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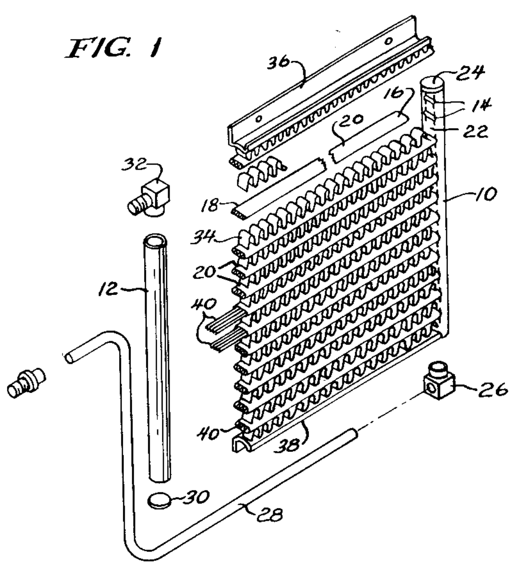
(54) **Heat exchanger.**

(57) A heat exchanger for use in a refrigeration or air conditioning system for cooling the refrigerant comprises a pair of spaced headers (10,12), and at least one heat exchange tube (20) extending between the headers defining a plurality of hydraulically parallel refrigerant flow paths, each of said flow paths having a hydraulic diameter of up to about 0.07 inches, the hydraulic diameter being defined as the cross-sectional area of the flow path multiplied by four and divided by the wetted perimeter of the flow path.

FIG. 1



EP 0 583 851 A2

This invention relates to a heat exchanger suitable for use in a refrigeration or air conditioning system for cooling the refrigerant. The invention is particularly applicable to a condenser for condensing a refrigerant using ambient air as a cooling medium.

Many condensers employed in air conditioning or refrigeration systems at the present time utilize one or more serpentine conduits on the vapour side. In order to prevent the existence of an overly high pressure differential from the vapour inlet to the outlet, which would necessarily increase system energy requirements, the flow passages within such tubes are of relatively large size to avoid high resistence to the flow of vapour and/or condensate.

This, in turn, means that the air side of the tubes will be relatively large in size. The relatively large size of the tubes on the air side results in a relatively large portion of the frontal area of the air side being blocked by the tube and less area available in which air side fins may be disposed to enhance heat transfer.

As a consequence, to maintain a desired rate of heat transfer the air side pressure drop will become undesirably large, and a commensurately undesirably large system energy requirement in moving the necessary volume of air through the air side of the condenser will result.

The present invention is directed to overcoming the above problems.

Summary of the Invention

It is the principal object of the invention to provide a new and improved condenser for use in air conditioning or refrigerant systems. More specifically, it is an object of the invention to provide such a condenser wherein the condenser has a lesser frontal area on the air side that is blocked by tubes allowing an increase in the air side heat exchange surface area without increasing air side pressure drop and without increasing vapor and/or condensate side pressure drop.

An exemplary embodiment of the invention achieves the foregoing objects in a condenser comprising a pair of spaced headers, one of the headers having a vapor inlet and the other of the headers having a condensate outlet. A condenser tube extends between the headers and is in fluid communication with each. The tube defines a plurality of hydraulically parallel substantially discrete fluid flow paths between the headers and each of the fluid flow paths has a hydraulic diameter in the range of about 0.015 to 0.040 inches.

In a preferred embodiment, there are a plurality of such tubes extending between the headers in hydraulic parallel with each other in sufficient number as to avoid high resistance to condensate and/or vapor flow.

The invention contemplates that the tubes be flattened tubes.

In a highly preferred embodiment, the invention contemplates that the plurality of flow paths in each tube be defined by an undulating spacer contained within the tubes.

Fins may be disposed on the exterior of the condenser tube and extend between the exteriors of adjacent ones of the condenser tubes.

The invention contemplates that the headers be defined by generally cylindrical tubes having facing openings, such as slots, for receiving respective ends of the condenser tubes.

Other objects and advantages will become apparent from the following specification taken in connection with the accompanying drawings.

Description of the Drawings

Fig. 1 is an exploded, perspective view of a condenser made according to the invention;

Fig. 2 is a fragmentary, enlarged, cross-sectional view of a condenser tube that may be employed in the invention;

Fig. 3 is a graph of the predicted performance of condensers with the same face area, some made in a prior art design and others made according to the invention, plotting heat transfer against cavity (hydraulic) diameter;

Fig. 4 is a graph comparing the present invention with the prior art construction showing air flow through each versus (a) the rate of heat transfer, (b) the refrigerant flow rate, and (c) the refrigerant pressure drop;

Fig. 5 is a further graph comparing the prior art construction with a condenser made according to the invention on the basis of air velocity versus the heat transfer per pound of material employed in making up the core of each; and

Fig. 6 is a further graph comparing the prior art construction with the present invention by plotting air velocity versus pressure drop across the air side of the condenser.

Description of the Preferred Embodiment

An exemplary embodiment of a condenser made according to the invention is illustrated in Fig. 1 and is seen to include opposed, spaced, generally parallel headers 10 and 12. According to the invention, the headers 10 and 12 are preferably made up from generally cylindrical tubing. On their facing sides, they are provided with a series of generally parallel slots or openings 14 for receipt of corresponding ends 16 and 18 of condenser tubes 20.

Preferably, between the slots 14, in the area shown at 22, each of the headers 10 and 12 is provided with a somewhat spherical dome to improve resistance to pressure as explained more fully in the commonly assigned, copending application of Saperstein et al, entitled "Heat Exchanger" U.S. application Ser. No. 722,653, filed April 12, 1985, the details of which are herein incorporated by reference.

The header 10 has one end closed by a cap 24 brazed or welded thereto. Brazed or welded to the opposite end is a fitting 26 to which a tube 28 may be connected.

The lower end of the header 12 is closed by a welded or brazed cap 30 similar to the cap 24 while its upper end is provided with a welded or brazed in place fitting 32. Depending upon the orientation of the condenser, one of the fittings 26 and 32 serves as a vapor inlet while the other serves as a condensate outlet. For the orientations shown in Fig. 1, the fitting 26 will serve as a condensate outlet.

A plurality of the tubes 20 extend between the headers 10 and 12 and are in fluid communication therewith. The tubes 20 are geometrically in parallel with each other and hydraulically in parallel as well. Disposed between adjacent ones of the tubes 20 are serpentine fins 34 although plate fins could be used if desired. Upper and lower channels 36 and 38 extend between and are bonded by any suitable means to the headers 10 and 12 to provide rigidity to the system.

As can be seen in Fig. 1, each of the tubes 20 is a flattened tube and within its interior includes an undulating spacer 40.

In cross section, the spacer 40 appears as shown in Fig. 2 and it will be seen that alternating crests are in contact along their entire length with the interior wall 42 or the tube 20 and bonded thereto by fillets 44 of solder or braze metal. As a consequence, a plurality of substantially discrete hydraulically parallel fluid flow paths 46, 48, 50, 52, 54, 56, 58 and 60 are provided within each of the tubes 20. That is to say, there is virtually no fluid communication from one of such flow paths to the adjacent flow paths on each side. This effectively means that each of the walls separating adjacent fluid flow paths 46, 48, 50, 52, 54, 56, 58 and 60 are bonded to both of sides of the flattened tube 20 along their entire length. As a consequence, there is no gap that would be filled by fluid with a lesser thermal conductivity. As a result, heat transfer from the fluid via the walls separating the various fluid flow paths identified previously to the exterior of the tube is maximized. In addition, it is believed that discrete flow paths of the size mentioned take advantage of desirable effects of heat transfer caused by surface tension phenomena.

A second advantage resides in the fact the condensers such as that of the present invention are employed on the outlet side of a compressor and therefore are subjected to extremely high pressure. Conventionally, this high pressure will be applied to the interior of the tubes 20. Where so-called "plate" fins are utilized in lieu of the serpentine fins 34 illustrated in the drawings, the same tend to confine the tubes 20 and support them against the internal pressure employed in a condenser application. Conversely, serpentine fins such as those shown at 34 are incapable of supporting the tubes 20 against substantial internal pressure. According to the invention, however, the desired support in a serpentine fin heat exchanger is accomplished by the fact that the spacer 40 and the crest thereof is bonded along its entire length in the interior wall 42 of each tube 20. This bond results in various parts of the spacer 40 being placed in tension when the tube 20 is pressurized to absorb the force resulting from internal pressure within the tube 20 tending to expand the tube 20.

One means by which the tubes 20 with accompanying inserts 40 may be formed is disclosed in the commonly assigned U.S. application of Saperstein, entitled "Tube and Spacer Construction For Use In Heat Exchangers", Serial No. 740,000, filed May 31, 1985, the details of which are herein incorporated by reference. A highly preferred means by which the tubes 20 with accompanying inserts 40 may be formed is disclosed in the commonly U.S. assigned application of Saperstein et al, entitled "Method of Making a Heat Exchanger", Serial No. 887,223, filed July 21, 1986, the details of which are also herein incorporated by reference.

According to the invention, each of the flow paths 48, 50, 52, 54, 56 and 58, and to the extent possible depending upon the shape of the insert 40, the flow paths 46 and 60 as well, have a hydraulic diameter in the range of about 0.015 to 0.040 inches. Given current assembly techniques known in the art, a hydraulic diameter of approximately 0.035 inches optimizes ultimate heat transfer efficiency and ease of construction. Hydraulic diameter is as conventionally defined, namely, the cross-sectional area of each of the flow paths multiplied by four and in turn divided by the wetted perimeter of the corresponding flow path.

The values of hydraulic diameter given are for condensers in R-12 systems. Somewhat different values might be expected in systems using a different refrigerant.

Within that range, it is desirable to make the tube dimension across the direction of air flow through the core as small as possible. This in turn will provide more frontal area in which fins, such as the fins 34, may be disposed in the core without adversely increasing air side pressure drop to obtain a better rate of heat transfer. In some instances, by minimizing tube width, one or more additional rows of the tubes can be included.

In this connection, the preferred embodiment contemplates that tubes with separate spacers such as illustrated in Fig. 2 be employed as opposed to extruded tubes having passages of the requisite hydraulic diameter. Current extrusion techniques that are economically feasible at the present for large scale manufacture of condensers generally result in a tube wall thickness that is greater than that that is required to support a given pressure using a tube and spacer as disclosed herein. As a consequence, the overall tube width of such extruded tubes is somewhat greater for a given hydraulic diameter than a tube and spacer combination, which is undesirable for the reasons stated immediately preceding. Nonetheless, the invention contemplates the use of extruded tubes having passages with a hydraulic diameter within the stated range.

It is also desirable that the ratio of the outside tube periphery to the wetted periphery within the tube be made as small as possible so long as the flow path does not become sufficiently small that the refrigerant cannot readily pass therethrough. This will lessen the resistance to heat transfer on the vapor and/or conduit side.

A number of advantages of the invention will be apparent from the data illustrated in Figs. 3-6 inclusive and from the following discussion. Fig. 3, for example, on the right-hand side, plots the heat transfer rate against the cavity or hydraulic diameter in inches at air flows varying from 450 to 3200 standard cubic feet per minute for production condenser cores made by the assignee of the instant application.

To the left of such data are computer generated curves based on a heat transfer model for a core made according to the present invention, the model constructed using empirically obtained data. Various points on the curves have been confirmed by actual tests. The curves designated "A" represent heat transfer at the stated air flows for a core such as shown in Fig. 1 having a frontal area of two square feet utilizing tubes approximately 24 inches long and having a 0.015 inch tube wall thickness, a 0.532 tube major dimension, 110 °F. inlet air, 180 °F. inlet temperature and 235 psig pressure for R-12 and assuming 2 °F. of subcooling of the exiting refrigerant after condensation. The core was provided with 18 fins per inch between tubes and the fins were 0.625 inches by 0.540 inches by 0.006 inches.

The curves designated "B" show the same relationship for an otherwise identical core but wherein the length of the flow path in each tube was doubled i.e., the number of tubes was halved and tube length was doubled. As can be appreciated from Fig. 3, heat transfer is advantageously and substantially increased in the range of hydraulic diameters of about 0.015 inches to about 0.040 inches through the use of the invention with some variance depending upon air flow.

Turning now to Figs. 4, actual test data for a core made according to the invention and having the dimensions stated in Table 1 below is compared against actual test data for a condenser core designated by the assignee of the present application as "1E2803". The data for the conventional core is likewise listed in Table 1 below.

Both the core made according to the invention and the conventional core have the same design point which is, as shown in Fig. 4, a heat transfer rate of 26,000 BTU per hour at an air flow of 1800 standard cubic feet per minute. The actual observed equivalence of the two cores occurred at 28,000 BTU per hour and 2,000 standard cubic feet per minute; and those parameters may be utilized for comparative purposes.

Viewing first the curves "D" and "E" for the prior art condenser and the subject invention respectively it will be appreciated that refrigerant flow for either is comparable over a wide range of air flow values. For this test, and those illustrated elsewhere in Figs. 4-6, R-12 was applied to the condenser inlet at 235 psig at 180 °F. The exiting refrigerant was subcooled 2 °F. Inlet air temperature to the condenser was 110 °F.

The greater refrigerant side pressure drop across a conventional core than that across a core made according to the invention suggests a greater expenditure of energy by the compressor in the conventional system than in the one made according to the subject invention as well.

Curves "F" and "G", again for the prior art condenser and the condenser of the subject invention, respectively, show comparable heat transfer rates over the same range of air flows.

Curves "H" and "J" respectively for the conventional condenser and the condenser of the subject invention illustrate a considerable difference in the pressure drop of the refrigerant across the condenser. This demonstrates one advantage of the invention. Because of the lesser pressure drop across the condenser when made according to the invention, the average temperature of the refrigerant, whether in vapor form or in the form of condensate will be higher than with the conventional condenser. As a consequence, for the same inlet air temperature, a greater temperature differential will exist which, according to Fourier's law, will enhance the rate of heat transfer.

There will also be a lesser air side pressure drop in a core made according to the invention than with the conventional core. This is due to two factors, namely, the lesser depth of the core and the greater free flow area not blocked by tubes; and such in turn will save on the fan energy required to direct the desired air flow rate through the core. Yet, as shown by the curves "F" and "G" the heat transfer rate remains essentially the same.

It has also been determined that a core made according to the invention, when compared with the conventional core, holds less refrigerant. Thus, the core of the invention reduces the system requirement for refrigerant. Similarly, there is lesser space required for installation of the inventive core because of its lesser depth.

As can be seen from the table, and in consideration with the data shown in Fig. 4, it will be appreciated that a core made according to the invention can be made of considerably lesser weight than a conventional core. Thus, Fig. 5 compares, at various air velocities, the heat transfer rate per pound of core of the conventional condenser (curve "K") versus heat transfer per pound of core of a condenser made according to the invention (curve "L"). Thus, Fig. 5 demonstrates a considerable weight savings in a system may be obtained without sacrificing heat transferability by using the core of the present invention.

TABLE 1
CONDENSER CORE PHYSICAL PROPERTIES
FOR FIGS. 3 AND 4

CORE PROPERTIES	CURRENT PRODUCTION 1E2803	PRESENT INVENTION
Depth (in.)	.938	.540
Heights (in.)	12.276	12.00
Length (in.)	24.13	23.259
Face Area ($ft.^2$)	2.057	1.938
Weight (lbs)	5.682	2.057
Ratio $\frac{\text{outside surface}}{\text{inside surface}}$	4.478	5.391
FIN PROPERTIES		
FPI	12	18
Fin Rows	13	21
Fin Thickness (in.)	.008	.004
Fin Height (in.)	.7502	.5018
Free Flow Area (ft^2)	1.444	1.554
Surface Area (ft^2)	37.110	33.389
Hydraulic Diameter (in.)	.1304	.0910
Fin Weight (lbs.)	2.163	.993
TUBE PROPERTIES		
No. Circuits	2	20
Tube Rows	14	20
Tube Thickness (in.)	.187	.075
Tube Wall (in.)	.027	.015
Tube Length (ft.)	15.168	2.047
Free Flow Area ($in.^2$)	.1556	.3200
Hydraulic Diameter (in.)	.07871	.0302
Outside Tube Surface (ft^2)	4.431	3.494
Inside Tube Surface (ft^2)	9.276	6.842
Tube Weight (lbs.)	3.519	1.064

Fig. 6, in curve "M" thereon, illustrates the air side pressure drop for a conventional core for various air flows. Curve "N" illustrates the air side pressure drop for the core of the present invention. It will be appreciated that the air side pressure drop, and thus fan energy, is reduced when a core made according to the invention is utilized.

Claims

1. A heat exchanger for exchanging heat between the ambient and a refrigerant that may be in a liquid or vapor phase, comprising:

a pair of spaced headers;

one of said headers having a refrigerant inlet;

one of said headers having a refrigerant outlet;

a heat exchanger tube extending between said headers and in fluid communication with each of said headers;

said tube defining a plurality of hydraulically parallel refrigerant flow paths between said headers;

each of said refrigerant flow paths having a hydraulic diameter up to about 0.07 inches;

hydraulic diameter being defined as the cross-sectional areas of a flow path multiplied by four (4) and divided by the wetted perimeter of the corresponding flow path.

2. The heat exchanger of claim 1 wherein there are a plurality of said tubes extending between said headers.

3. The heat exchanger of claim 1 wherein said outlet is a condensate outlet and said heat exchanger is a condenser.

4. A heat exchanger for exchanging heat between the ambient and a refrigerant that may be in a liquid or vapor phase, comprising:
first and second spaced headers;
an inlet in one of said headers;
an outlet in the other of said headers;
means including at least one tube means in fluid communication with said headers and defining a plurality of hydraulically parallel refrigerant flow paths extending between said headers in a plurality of generally parallel runs, said refrigerant flow paths having a relatively small hydraulic diameter up to about 0.07 inches where hydraulic diameter is four (4) times the cross-sectional area of the flow path divided by the wetted perimeter of the flow path; and
serpentine fins extending between and bonded to adjacent ones of said runs.

5. The heat exchanger of claim 4 wherein said flow paths include a crevice.

6. A condenser for a refrigerant in a cooling system comprising:
a pair of spaced, generally parallel, elongated headers, each having an elongated refrigerant receiving interior;
a vapor inlet in one of said headers;
a condensate outlet from one of said headers;
said headers each having a series of elongated generally parallel slots with the slots in the series on one header aligned with and facing the slots in the series on the other header;
a tube row defined by a plurality of straight tubes of flat cross section and with flat side walls and having opposed ends extending in parallel between said headers, the ends of said flat cross section tubes being disposed in corresponding aligned ones of said slots and in fluid communication with the interiors of said headers, at least some of said tubes being in hydraulic parallel with each other;
web means within said flat cross section tubes and extending between and joined to the flat side walls at spaced intervals to (a) define a plurality of flow paths within each flat cross section tube between said headers;
to (b) absorb forces resulting from internal pressure within said condenser and tending to expand the flat cross section tubes;
and to (c) conduct heat between both said flat sides and fluid in said flow paths, said flow paths being of relatively small hydraulic diameter defined as the cross-sectional area of the corresponding flow path multiplied by four and divided by the wetted perimeter of the corresponding flow path; and
serpentine fins incapable of supporting said flat cross section tubes against substantial internal pressure extending between facing flat side walls of adjacent flat cross section tubes.

7. The condenser of claim 6 wherein said web means are defined by undulating spacers within each flat cross section tube and having crests with alternating crests bonded to opposite flat side walls.

8. The condenser of claim 6 wherein said web means and/or said flat side walls further define at least one concave zone at the intersection of converging surface segments in each of said flow paths extending along the length thereof.

9. The condenser of claim 8 wherein there are a plurality of said concave zones for at least some of said flow paths.

10. A condenser for a refrigerant in a cooling system comprising: a pair of spaced, generally parallel, headers (10, 12); a vapor inlet (32) in one of said headers; a condensate outlet (26) from one of said headers; said headers each having a series of elongated generally parallel slots (14) with the slots in

the series on one header aligned with and facing the slots in the series on the other header; a tube row defined by a plurality of straight tubes (20) of flat cross-section and with flat side walls and having opposed ends extending in parallel between said headers, the ends of the said flat cross section tubes being disposed in corresponding aligned ones of said slots and in fluid communication with the interiors of said headers, at least some of said tubes being in hydraulic parallel with each other; web means (40) within said flat cross-section tubes and extending between and joined to the flat side walls at spaced intervals to (a) defined a plurality of discrete, hydraulically parallel flow paths within each flat cross-section tube that extend between said headers; to (b) absorb forces resulting from internal pressure within said condenser and tending to expand the flat cross-section tubes; and to (c) conduct heat between both said flat sides and fluid in said flow paths, said flow paths being of relatively small hydraulic diameter which is defined as the cross-sectional area of the corresponding flow path multiplied by four and divided by the wetted perimeter of the corresponding flow path; and serpentine fins (34) extending between facing flat side walls of adjacent flat cross-section tubes.

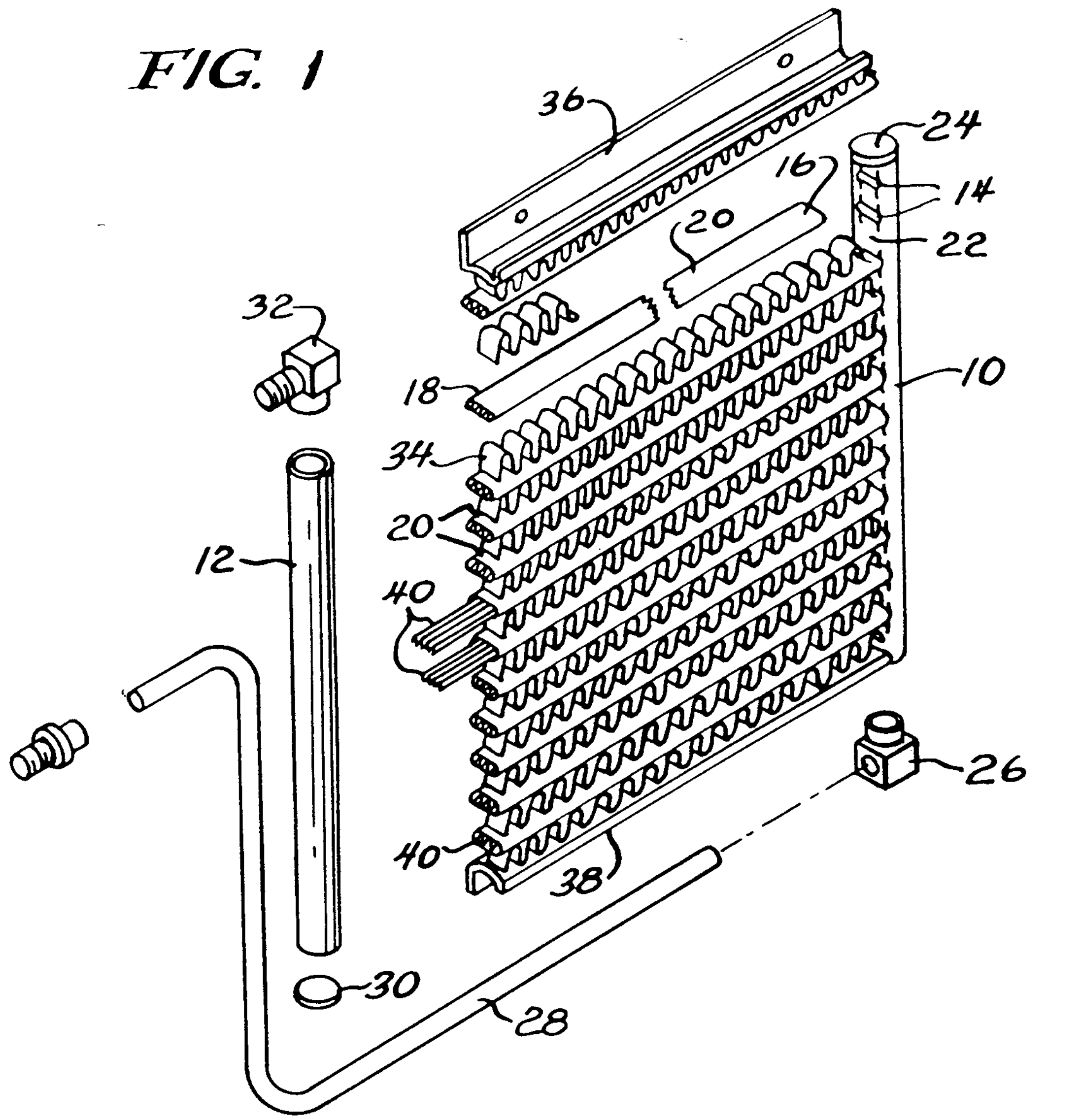
11. A condenser according to Claim10 wherein said headers are defined by generally cylindrical tubes.

12. A condenser according to claim 10 or claim 11 wherein, between said slots, each of said headers is provided with a part-spherical dome.

13. A condenser according to any of claims 10-12 wherein the web is joined to said flat side walls by fillets (44) of solder or braze metal.

14. A condenser according to any preceding claim, wherein said serpentine fins are incapable of supporting said flat cross-section tubes against substantial internal pressure.

FIG. 1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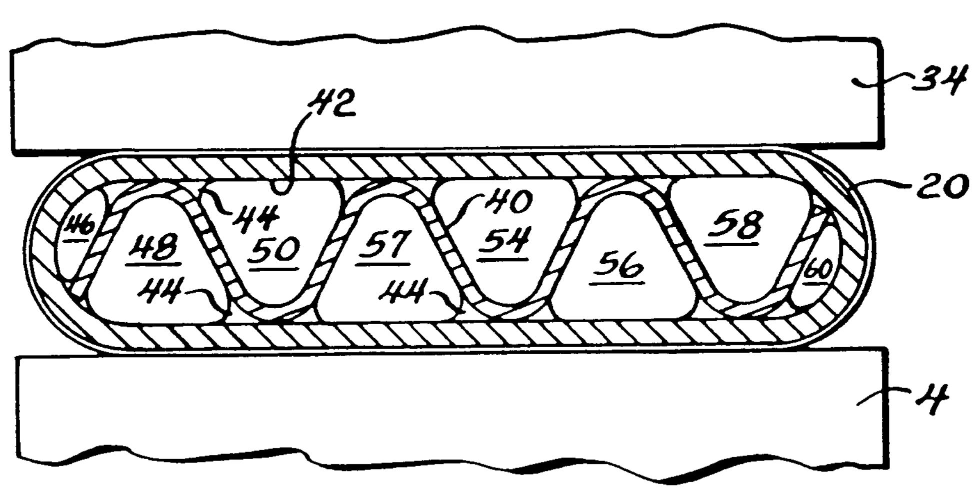
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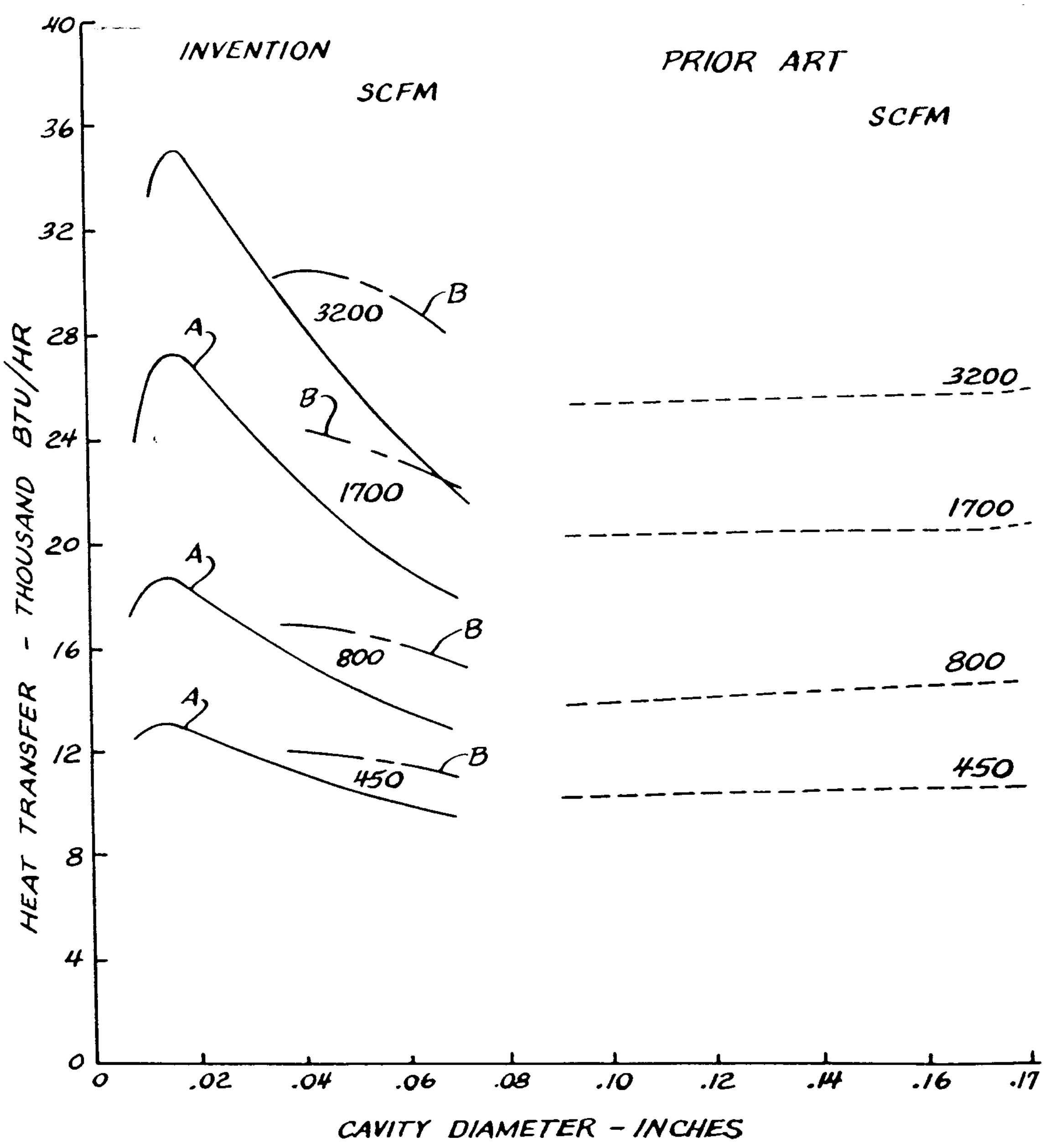
FIG. 2



INVENTION
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PRIOR ART
SCFM
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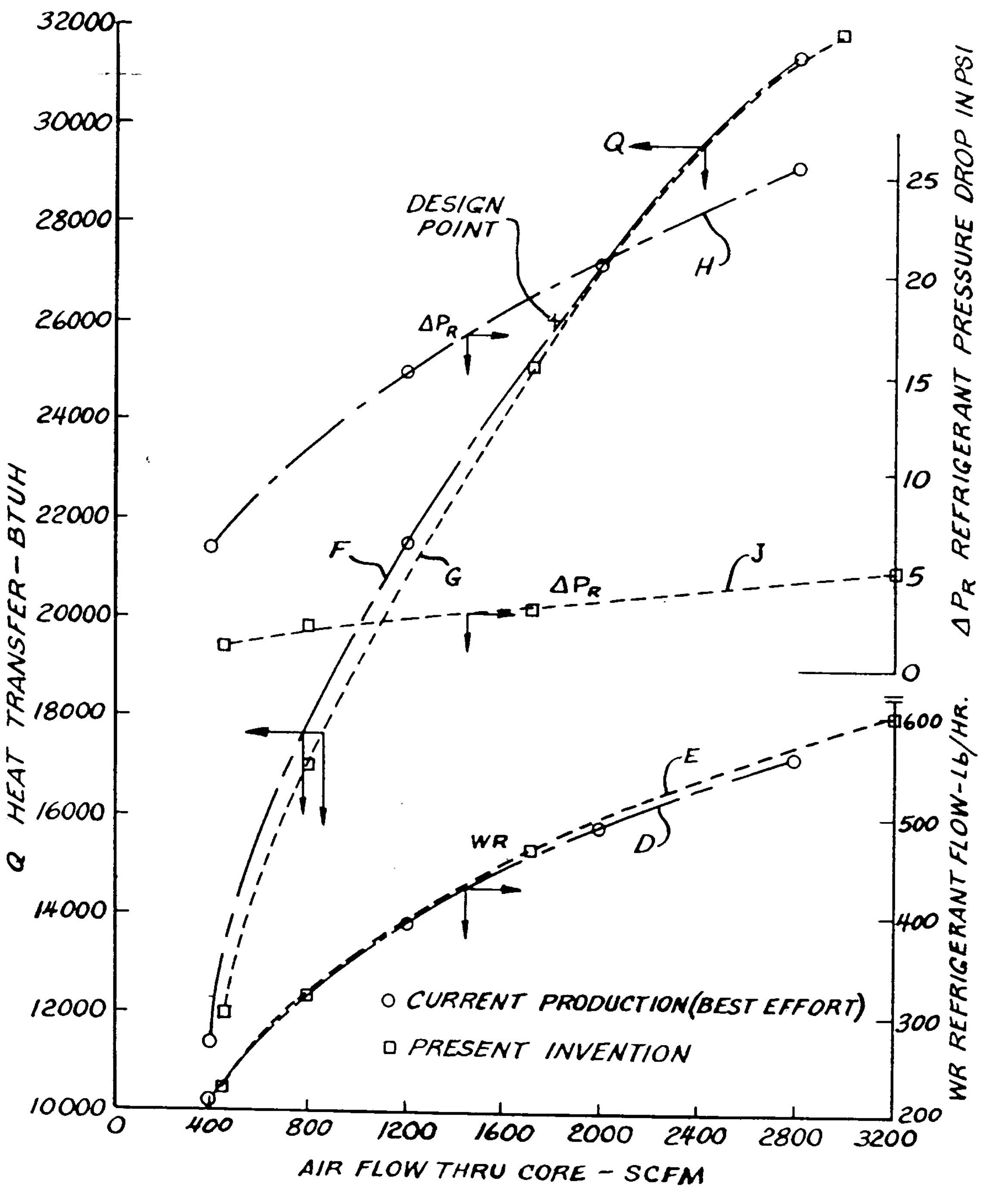
CAVITY DIAMETER - INCHES

FIG.3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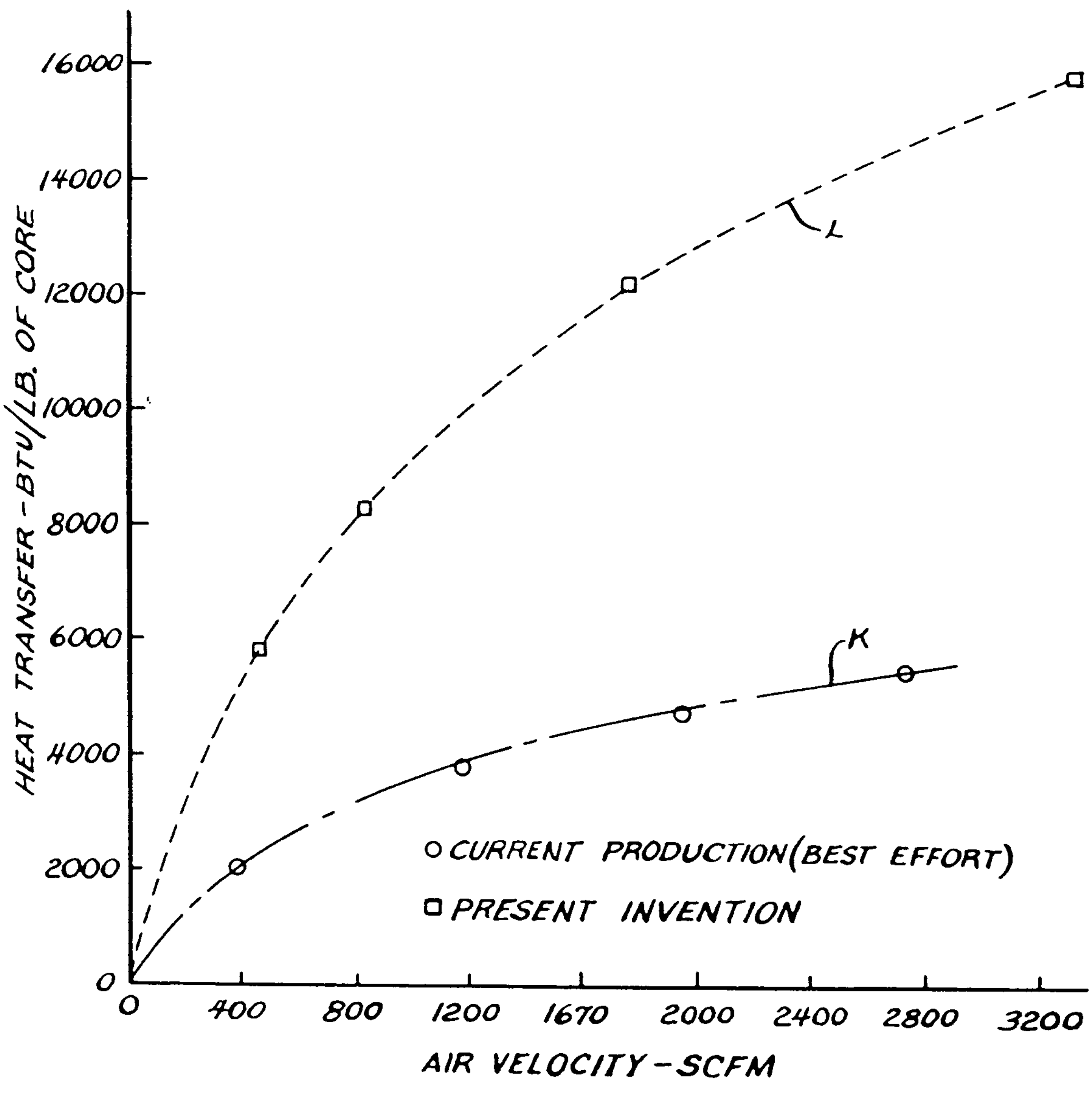
□ PRESENT INVENTION

FIG. 4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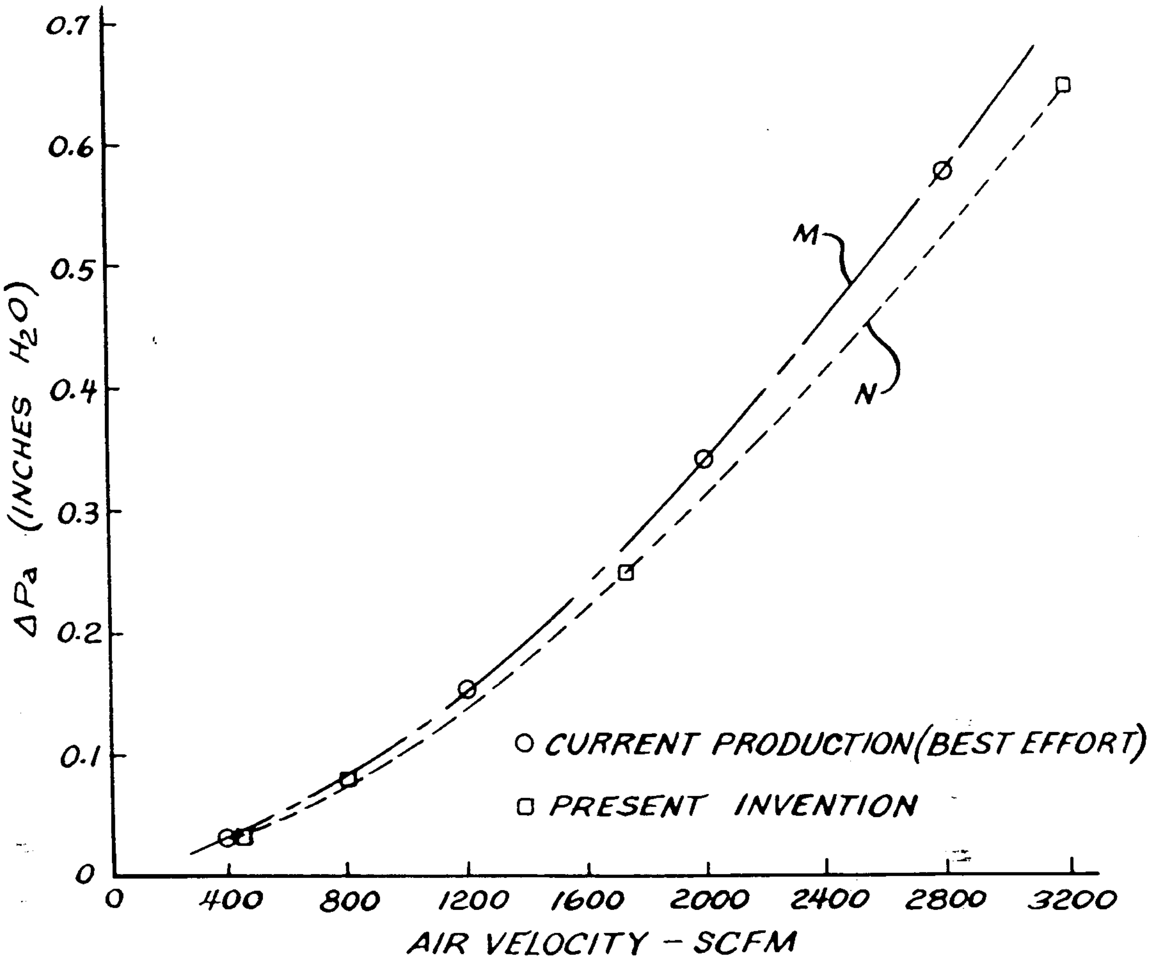
CURRENT PRODUCTION (BEST EFFORT)
PRESENT INVENTION

FIG. 5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□ PRESENT INVENTION
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AIR VELOCITY - SCFM

FIG. 6