walls 4, 14 are, moreover, provided with a pair of horizontal, water-filled baffles 12, 13, extending into the secondary combustion chamber 15 and giving the fuel gas flow a winding passage 11 to the secondary combustion chamber 15. At a suitable point in this winding passage 11 there is a secondary air intake 19, and the secondary air coming in is well mixed with the fuel gases by means of the devices provided.

From the secondary combustion chamber 15 the flue gases enter the convection area of the heating boiler, which consists of a number of water-filled partitions 14, 19, 36, 21, 32, alternately projecting from the lower part of the boiler upwardly and projecting downwardly from the upper part, so that the flue gases proceed in the vertical spaces 40, 38, 30, 31 formed between them in zigzag windings before leaving the boiler

through smoke gas exit **33** and through flue gas fan **28** driven by motor **29**. The flue gases enter the flue gas system through an opening **17**, which is left by partition **14** at the top, below the roof **16** of the boiler, which is filled with water and provided with suitably arranged soot-holes **18, 20, 22**. The sides of partitions **14, 19, 36, 21, 32** facing each other are fitted with vertical flanges **27**, which face each other in pairs and divide the flue gas flow into parallel currents.

To render possible the attaining of high economical gas flow rates, the flues of the boiler, **40, 38, 30, 31**, are shaped at their ends as diffusers **39, 24, 34, 26**, in connection with every change in direction occurring, **37, 23, 35, 25**, i. e. when changing from downward direction to upward and vice versa, so that the gas flow rate is reduced before its direction is changed. This has been attained by making the ends **39, 24, 34, 26** of the partitions tapering at the bends **37, 23, 35, 25** of the flue system, so that the flues diverging gradually at these points and the pressure losses in the bends are reduced for the above mentioned purpose.

The example of design shown of the heating boiler is constructed with regard to the shaping of the heating surface so that the above stated, economically advantageous gas flow rate in the convection area of (3 to 10)N/m.^{2}s., especially 6N/m.^{2}s. or about that figure, is attained.

What I claim is:

1. A water heating boiler of the water wall type having a combustion chamber, flue gas passages of substantially rectangular cross section in said boiler defined by water filled parallel and substantially vertical partitions, said partitions alternately terminating in spaced relation to the upper and lower internal walls of said boiler, thus providing an elongated serpentine path for said gases, the free ends of said partitions diverging in the direction of gas flow to provide diffusers whereby the flow rate at these points is reduced, a flue gas outlet and a circulating fan disposed between said passages and said outlet, the capacity of said fan and the cross sectional area and arrangement of such passages being so coordinated that the rate of flow of said gases within said passages is within the limits (3 to 10)N/m.^{2}s. and the heat transfer coefficient is substantially equal to BW$^{0.8}$ and in which

N=normal gas volume in cubic meters at 0° and 760 mm. Hg.
m.^{2}=cross section of flue in square meters
s.=seconds.
B=3.8$d^{-.25}$
d=hydraulic diameter of the flue and
W=gas velocity in meters/sec.

2. A water heating boiler of the water wall type having a combustion chamber, flue gas passages in said boiler defined by water filled parallel and substantially vertical partitions, said partitions alternately terminating in spaced relation to the upper and lower internal walls of said boiler thus providing an elongated serpentine path for said gases, the free ends of said partitions diverging in the direction of gas flow to provide diffusers whereby the flow rate at these points is reduced, a flue gas outlet and means disposed between said passages and said outlet for forcibly circulating said gases, the capacity of said means and the cross sectional area and arrangement of said passages being so coordinated that the rate of flow of said gases is within the limits (3 to 10)N/m.^{2}s. and the heat transfer coefficient is substantially equal to BW$^{0.8}$ and in which

N=normal gas volume in cubic meters at 0° and 760 mm. Hg
m.^{2}=cross section of flue in square meters
s.=seconds
B=3.8$d^{-.25}$
d=hydraulic diameter of the flue and
W=gas velocity in meters/sec.

3. A heating boiler as defined in claim 1 in which the diverging ends of said partitions are constructed of refractory material.

4. A heating boiler as defined in claim 1 in which a second combustion chamber is disposed between the first combustion chamber and said flue gas passages.

5. A water heating boiler of the water wall type having a combustion chamber, flue gas passages in said boiler defined by water filled parallel and substantially vertical partitions, said partitions alternately terminating in spaced relation to the upper and lower internal walls of said boiler thus providing an elongated serpentine path for said gases, the free ends of said partitions diverging in the direction of gas flow to provide diffusers whereby the flow rate at these points is reduced, opposed vertical flanges on said partitions providing separate relatively narrow gas flow paths, a flue gas outlet and means disposed between said passages and said outlet for forcibly circulating said gases, the capacity of said means and the cross sectional area and arrangement of said passsages being so coordinated that the rate of flow of said gases is within the limits (3 to 10)N/m.^{2}s. and the heat transfer coefficient is substantially equal to BW$^{0.8}$ and in which

N=normal gas volume in cubic meters at 0° and 760 mm. Hg.
m.^{2}=cross section of flue in square meters
s.=seconds
B=3.8$d^{-.25}$
d=hydraulic diameter of the flue and
W=gas velocity in meters/sec.

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