

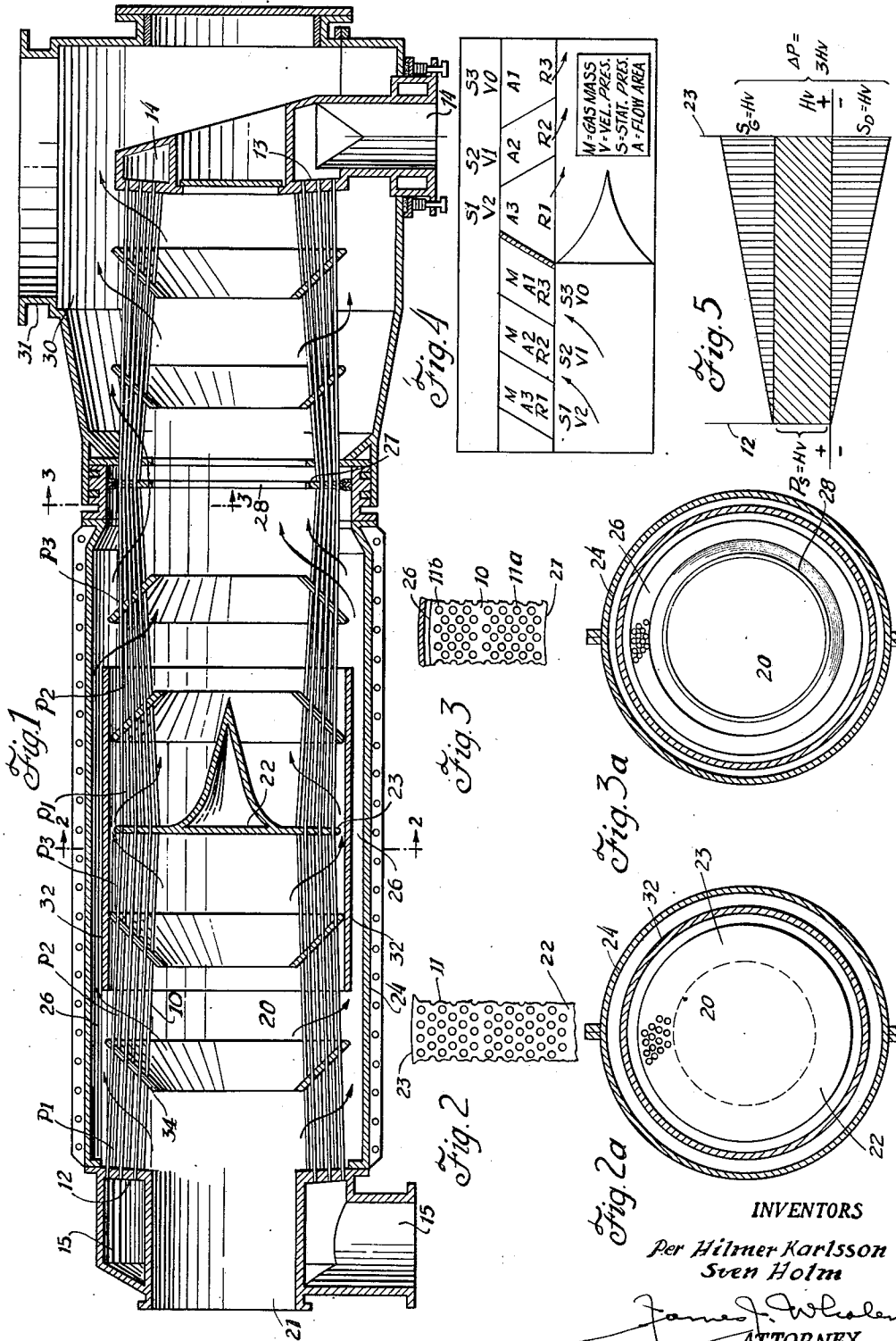
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TUBULAR HEAT EXCHANGER

2,606,006

Filed April 3, 1946

2 SHEETS—SHEET 1



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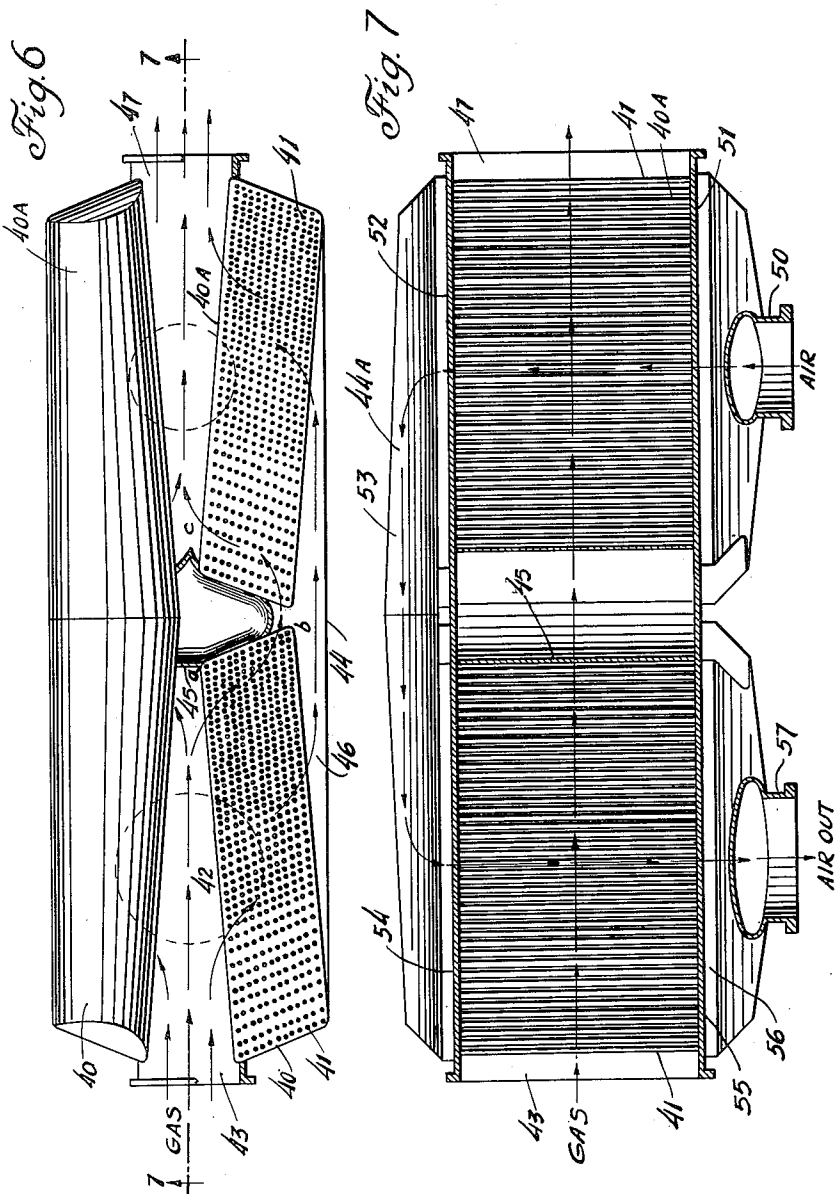
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TUBULAR HEAT EXCHANGER

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2 SHEETS—SHEET 2



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TUBULAR HEAT EXCHANGER

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3 Claims. (Cl. 257-235)

1

The present invention relates to heat exchange apparatus and more particularly to a tubular heat exchanger adapted, for example, for use in a gas turbine or other gas engine cycle to recover the heat in the gases exhausted from the turbine for preheating the working medium (air).

A feature of this invention is a heat exchange apparatus having a rate of heat transfer above what is ordinarily attained with heat recovery apparatus transmitting the heat from one gas to another.

An object of the invention is to attain a minimum flow resistance in the exchanger. The exhaust gases enter at a very high velocity and an object is to utilize the velocity of the gases to the best advantage in overcoming flow resistance.

Other objects and advantages of the invention will become apparent upon consideration of the following detailed description of illustrative embodiments of the invention when read in conjunction with the accompanying drawings in which:

Figure 1 is a longitudinal sectional view through a tubular heat exchanger embodying the invention;

Figures 2 and 3 are fragmentary sectional views on the correspondingly designated section lines in Figure 1 and illustrate two different tube spacing arrangements at points spaced longitudinally of the heat exchanger;

Figures 2a and 3a are diagrammatic views indicating the areas occupied by the tubes at the positions where the section lines for Figs. 2 and 3 are located;

Figure 4 is a diagram illustrating the relation between the areas for free flow of gases through the tube bundle and the pressure and velocity conditions existing at several locations longitudinally of the heat exchanger;

Figure 5 is a diagram showing the relation between the gas velocities and areas for gas flow in the first pass through the tube bundle.

Figure 6 is a view partially in plan and partially in diagrammatic section of another arrangement of a multi-pass heat exchanger embodying the invention wherein the heat transfer tubes are arranged in several banks and disposed normally to the direction of gas flow rather than extending in the general direction thereof; and

Figure 7 is a sectional view on the line 7-7 in Figure 6.

The heat exchanger illustrated in Figure 1 has an annular tube bundle 10 made up of a multiplicity of tubes 11 of a diameter of $\frac{1}{8}$ " or less

2

with thin walls and rolled at their ends into the tube sheets 12 and 13. The air to be heated is distributed to the tubes 11 through an inlet header 14 at the right and discharged from the exchanger through an outlet header 15. The tubes 11 instead of being dispersed over the entire area of the circular or rectangular tube sheets as in conventional tubular heat exchangers are disposed in a plurality of concentric rows to form the annular tube bundle 10 whose tubes surround the central space 20 within the heat exchanger. The heating gas enters through a duct 21 extending through the header 15 to the central space 20 within the bundle so that the gas flows longitudinally towards a baffle which is constituted by the central imperforate portion 22 of a tube spacing plate 23. Intermediate portions of the tubes 10 extend through plate 23 and a second plate 27 of annular form maintains the tubes in desired mutually spaced relations. The wrapper sheet or shell 24 surrounding the tube bundle 10 is spaced away from the latter between tube sheet 12 and the tube spacing plates 23, 27 so that the gases flowing from the central space 20 radially outward across portions of the tubes 10 in the first pass may flow longitudinally in a passage 26 formed between the shell 24 and the tube bundle, over the periphery of the baffle 23 and to the second pass. The spacing plate 27, being annular in form, provides a central aperture 28 through which the gases flow to enter the third pass wherein they flow radially outward across the right hand portions of the tubes 10 into a collecting chamber 30 from which they are discharged through an outlet 31.

The tubes 11 are connected into the tube sheet 12 at the gas inlet end of the exchanger in closely spaced relation in concentric circular rows and may be staggered in adjacent rows, if desired. Between the tube sheet 12 and the baffle plate 23, the axes of the tubes forming each of the concentric rows are parallel but become more closely spaced circumferentially due to the fact that the circles in which their holes are distributed in the plate 23 are made of smaller diameter than in the tube sheet 12. Consequently the intertube area free for gas flow is less between tube portions at baffle 23 than adjacent that part of the first pass contiguous to the inlet 21. In passing through the second spacing plate 27 the tubes project through openings which are arranged in circles of larger diameter than in the plate 23 but with the inner and outer circles 11a, 11b (Fig. 3) closer together

3

radially than at baffle 23 (Fig. 2) thereby still further decreasing the free area for gases to flow among the tubes as may be noted by examination of Figs. 2 and 3. At the tube sheet 13 the tubes may be located in concentric rows that are of smaller diameter than in the spacing plate 27, thus repeating the relationship that exists between the tube spacings at the inlet tube sheet 12 and the first baffle plate 23.

This variation in the spacings between adjacent tubes longitudinally thereof enables full use to be made of the velocity head of the gas entering inlet 21 (at 100 feet per second and over) and produces so-called stagnation point flow in the first pass of the unit. This type of flow is an almost perfect conversion from velocity head to static pressure head. Its simplest form is a jet of gas impinging on a flat plate whose surface is perpendicular to the axis of the jet. The velocity of the jet gradually decreases as it approaches the plate while at the same time the static pressure rises in the same proportion until the surface of the plate is reached, where the velocity is zero and the static pressure is a maximum. In the first pass the jet of gas entering inlet 21 at maximum velocity V_2 (Fig. 4) comes to zero velocity V_0 at the baffle 22 where the static pressure has increased from S_1 to S_3 and is a maximum. This increased static pressure is then used to create velocity through the tube bank for overcoming the greater resistance R^3 due to closer tube spacings at the inner end of the first pass. Thus, to obtain substantially uniform distribution of flow over the tube bank and avoid the tendency of gas to rush toward the inner end of the pass and flow over the tube portions near baffle 22 while largely bypassing the outer end portions of the tubes, the annular bundle 10 is tapered so as to give smaller gas flow area A_1 for flow between the tubes at the far end of the first pass where static pressure is at a maximum S_3 , and a larger area A_3 at the entrance end where static pressure is at a minimum S_1 .

Figure 5 shows the basis of calculations for proportioning the taper of the tube bank. For this particular design it has been assumed that the static pressure (P_s) at the entrance necessary for flow through the tube bank at a given velocity is equal to one velocity head (Hv). In the central space 20 within the tube bank the static pressure increases from the inlet end to the baffle 22 as indicated by Sg due to conversion from velocity pressure to static pressure. The static pressure existing in the outer passage 26 between the tube bundle 10 and casing 24 decreases as represented by Sd , this decrease resulting from the increase in velocity of the gases discharging over tube portions near baffle 22 into the space 26. This diagram graphically shows that, assuming 100% conversion from velocity head to pressure head, $P = Hv = Sg = Sd$ and therefore the pressure differential ($P + Sg + Sd$) at the inner end of the first pass or at the baffle 22 is three times that (P) at the outer end.

Figure 5 shows diagrammatically that at baffle 22 there is a drop in static pressure through the bank amounting to $3Hv$ while near inlet 21 the drop is Hv . Where V_1 and V_2 represent velocities through bundle 10 near inlet 21 and at baffle 22 the conventional gas flow formula

$$Hv = \frac{V^2}{2g}$$

where V is velocity and g is acceleration due to

4

gravity indicates the pressure differential at baffle 22 is

$$\frac{V_2^2}{2g} = 3 \times \frac{V_1^2}{2g} \text{ or } V_2^2 = 3 \times V_1^2 \text{ or } V_2 = V_1 \times \sqrt{3}$$

Since the flow through any section is equal to velocity times density times flow area, then for uniform mass rate of flow through the tube bank (if area for flow per unit length of tube bank is represented by A and D represents density of fluid) $V_1 \times D \times A_1 = V_2 \times D \times A_2$ it follows that for the assumed $Hv = P$ the area ratio

$$\frac{A_1}{A_2} \text{ should} = \sqrt{3}$$

to obtain a velocity ratio

$$\frac{V_2}{V_1} = \sqrt{3}$$

It is evident that if the resistance through the tube bank is less than Hv , the ratio

$$\frac{A_1}{A_2}$$

will be greater than $\sqrt{3}$ and smaller than $\sqrt{3}$ if the resistance through the tube bank is greater than Hv . It has been assumed herein that conversion from velocity head to static head and vice versa would be 100%. This is, of course, not possible with any kind of apparatus, due to friction etc. but 100% conversion has been used to simplify the explanation.

The ratio between areas for flow through the outer and inner ends of the first pass is therefore in reverse proportion

$$\left(\text{or } \frac{\sqrt{3}}{1} \right)$$

for uniform mass flow velocity rate for the whole length of pass, so that the area at the inner end is approximately 58% of that at the outer end. It is evident that a higher rate of heat transfer results from the higher velocity as compared with a tube bundle having tube rows that are parallel from end to end. A study of Figs. 4 and 5 makes it clear that this additional velocity through the bank and increased heat transfer is obtained by using the high velocity head of the entering gas. The same area ratio can be had in the second pass by narrowing the radial depth of the tube bank at the second baffle 27 where the static pressure is built up at the far end of the annular space 26. The same principle may be applied again in the third pass by reducing the diameter of the tube circles as was done at baffle 23.

At the baffle 23 the gas flows across the tube bank at substantially right angles and at very high velocity. To prevent this gas stream from bridging across the annular passage 26 and thus choking off the stream of gases from the inlet end of this passage a deflector 32 is provided. The frusto-conical deflectors 34 are for the purpose of directing parts of the gas stream along their outer faces to flow over the front part of the respective passes.

In the heat exchanger illustrated in Figs. 6 and 7 there are two banks 40 in each pass each comprising tubes 41 (represented by dots in Fig. 6) disposed so as to extend normally to the general direction of gas flow. These tubes are arranged in spaced relation in a plurality of parallel rows with the tube banks 40 disposed at opposite sides of the central space 42 through which the gas entering through inlet 43 flows toward the baffle

plate 45. Here the banks 40 are of rectangular form and in the first pass are arranged with their inner ends in convergent relation to each other and disposed between the inlet 43 and the baffle 45. In the second pass the banks 40A are arranged in divergent relation. These inclined relations provide the required passage 46 between the outer portion of each bank and the casing 44 through which gas may flow to pass over the tubes in the banks 40A in the second pass to be finally discharged through the outlet 47.

In order to obtain substantially uniform mass flow through the banks 40, the tubes in each longitudinal row of each bank are more closely spaced adjacent the baffle 45 than at the portion of the bank adjacent the inlet 43. Likewise in the banks 40A in the second pass of the heat exchanger the tubes at the far end near the outlet 47 are more closely spaced than at the inner end of these banks near baffle 45. Thus, the same principles of conversion of velocity head into static pressure and effective utilization thereof to attain better heat transfer apply to this construction also.

In this form the air to be heated enters through a bottom inlet 50 at the right hand end of the apparatus and flows into the lower ends of the tubes of both banks 40A at this end of the apparatus, the tubes 41 being connected at their ends to tube sheets 51 and 52. The top part of the casing 44A is spaced above the tube sheet 52 to provide a passage 53 for gases to flow longitudinally to enter the upper ends of the tubes 41 in the tube banks 40. The lower ends of the tubes in the banks 40 are seated in the tube sheet 55 located in the chamber 56 into which the outlet duct 57 is connected. Thus air which is first heated in the banks 40A absorbs further heat in the tubes of the banks 40 before being discharged through the air outlet 57.

The arrangement of tubes shown in Figs. 6 and 7 has the advantage that the tube spacing can be varied within a greater range than is possible with tubes continuous through all passes. For instance, due to stagnation point flow, and velocity head conversions, there is always a higher pressure differential between *a* and *b* (Fig. 6) than between *b* and *c*. Therefore, the resistance to flow should be higher between *a* and *b* than between *b* and *c* and the tube spacing closer where the higher resistance is required and good distribution can therefore be obtained. This is not possible with continuous tubes because next to the baffle the tube spacing must remain the same from one pass to the other. Another advantage with this arrangement of tubes is that the tube rows can be arranged in any desired angle in relation to the direction of flow of the gas entering the unit so that the gas stream passing through tube bank at the gas inlet end will contact all the tubes without dead corners and ineffective heating surface.

This application is a continuation-in-part of that filed in our name on November 17, 1944, under Serial No. 563,816, now abandoned.

What we claim is:

1. In a heat exchanger; an elongated casing formed in one end wall with an inlet opening directly into a narrow gas passage aligned therewith and extending longitudinally of said casing and which is several times as long as it is wide; a baffle extending transversely within said casing perpendicular to the axis of said passage with the periphery of the baffle spaced from the adjacent wall of said casing; and tube banks at opposite sides of the axis of said passage comprising tubes normal to the axis of said passage disposed in alinement in spaced parallel rows that extend longitudinally of said passage from said inlet to said baffle and inclined at an angle with respect to its longitudinal axis with the banks being located further from said axis at the inlet end of said passage and the tubes in the end portions of said rows adjacent said baffle being spaced closer together in said rows than tubes adjacent said inlet.

2. In a heat exchanger; an elongated casing formed in opposite end walls with an inlet and an outlet communicating directly with a narrow gas passage aligned therewith and extending longitudinally of said casing and which is several times as long as it is wide; a baffle extending transversely within said casing intermediate said inlet and outlet with the periphery of the baffle spaced from the enclosing casing to permit gas flow between portions of the passage located along its said longitudinal axis at either side of said baffle; and pairs of tube banks bounding opposite sides of said passage at each side of said baffle comprising tubes normal to the axis of said passage disposed in alinement in spaced parallel rows that extend longitudinally of said passage at angles inclined toward the baffle with respect to the longitudinal axis of said passage with the banks being located closer to said axis adjacent said baffle and the tubes in the end portions of said rows adjacent said baffle being spaced closer together than tubes adjacent said inlet.

3. A heat exchanger as recited in claim 2 including inlet and outlet headers for another fluid located at one side of said casing parallel to the longitudinal axis of said passage and in fluid communication with the tubes in said tube banks.

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REFERENCES CITED

The following references are of record in the file of this patent:

UNITED STATES PATENTS

Number	Name	Date
898,932	Schmug	Sept. 15, 1908
1,740,318	Smith	Dec. 17, 1929
2,149,007	Cassidy	Feb. 28, 1939
2,232,935	Bailey	Feb. 25, 1941
2,519,084	Tull	Aug. 15, 1950