Dec. 27, 1966 H. J. WOOD 3,294,161

HEAT EXCHANGERS

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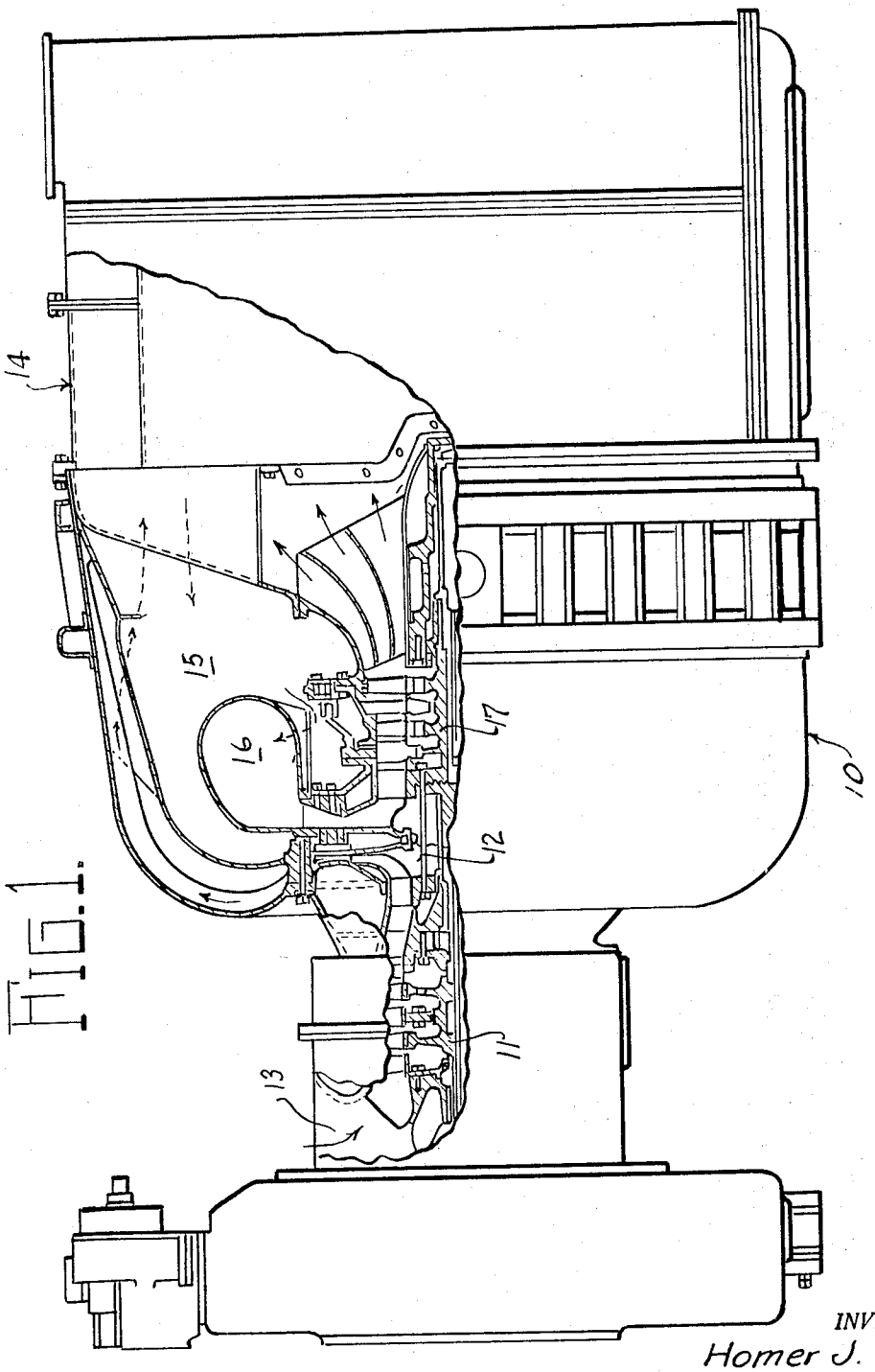


FIG. 1.

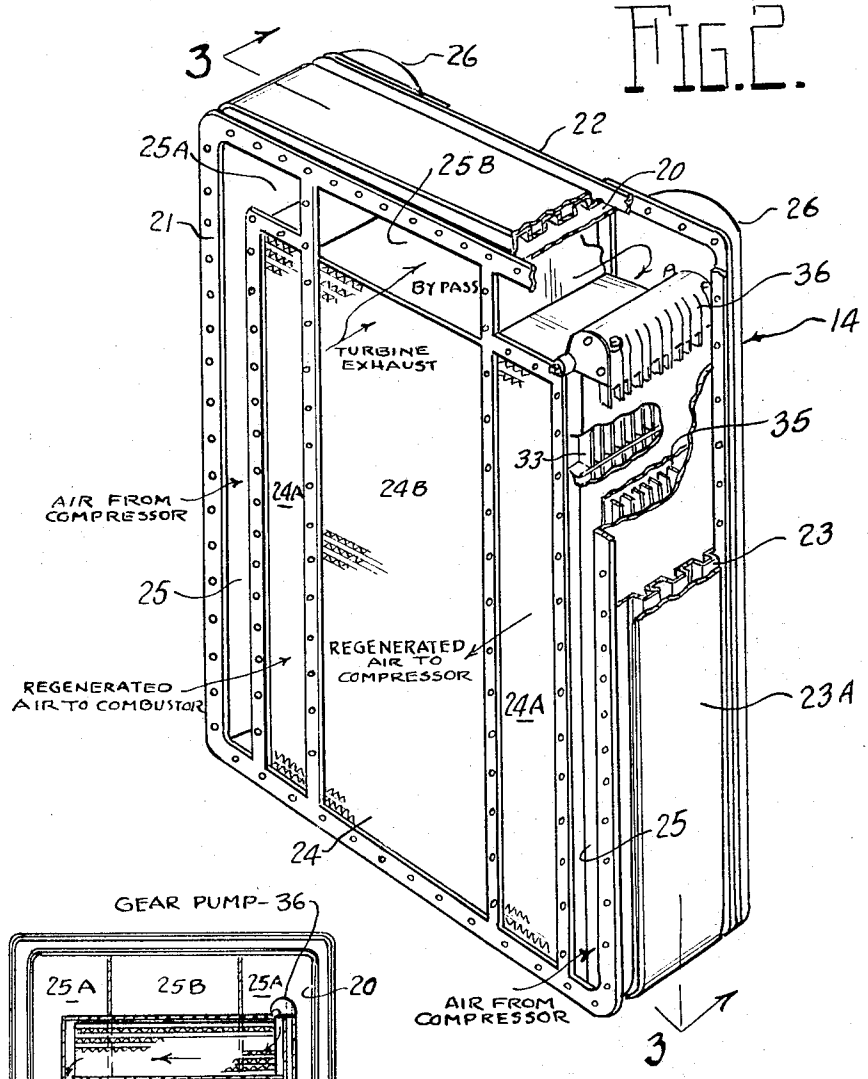
INVENTOR.
Homer J. Wood
BY
Hauke & Hauke
ATTORNEYS

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FIG. 2.



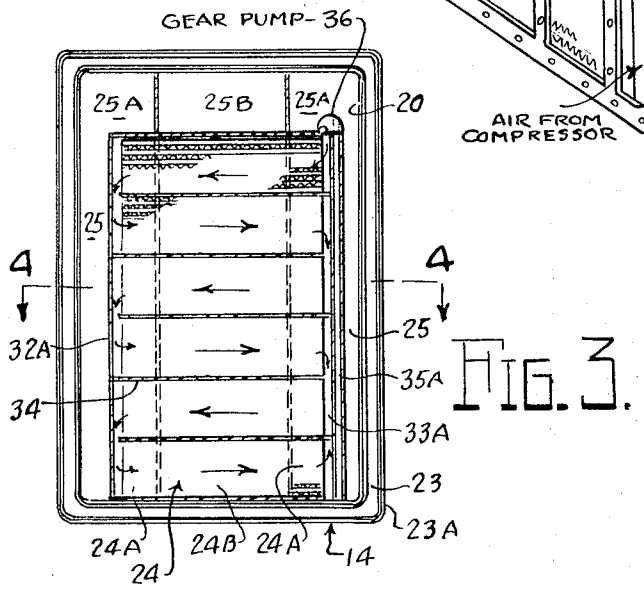


FIG. 3.

INVENTOR.
Homer J. Wood
BY Hauke & Hauke
ATTORNEYS

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FIG. 6.

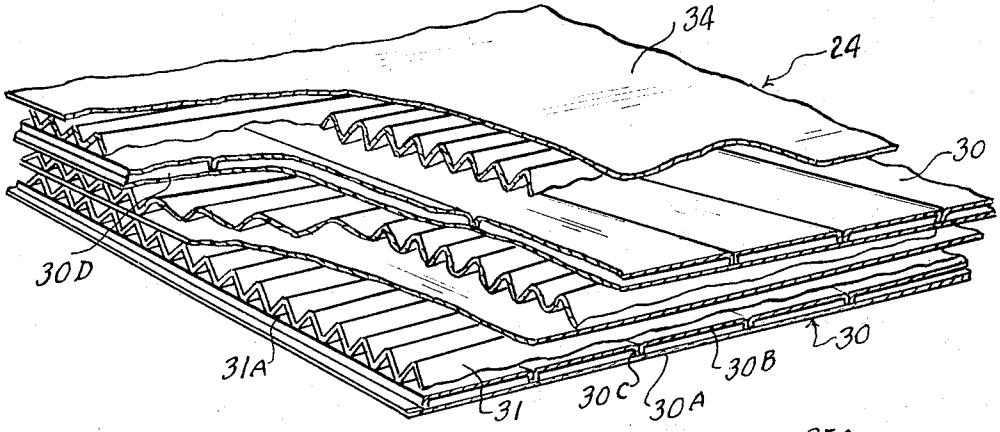
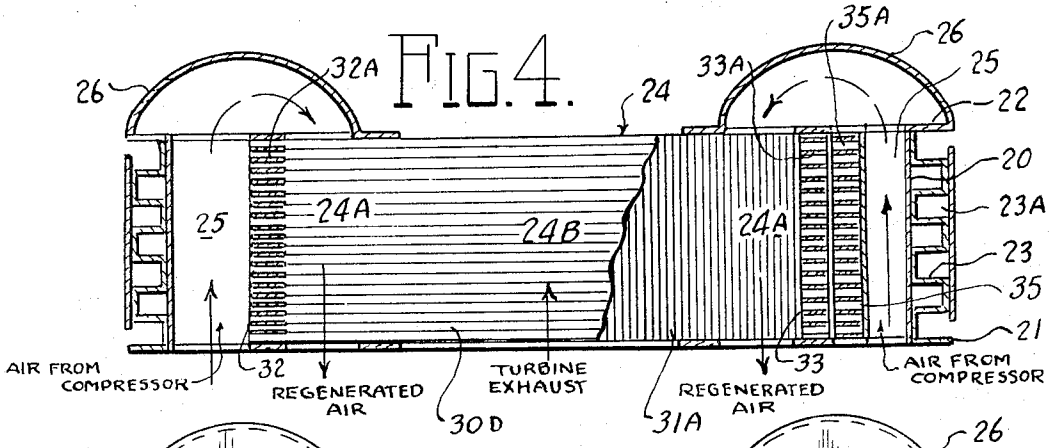


FIG. 4.



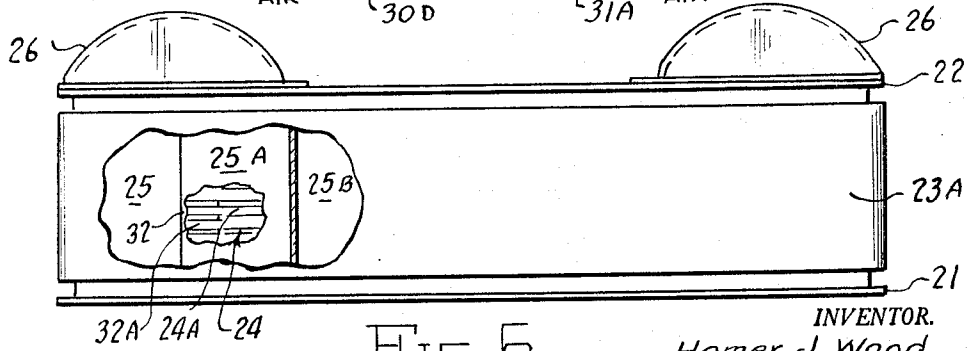


FIG. 5.

INVENTOR.
Homer J. Wood
BY Hauke & Hauke
ATTORNEYS

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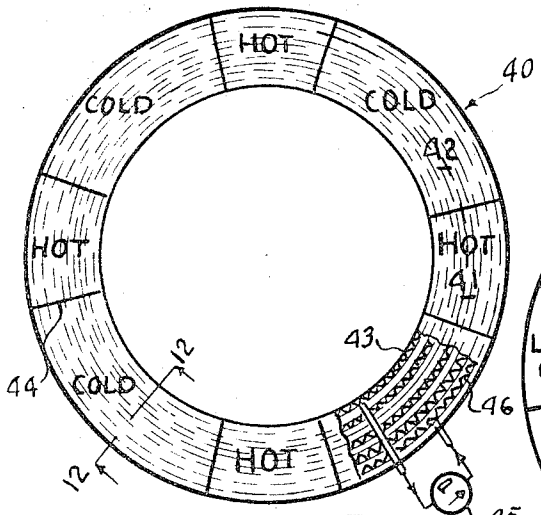


FIG. 7.

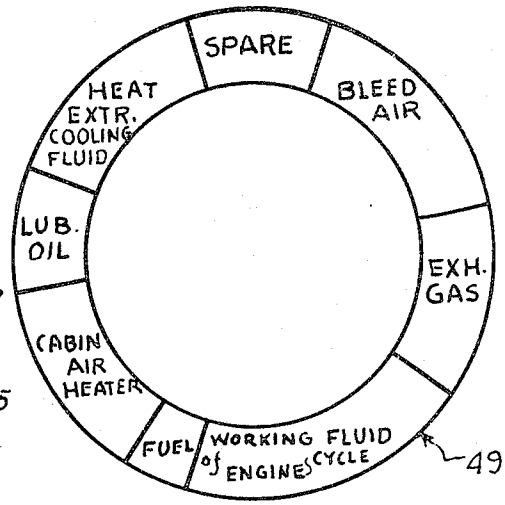


FIG. 8.

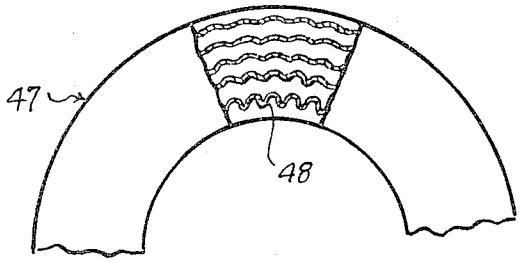


FIG. 9.

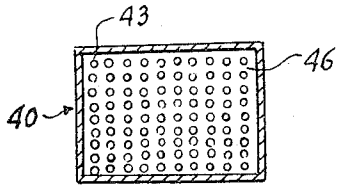


FIG. 12.

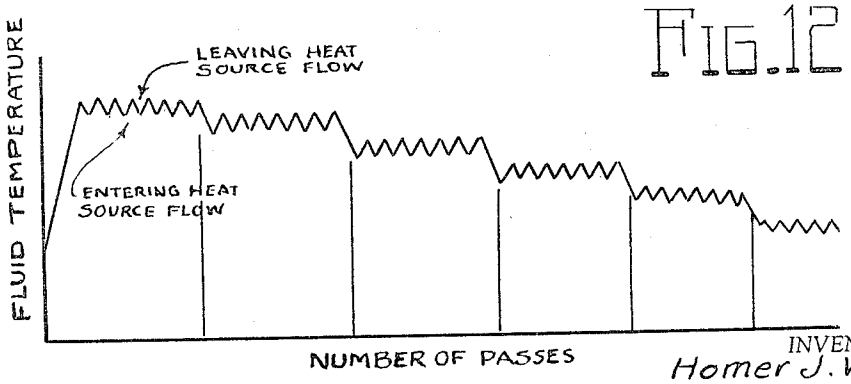


FIG. 11.

INVENTOR.
Homer J. Wood

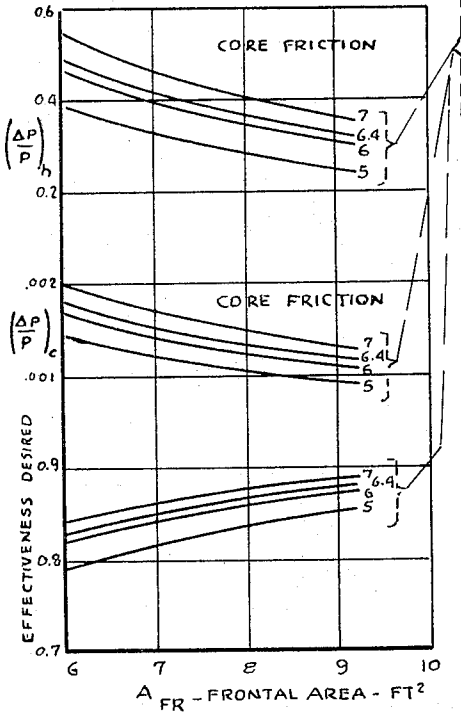
BY Hauke & Hauke

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HEAT EXCHANGERS

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FIG. 13.



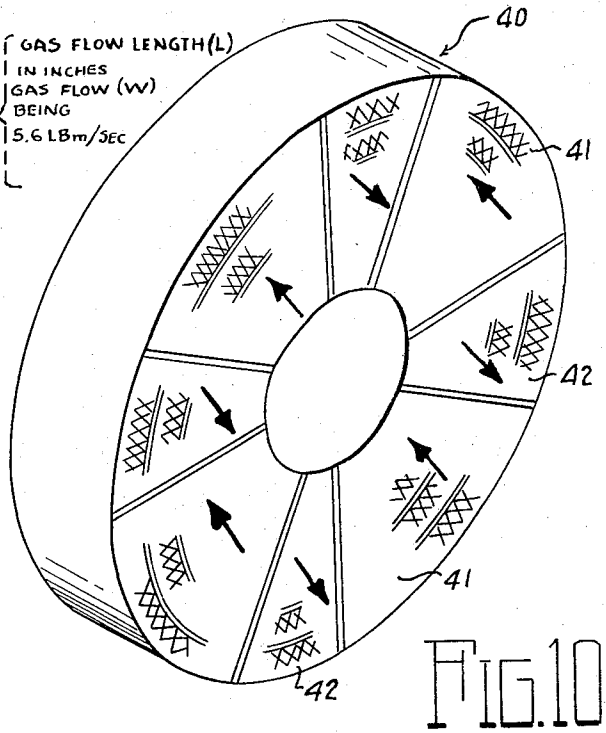


FIG. 10

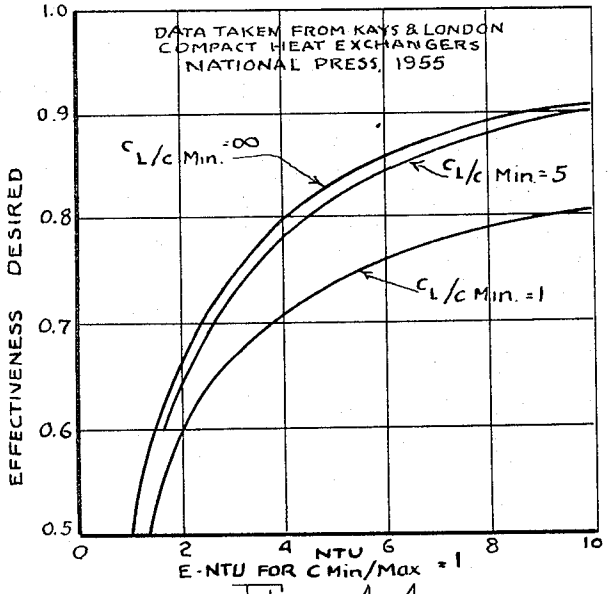


FIG. 14.

INVENTOR.
Homer J. Wood

BY Hauke & Hauke

ATTORNEYS

United States Patent Office

3,294,161

Patented Dec. 27, 1966

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HEAT EXCHANGERS

Homer J. Wood, Sherman Oaks, Calif., assignor to Continental Aviation and Engineering Corporation, Detroit, Mich., a corporation of Virginia

Filed July 3, 1961, Ser. No. 121,748
6 Claims. (Cl. 165—140)

My invention relates to heat exchangers and more particularly to a stationary type heat exchanger in which a heat transfer fluid serves to transport heat from a hot fluid stream to an adjacent cold fluid stream preferably flowing counter to the hot fluid stream.

The present heat exchanger was developed from a concept intended for application as a gas turbine regenerator, and the present description will so refer to such an application, although it will be seen that many other uses for it may be found.

Gas turbine regenerator construction must take into consideration the envelope and flow arrangement of the power plant and in addition must allow for loss of performance of the complete package due to any leakage between the high-pressure side and the low-pressure side of the heat exchanger.

Engine profile is also important in many applications, notably automotive, where the permissible envelope is dictated by considerations beyond the control of the power plant designer.

For a given heat transfer problem, with fluid flow rates and effectiveness fixed, the smallest possible heat exchanger will be of the counterflow configuration. Therefore, efforts to reduce weight and bulk logically start from the counterflow rather than the crossflow arrangement. However, ordinary direct counterflow heat exchangers of the extended surface or compact type required for a regenerator, present extremely complex ducting problems.

Other efforts to reduce the size and weight of regenerators have resulted in the evolution of the rotary counterflow regenerator. This type of regenerator is inherently smaller than a crossflow static regenerator of the same effectiveness due to thermodynamic considerations. Furthermore, in most cases, the rotary regenerator results in simpler ducting for the hot and cold fluids; hence a more desirable profile or frontal area.

The main disadvantages of the rotary regenerator are seal leakage and carry-over, both of which result in leakage from the high to low pressure sides of the unit with resulting loss of engine performance.

An object of the present invention is to simplify and improve heat exchangers by providing a construction which combines the best features of both the rotary counterflow and static crossflow concepts.

Another object of the invention is to improve heat exchangers by eliminating the seal leakage and carry-over problems inherent in rotary regenerators as well as the bulky ducting heretofore required in the static type regenerators.

A further object of the invention is to reduce weight and bulk of static heat exchangers by constructing same using principles of rotary counterflow regeneration.

Yet another object of the invention is to construct a new type of heat exchanger by providing a composite of several liquid metal to gas heat exchangers with the liquid metal flowing in cross-flow to the gas and in which the hot gas flows in a direction counter to that of the cold gas, the hot and cold gases flowing through adjacent areas.

For a more complete understanding of the invention, reference may be had to the accompanying drawings illustrating several embodiments of the invention in which

like reference characters refer to like parts throughout the several views and in which

FIG. 1 is a partially sectional elevational view of a preferred gas turbine engine incorporating a regenerator system embodying the present invention.

FIG. 2 is a perspective view, partially broken away, of a heat exchanger adapted for connection with a gas turbine and embodying the present invention.

FIG. 3 is a vertical transverse cross-sectional view of the structure of FIG. 2, as taken substantially on the line 3—3 of FIG. 2.

FIG. 4 is a cross-sectional view taken substantially on the line 4—4 of FIG. 3.

FIG. 5 is a top view of the heat exchanger of FIGS. 2 and 3.

FIG. 6 is a fragmentary perspective view of a core section used in the heat exchanger.

FIG. 7 is a diagrammatic view of another embodiment of the invention.

FIG. 8 is a diagrammatic view of a further embodiment of the invention.

FIG. 9 is a fragmentary diagrammatic view of yet another embodiment of the invention.

FIG. 10 is a perspective diagrammatic view of the heat exchanger illustrated in FIG. 7.

FIG. 11 is a diagram illustrating the temperature excursion of heat transporting fluid passing through source and sink and successive planes of tubes of the present regenerator from heat source to cold end.

FIG. 12 is a cross-sectional view taken substantially on the line 12—12 of FIG. 7.

FIG. 13 is a diagram illustrating the results of theoretical performance analysis of the present heat exchanger.

FIG. 14 is a data chart used in the analysis of the heat exchanger.

In FIG. 1, a preferred gas turbine engine **10**, to which the present heat exchanger may be adapted, comprises an axial compressor structure and a radial compressor structure **12** which are operable to compress air entering the air inlet **13**.

The compressed air is first conducted to a regenerator **14** where it becomes heated, then returned to a chamber **15** as indicated by flow arrows, and then directed into a combustor **16**, and is mixed with fuel. The mixture is burned, and the hot gases are expelled to operate a turbine **17** and then exhausted through the regenerator **14**.

The regenerator operates to transfer heat from the exhaust to the compressed air. Actual construction of the gas turbine engine is immaterial to the present invention which is directed to the regenerator itself. It will be understood that although the regenerator to be described was developed with its application to gas turbines in mind, the principles and construction to be described may have other applications.

The preferred regnerator **14** as illustrated in FIGS. 2–6 comprises a peripheral frame structure **20** having forward and rear flanges **21** and **22** adapted for assembly in a gas turbine engine, and a reinforcing channel frame **23** with a peripheral casing **23A**.

Within the frame structure **20**, and inwardly spaced from three sides thereof, is a core assembly **24**. Spaces **25** between opposite sides of the core assembly **24** and the sides of the frame structure and extending over portions of the core assembly as at **25A** are adapted for connection with ducts carrying the air from the compressor. Space **25B** over the core structure **24** may be adapted for connection with a turbine exhaust bypass.

Shroud structures **26** carried on the rear side of the frame structure **20** direct the compressor air around and

forward through spaced side potions 24A of the core structure 24, these portions 24A being adapted for connection to ducts carrying regenerated air back to the engine chamber 15. The exhaust from the gas turbine 10 normally flows through a center portion 24B of the core structure 24 and in a direction counter to the direction of air flow through the core portions 24A.

The core structure 24, as illustrated in FIG. 6, comprises a plurality of parallel levels of flattened conduit structures 30 each having a lower plate 30A and an upper plate 30B flanged as at 30C to divide the structure into a plurality of flat broad passages 30D.

The levels of conduit structures 30 are separated by a triangularly corrugated sheet 31 providing, in effect, finned passages 31A extending normal to the passages 30D. It will be seen that the result of using the present triangularly finned structure is to provide a central multi-passage hot gas duct intermediate and adjacent multi-passage cold gas ducts, the ducts having common separating side walls.

A return header 35 is provided adjacent the header 33 and is divided into a plurality of vertical passages 35A. A multisection fluid pump 36 is provided to pump the heat transfer fluid from the upper end of individual header passages 35A into the upper ends of the header passages 33A, from which the fluid flows into the uppermost levels of the conduit structures 30 for circulation back and forth and from top to bottom.

The heat exchange fluid used in the above-described regenerator is preferably a liquid metal such as bismuth-lead tin, sodium potassium or mercury. Sodium potassium is good because of its low melting point (12° F.), low weight and efficient heat transport properties. Alkali metals are generally avoided in gas turbine engine applications because of their violent oxidation reaction in the event of a leak.

The operation of the heat exchanger 14 will be apparent. Heat from the turbine exhaust flowing through the passages 31A in the center core section 24B is absorbed by the cross-flowing liquid metal in the passages 30D and transported to the side core sections 24A where the heat is given up to the air flowing counter to the exhaust flow.

The liquid metal flow is confined by the conduits 30D being individually headered so that stratification of temperature is maintained between the forward and rear faces of the core structure 34 to preserve the counter-flow effect. The liquid metal pumping rates are not critical as long as they are above minimum values and regulation of metal pumping is not required. In the present regenerator 14, negligible horsepower is required to drive the pump 36, compared with turbine power output.

Thermal behavior of the regenerator 14 is quite similar to that of a rotary regenerator, where the liquid metal flow is analogous to a moving heat transfer matrix. However, there are no carry-over and leakage losses as in rotary regenerators, and the only inhibition on metal flow rates is pumping power. Moreover, comparisons made show that for equal effectiveness, pressure loss and hydraulic radii (on gas and air passages), the above described regenerator has about 30 percent less bulk than a rotary regenerator and equal weight. Also, it will be seen that the present device has simple basic construction and is easy to clean because of the straight-through pasages.

It is also of great importance to note that the present use of transverse metal flow avoids the problem, encountered in those heat exchangers in which gas and metal move in counterflow, of the requirement that metal flow rates be controlled and maintained proportional to gas flow rates. In the present concept, such control is not necessary.

FIG. 10 is a perspective view of another heat exchanger 40 also adapted to a gas turbine, embodying the invention but constructed somewhat differently. As in the heat exchanger 14, it is essentially a composite of several liquid metal to gas heat exchangers with the liquid metal flowing in cross-flow to the gas and the hot gas from the turbine exhaust flowing through sections 41 counter to the flow of cold gas from the compressor through intermediate sections 42.

In this form, the heat exchanger bears a closer resemblance to a rotary regenerator, the essential difference being that heat is transported from the hot sections 41 to the cold sections 42 by circumferential motion of fluid within tubes rather than by physical rotation of the heat transfer elements which are necessary features of the conventional rotary regenerator.

FIG. 7 illustrates diagrammatically one possible arrangement in which a tube 43 is indicated as a single continuous tube in one plane or disc connected with a pump 45. The hot and cold gases flow axially through preferably triangularly shaped passages 46. A plurality of adjacent parallel tubes 43 would be provided as shown in FIG. 12 to occupy the desired axial length, so that operational temperature differences between adjacent circumferential segments in each plane will be small although temperature differences in the axial direction of the regenerator will have the conventional differences associated with heat transfer principles. The temperature excursion is illustrated in FIG. 11, in which the rise and fall of temperature within each "pass" or circuit of fluid flow in each plane will be relatively small, and mean temperature of the several passes will have the differences indicated, the pass at the hot gas inlet side being to the left in the chart. This temperature excursion will be generally the same regardless of the arrangement of heat transport fluid ducting and headers used.

In accomplishing heat transfer in and out of each tube 43, the fluid flow rate is maintained at a circumferential velocity generally comparable to the matrix motion in a rotary regenerator. In doing this, it is apparent that a "slug" of fluid will be heated by the hot gases in the time interval required to pass a distance equal to the pitch of radial separators 44. In passing into the cold section 42, this heat will be transferred to the cold air stream in accordance with conventional heat transfer principles. The cycle is then repeated continuously.

When there is an appreciable change in radius between the inner and outer coils of tubes 43, it will be seen that a preferred mean cycle may be arranged for the mean diameter tube cycle. The fluid transport rotational speed will vary, with slowest angular rotation occuring in the larger outer diameter length of tube and fastest rotation occurring in the smaller inner diameter length of tube. In this sense, the dwell time for the present heat exchanger will differ from the dwell time in a rotary regenerator. Any other tube may of course be selected for the optimum dwell speed. The use of low conductivity thin wall tubing such as stainless steel will reduce circumferential heat transfer to a minimum. There is no apparent great disadvantage in varying the dwell time over a limited range of variation from the mean dwell time. Some increases in size of the heat exchanger will result from deviation from an optimum dwell time, but this is compensated for by the simplicity of the system.

FIG. 9 illustrates diagrammatically a fragment of a heat exchanger 47 in which the heat transfer fluid tubes 48 are indicated as being contoured to permit equal passage time through all respectively radially spaced portions of the tubing, eliminating variations in fluid transport rotational speed and thus giving more uniform heat transfer efficiency over the radial dimension.

FIG. 8 illustrates diagrammatically a consolidated heat exchanger 49 having a plurality of sections adapted for connection with several consecutive heat transfer services as may be needed in a gas turbine engine. Heat transfer fluid tubes have been omitted for clarity, but it will be apparent that they will be generally similar to those indicated in FIG. 7.

It will be seen from the foregoing descriptions that the present heat exchanger accomplishes a close union of the

essential elements of the typical fluid transport regenerator in which the hot gas-to-liquid section heretofore has been remote from and connected by lengths of piping from the hot fluid to the cold air regenerator. In one sense, the present heat exchanger might be considered a liquid transport arrangement in which the connecting pipes have substantially zero length since they extend directly from hot to cold sections. It may also be thought of as consisting of series segments of a multi-pass cross flow heat exchanger in which the interconnecting header sections have been eliminated, the "headers" of the structure in FIGS. 2–6 being, in effect, additional individual connecting passages disposed in the cold sections.

Thus the present device has many advantages when compared to conventional heat exchangers. It is a structurally sound element that is readily adapted to engine structure. It eliminates the disadvantages of the rotary regenerator, viz: it needs no mechanical drive, hence no bearing and gears; it eliminates leakage between gas passages and avoids troublesome seals; it has no carry-over of the gases from one passage to the other and can thus be used with radically different gases. It is not limited to gas-to-gas heat exchange but can be readily adapted for liquid to gas, gas to liquid, and liquid to liquid heat exchange, or a combination of these may be handled by one assembly. It also eliminates the bulky and/or complex ducting heretofore required by static type counter-flow heat exchangers. Moreover, the elimination of liquid transfer ducts as are used where the hot and cold systems are separated avoids the problem of cold weather operation in which freezing can occur in the transfer ducts. The present concept envisions surrounding all of the liquid metal passages with heated air which is more than adequate on cold starting to thaw the liquid metal.

In all of the embodiments of the invention presented herein, it will be seen that the device may be compared to a rotary regenerator with the extended surface of matrix on the gas side stationary and liquid metal flowing in cross-flow to the gas taking the place of the rotating mass.

In order to determine the number of transfer units (TU) or core sections required in the present device, it is necessary to first establish the temperature distribution in the fluid streams as they proceed through the heat exchanger. It is apparent that the effectiveness—NTU relationships which have been established for rotary regenerators implicitly include consideration of the temperature distribution.

Consider the heat exchanger divided into N modules, each consisting of a hot-gas to liquid-metal and a cold-gas to liquid-metal section. Since the entire flow of liquid metal passes first through the hot section and then through the cold section, and $1/N$ times the total gas flow passes through each section of the module, the capacity rate ratio for the module will be NC_L/C_{min}, where C_L is the liquid capacity rate and C_{min} is the gas capacity rate (that is, we may increase the effective capacity rate ratio between the liquid metal side and the gas side by increasing the number of modules). In the limit, as N approaches infinity, the weight of liquid metal flow required to achieve a finite capacity rate ratio tends to zero. Also, as N approaches infinity, there is a hot air pass of infinitesimal width, on each side of a cold air pass. But, this is simply a direct gas-to-gas heat exchanger of the counterflow type. Therefore, we may conclude that if N and/or C_L is much greater than the capacity rate (C_{min}) of the gas flow, the performance of the heat exchanger will be that of a counterflow heat exchanger.

Using a capacity rate ratio of five between the liquid metal and the gas for each module and C_c/C_h equal to 1.0 where C_c is the capacity rate of cold gas and C_h is the capacity rate of hot gas, measured in B.t.u./sec.°R., it may be seen from FIG. 14, that the NTU required for an effectiveness of 0.80 is 4.25. It is interesting to note that a cross-flow heat exchanger of C_c/C_h equal to 1.0 with an NTU of 4.25 will have an effectiveness of 0.75; and that a cross-flow heat exchanger with an effectiveness of 0.8 would require an NTU of approximately eight (8). Thus, it is apparent that in order to minimize the heat exchanger size (of which NTU is a measure for a given gas flow rate) it is highly desirable to achieve a counterflow arrangement.

FIG. 13 presents the results of performance analysis in the form of effectiveness $(\Delta P/P)_c$ or cold gas pressure drop and $(\Delta P/P)h$ or hot gas pressure drop being plotted versus frontal area, A_{fr}. It will be noted (lowermost set of curves) that for a gas flow W of 5.6 pounds per second, it is possible to achieve an effectiveness of 0.8 with a frontal area of 6.5 square feet and an air flow length of five inches. The cold side pressure drop factor at this point would be 0.0013 and the hot side pressure drop factor would be 0.0355. The total $\Delta P/P$ would be 0.368. It should be noted that this pressure drop is core friction only, and no effort has been made to account for the entrance or exit pressure losses or flow acceleration. The hot side pressure drop (which is more than ten times the cold side pressure drop) will not be adversely affected by flow acceleration, since the gas is being cooled and hence any acceleration term would tend to reduce the calculated pressure drop. A final design solution of the problem would of course include calculation of these pressure drop terms.

Depending upon the type of liquid metal pump available, one may connect the liquid metal passages in series or parallel since either scheme will result in the same flow rate of metal passing a given cross-section of the heat exchanger. If a high head, low volume flow pump is more feasible than a low head, high flow pump, it will probably be desirable to connect the passages in series.

Although the required power is very low, every effort has been made to be conservative in this calculation. One explanation for the low power required is that with N=4, which is the specific case calculated, there is a relatively low value for C_{min}/N and the liquid metal flow rate required to achieve NC_L/C_{min} equal to 5 is therefore also low.

The heat exchanger weight has also been calculated theoretically. Using 0.004 thick stainless steel, the core density is 98 lbm./ft.^{3}. There will be an additional 6.35 lbm./ft.^{3} of liquid metal if Nak (56 percent Na 44 percent K) at 1200° R. is used. The total density is thus 104.85 lbm./ft.^{3}. A unit with a frontal area A_{fr} of 6.5 ft.^{2} and length L of 5.0 inch would weigh 281 lbm. This is 0.468 lbm./H.P. weight penalty for the regenerator.

Although it may be possible to use fin and plate material thinner than 0.004 inch, a conservative approach has been taken. The use of triangular fin results in the least weight for a given fin thickness and unit performance.

The performance shown in FIG. 13 is for a unit with mild steel fins. Mild steel has a conductivity of 23 B.t.u./hr.° R. ft. compared to 32 B.t.u./hr.° F. ft. for nickel.

The foregoing analysis is based on the assumption of a capacity rate ratio between the hot and cold gases of nearly 1.0. This assumption is valid since the air fuel ratio in the particular gas turbine application considered is approximately 840:1.

Although only a few embodiments of the invention are described, it will be apparent to one skilled in the art to which the invention pertains that various changes and modifications may be made therein without departing from the spirit of the invention or the scope of the appended claims.

I claim:

1. A heat exchanger comprising

(a) a core structure having a plurality of closely spaced parallel passages extending from one face to an opposite face of said structure,

(b) a hot fluid duct and a cold fluid duct connected with adjacent areas of said core structure and adapted to provide respectively hot and cold fluid flow through adjacent groups of said passages,

(c) means transferring heat from the areas of said core structure connected with said hot fluid duct to the areas of said core structure connected with said cold fluid duct and comprising a plurality of fluid conduits extending through said core structure areas normal to said passages and in heat exchange relation therewith, and means circulating a heat transporting fluid through said fluid conduits simultaneously with the flow of fluid through said hot and cold fluid ducts.

2. The heat exchanger as defined in claim 1 and in which said fluid conduits are arranged in a plurality of spaced transverse levels and are connected to headers at opposite ends, said headers being arranged to direct fluid through a group of said conduit levels in one direction and subsequently through a group of conduit levels in the opposite direction.

3. The heat exchanger as defined in claim 1 and in which said fluid conduits are arranged in a plurality of spaced transverse levels and are connected to headers at opposite ends, said headers being arranged to direct fluid in repeated passes through said conduit levels, said levels being arranged in groups and the fluid passing through adjacent groups being directed in opposite directions.

4. The heat exchanger as defined in claim 1 and in which said fluid conduits are arranged in a plurality of spaced transverse levels and having a corrugated sheet disposed intermediate and spacing said conduit levels to provide adjacent substantially triangular passages between said levels.

5. In combination with the gas turbine engine having an air induction means and an exhaust gas conducting means, means for transferring the heat from the exhaust gases carried by said exhuast gas conducting means to the air conducted by said air induction means comprising

(a) a first heat exchanger and a second heat exchanger, said heat exchangers having a substantially common side wall,

(b) said first heat exchanger being connected with said air induction means,

(c) said second heat exchanger being connected with said exhaust gas conducting means,

(d) conduits passing through said side wall and said heat exchangers and means circulating a heat transporting fluid through said conduits in heat exchange relation with and in cross flow relation to both of the fluids conducted through said heat exchangers.

6. A heat exchanger comprising

(a) a hot fluid conducting duct and a cold fluid conducting duct;

(b) said ducts having a substantially common sidewall;

(c) means transferring heat from said hot fluid conducting duct to said cold fluid conducting duct and comprising fluid conduits passing through said side wall and said ducts and means circulating a heat transporting fluid through said conduits simultaneously and in heat exchange relation with and in cross flow relation to the fluids conducted through said ducts; and

(d) said heat exchanger having a plurality of said hot ducts and a plurality of cold ducts each arranged intermediate two of said hot ducts, said ducts being arranged parallel to each other to form a circumferential drum, and said conduits arranged in coils to extend through said ducts.

References Cited by the Examiner

UNITED STATES PATENTS

2,413,225	12/1946	Griffith	60—39.66
2,469,028	5/1949	Balaieff	165—140 X
2,650,073	8/1953	Holm	165—140
2,674,849	4/1954	Bowden	60—39.66
2,731,239	1/1956	Andersen	60—39.51
2,995,344	8/1961	Hryniszak	257—245
3,138,925	6/1964	Machery	60—39.18

FOREIGN PATENTS

668,493	12/1938	Germany.
1,088,027	9/1960	Germany.
734,938	8/1955	Great Britain.

ROBERT A. O'LEARY, *Primary Examiner.*

HERBERT L. MARTIN, CHARLES SUKALO, FREDERICK L. MATTESON, *Examiners.*

T. W. STREULE, *Assistant Examiner.*