March 30, 1948. R. M. GATES 2,438,851

PLATE ARRANGEMENT FOR PREHEATERS

Filed Nov. 1, 1943

Fig. 1

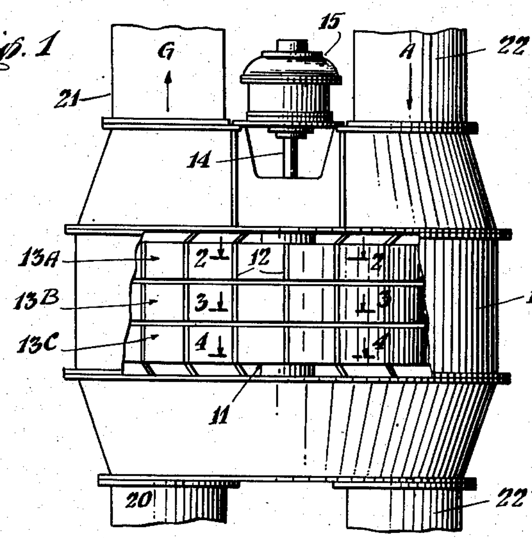

Fig. 2

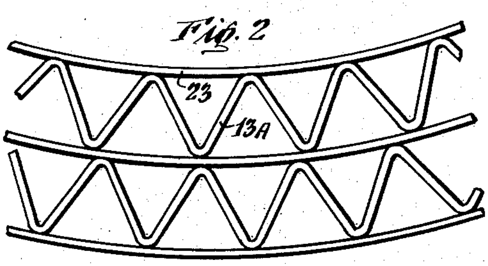

Fig. 3

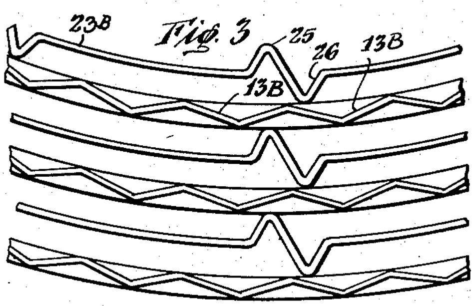

Fig. 4

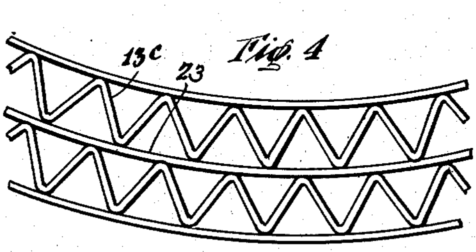

Fig. 5

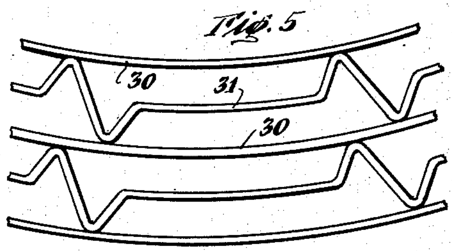

Fig. 6

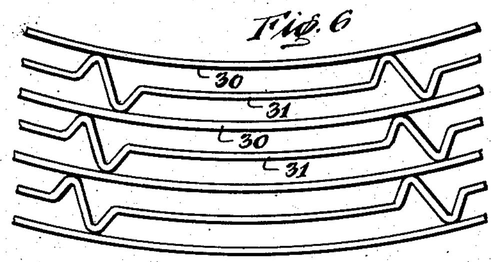

Fig. 7

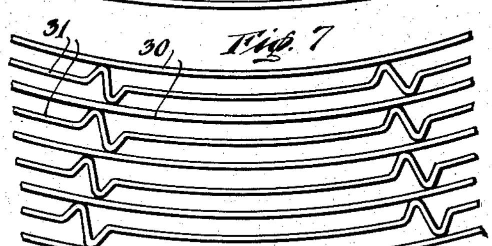

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PLATE ARRANGEMENT FOR PREHEATERS

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4 Claims. (Cl. 257—6)

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The present invention relates to improvements in regenerative heat exchangers and particularly to minimizing clogging of the gas and air passages through the heat transfer plates in an air preheater of the rotary type.

In order to obtain the desired heat transfer, most preheaters have the heat transfer plates packed in so closely that the channels through the heat absorbing surface are very narrow. When air preheaters are operated under such conditions that the temperature of the metallic heat transfer plates is at or below the dew point of the gases entering the preheater, moisture is condensed from the gases and causes corrosion of the plates. Fly ash and other material may adhere to the plates thereby reducing the free area of the gas and air channels. This may result in clogging of the channels, particularly at the "cold end" of the preheater where condensation may occur on the surfaces of the plates due to moisture in the entering air or in cooled gases leaving the preheater. The present invention contemplates minimizing such clogging and its effects by increasing the net free area for gas flow between the heat transfer surfaces at the cold end of the preheater while maintaining at other points an arrangement of heat transfer surface adapted to effect the desired heat transfer. This is accomplished by utilizing several types or arrangements of heat transfer plates so that the free gas areas at the cold end of the preheater may be different than at the hot end.

The draft and air pressure losses in an air preheater are dependent upon the velocity of the gases and air, and upon the friction factor. If the spaces at the cold end are made wider, there is a small reduction in velocity, thus tending to reduce the draft loss. It is then necessary to increase the length of the gas passages so that enough heating surface is installed to reduce the gases to the proper final temperature. The net result of these two factors is practically no change in the total resistance.

The friction factor increases rather rapidly with decrease of the spacing or mean hydraulic radius, i. e. the ratio of the area free for the passage of gases to the periphery of the gas contacted surface. If the spacing between plates at the cold end is increased with consequent increase in the hydraulic radius, the actual draft loss and air pressure loss are decreased because of the lower friction factor. Should ash deposits build upon the surface and cause clogging, the friction of more widely spaced plates remains lower than that in the closer spaced plates until

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such time as the mean hydraulic radius is decreased by the clogging to the point where it is equal to the hydraulic radius of the closer spaced plates. Since deposits of material causing clogging are usually the same in thickness, whether the plates are spaced on wide or narrow centers, it is obvious that the resistance of the preheater is always less with the plates on wide centers at the cold end. In the above discussion it has been assumed that regardless of the spacing of the plates at the cold end, the same overall heat recovery is desired. The metal temperature in the plates at the cold end is, therefore, the same, regardless of the spacing, and the possibility of clogging is the same in both cases. The effect of wider spacing on the plates is chiefly to reduce the effect of clogging on the resistance of the preheater.

In the drawings:

Figure 1 is an elevational view, partly broken away, of a Ljungstrom type regenerative air preheater embodying the present invention;

Figures 2, 3 and 4 are sectional views on correspondingly designated section lines in Figure 1 and illustrate three different types of heat absorbing surface in the rotor.

Figures 5, 6 and 7 are views corresponding to Figs. 2 to 4 but showing a modified arrangement of the heat transfer surface utilizing flat rather than corrugated or undulated plates.

In the drawings the numeral **10** designates the housing of an air preheater of the Ljungstrom type having a cylindrical rotor **11** divided by radial partitions or diaphragms **12** into a plurality of sector-shaped compartments. Each rotor compartment contains regenerative heat transfer surfaces in the form of undulated or corrugated plates **13A**, **13B**, **13C** which first absorb heat from the hot gases when passing through the gas side of the preheater and then impart the heat to air passing through the air side of the heater, as the rotor is turned slowly about its axis **14** by a motor **15** operating through suitable reduction gearing. The inlet and outlet ducts for gas are designated **20** and **21** in Figure 1 and those for air are numbered **22** and **22'** respectively. The preheater construction as described above is conventional.

In carrying out the invention undulated or corrugated plates are employed as usual for the heat transfer elements but they are mounted in two or more superimposed layers in the rotor **11**. As shown there are three layers, the elements **13A** in the uppermost layer (Fig. 2) near the cold end of the preheater having deeper corrugations than the elements 13B in the intermediate layer (Fig. 3) or in the lowermost layer (Fig. 4). In the uppermost and lowermost layers the elements 13A and 13C are maintained in spaced parallel relation by the spacing plates 23. Thus the air and gas passages through the preheater are divided into a multiplicity of small channels and those located in the zone where the uppermost layer of elements (Fig. 2) is positioned are deeper radially of the rotor than the channels in the lower zone (Fig. 4) and also are of greater cross-sectional area. The same effect is produced in the intermediate layer (Fig. 3) by forming the spacing plates 23B with notches or corrugations 25, 26 which make the radial distance between elements 13B or between them and the adjacent spacers 23B less than between elements 13A in the uppermost and greater than between elements 13C in the lower layer, the arrangement being substantially the same as in Figs. 2 and 4 except that the corrugations determining the radial width of the channels for the flow of gas and air are in the spacers 23B rather than in the elements 13B themselves.

Here, however, the aggregate gas free area is greater than in the channels of Fig. 4 while the gas contacted area of the plates 13B (and spacers 23B) is less.

With the arrangement described it is possible to pack the heat transfer plates in the lowermost layer or layers as closely together as required to obtain the desired heating effect while at the same time by utilizing plates having deeper corrugations in the cooler zones where corrosion is apt to occur clogging of the preheater is minimized. Furthermore, the heat transfer plates in the intermediate or lowermost layers may be replaced, if required, by others more closely packed in order to increase the amount of heat transfer surface and the heat recovery. Conversely if less surface is required because the temperature of the entering gases is not as high as was anticipated in designing the preheater, the plates in the lowermost or intermediate area may be replaced by others having deeper corrugations.

The invention permits the use of an optimum spacing between the plates forming the heating surface and each layer thereof in order to minimize the effect of deposits on the surface on the fluid resistances. Where deposits form on the heating surface of an air preheater, the thickness of these deposits increases as the temperature of the surface decreases. At the cold end where greatest deposits of solids on the surface take place, the optimum spacing of the plates would be that at which the deposits are prevented from bridging over the space between the plates and closing the passages. As the spacing between the plates or the hydraulic diameter increases, less force is required to prevent the deposits from bridging over and closing the spaces between the plates than when smaller spaces or hydraulic diameters are used, the force available being in the velocity of the fluids passing over the surface. It is possible to so select a spacing for any given rate of flow and any given type of deposit that bridging over by the deposit of the spaces between the plates is prevented and this spacing would be the optimum. By selecting an optimum spacing of heating surface in each layer of a preheater as covered by the invention, it is possible to select a larger height of notch for the portion of the surface where the heaviest deposit occurs or to graduate the depth of notch or spacing as operating conditions demand. Assuming that for

the portion of the surface where there is a deposit of .22 mm. thickness a height of notch of 5 mm. instead of 2.5 mm. were used, the pressure drop increase for this portion of the surface would be decreased from approximately 75% to roughly 30%.

Thus another aim of the invention is to provide the spacing between the plates in each individual layer of surface so as to obtain a minimum increase in fluid resistance over the surface due to deposits thereon, the spacing in each layer corresponding to this condition being what we term the optimum spacing.

It will be apparent that in order to vary the "hydraulic radius" there are ways other than changing the depth of plate corrugations of heat absorbing surface. For example, in a given gas passage even flat plates 30 may be utilized with spacers 31 having notches of appropriate depths to maintain them in spaced positions as appears in Figs. 5 to 7, there being fewer plates 30 separated by spacers 31 wiht deeper corrugation-like notches in the section at the cold end with the result that less of the passage is occupied by the plates 30 themselves and their spacers 31 with consequent increase in the hydraulic radius. A similar effect may be attained when using corrugated plates if instead of changing the radial depth of the corrugations their pitch is altered and the corrugations are thereby formed further apart so that the overall length or surface of each plate is decreased.

In addition it may be mentioned that for a given standard height of rotor, the desired heat recovery can be obtained by varying the radial depth of the corrugations or the height of the heating elements. In other words, it would not be necessary to have so many different rotor heights as now being used but instead vary the radial depth of the corrugations or the height of the heating surface in order to obtain the required heat recovery.

This application is a continuation of that filed in my name on December 27, 1941, under Serial No. 424,562 and now abandoned.

What I claim is:

1. In a regenerative air preheater or the like having a rotor comprising a cylindrical shell divided by radial partitions into a plurality of sector-shaped compartments for carrying regenerative heat exchange material through air and gas passages formed in the heater housing, and inlet and outlet ducts for hot gas and relatively cool air so connected to opposite ends of said passages as to provide for counterflow of the hot and cold gases through said preheater; regenerative heat exchange material comprising a plurality of superimposed layers of substantially parallel plates mounted in the rotor compartments and formed with corrugations spacing the plates and disposed generally parallel to the direction of fluid flow to form a multiplicity of channels parallel to the direction of fluid flow with the plates in the layer nearest the air inlet and gas outlet end of the rotor having corrugations of greater depth than those in the other layers so as to provide channels of greater cross-sectional area through the first mentioned layer.

2. In a regenerative air preheater or the like having a rotor comprising a cylindrical shell divided by radial partitions into a plurality of sector-shaped compartments for carrying regenerative heat exchange material through air and gas passages formed in the heater housing parallel to the rotor axis, and inlet and outlet

ducts for hot gas and relatively cool air so connected to opposite ends of said passages as to provide for counterflow of the hot and cold gases through said preheater; regenerative heat exchange material comprising a plurality of superimposed layers of substantially parallel plates mounted in the rotor compartments and formed with corrugations spacing the plates and disposed generally parallel to the direction of fluid flow to form a multiplicity of channels parallel to the rotor axis with the plates in the layer nearest the air inlet end of the rotor having corrugations of greater depth than those in the other layers; and members mounted between and contacting the plate corrugations to radially space the plates of each layer so arranged as to provide channels of greater radial depth through the layer of plates adjacent the air inlet end of the heater than in the other layers.

3. In a rotary regenerative heat exchanger for gaseous media having a casing providing passages therethrough for hot and cold gases, respectively, inlet and outlet ducts so connected to the ends of said passages as to provide for counterflow of the hot and cold gases through said exchanger, and a rotor comprising a cylindrical shell divided by radial partitions into a plurality of sector-shaped compartments regenerative material to be carried successively through said passages; said material being divided into a plurality of contiguous sections spaced axially of the rotor and each substantially filled with a plurality of undulated plates separated by intervening spaces with the plate undulations shaped so that adjacent surfaces thereof form a multiplicity of channels for flow of gas therethrough, the plate undulations being proportioned differently in different sections with the heat exchange surface presented thereby for heat exchange being least and the total area of the channels for gas

flow being greatest in the section at the cold end of the rotor.

4. In a rotary regenerative heat exchanger for gaseous media having a casing providing passages therethrough for hot and cold gases, respectively, inlet and outlet ducts so connected to the ends of said passages as to provide for counterflow of the hot and cold gases through said exchanger, and a rotor comprising a cylindrical shell divided by radial partitions into a plurality of sector-shaped compartments regenerative material to be carried successively through said passages; said material being divided into a plurality of contiguous sections spaced axially of the rotor and each substantially filled with a plurality of undulated plates separated by intervening spaces with the plate undulations shaped so that adjacent surfaces thereof form a multiplicity of channels for flow of gas therethrough, the plate undulations being proportioned differently in different sections with the heat exchange surface presented by the plates being least and both the hydraulic radius of the individual channels defined by the plate undulations and the total area for gas flow therethrough being greatest in the section at the cold end of the rotor.

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