(19) Europäisches Patentamt
European Patent Office
Office européen des brevets

(11) EP 0 219 974 B1

(12) EUROPEAN PATENT SPECIFICATION

(45) Date of publication and mention of the grant of the patent:
06.11.1996 Bulletin 1996/45

(51) Int Cl.[6]: **F25B 39/04**, F28F 1/02

(21) Application number: **86307161.9**

(22) Date of filing: **17.09.1986**

(54) **Condenser with small hydraulic diameter flow path**

Verflüssiger mit einen kleinen hydraulischen Durchmesser aufweisender Strömungsbahn

Condenseur à branche d'écoulement à petit diamètre hydraulique

(84) Designated Contracting States:
AT DE FR GB IT NL SE

(30) Priority: **02.10.1985 US 783087**
05.09.1986 US 902697

(43) Date of publication of application:
29.04.1987 Bulletin 1987/18

(60) Divisional application: **93202885.5**

(73) Proprietor: **Modine Manufacturing Company**
Racine Wisconsin 53401 (US)

(72) Inventors:
- **Guntly, Leon Arnold**
  Wisconsin 53404 (US)
- **Dudley, Jack C.**
  Racine Wisconsin 53403 (US)

(74) Representative: **Allden, Thomas Stanley et al**
A.A. THORNTON & CO.
Northumberland House
303-306 High Holborn
London WC1V 7LE (GB)

(56) References cited:
EP-A- 0 237 164
WO-A-84/01208
CH-A- 221 087
GB-A- 1 559 318
GB-A- 2 059 562
GB-A- 2 133 525
US-A- 1 768 258
US-A- 1 958 226
US-A- 2 136 641
US-A- 3 486 489
US-A- 4 190 105

- **PATENT ABSTRACTS OF JAPAN, vol. 9, no. 73 (M-368)[1796], 3rd April 1985; & JP-A-59 205 591 (NIPPON DENSO K.K.) 21-11-1984**

Printed by Jouve, 75001 PARIS (FR)

Description

This invention relates to a condenser, and more particularly to a condenser for use in an air conditioning or refrigeration system for condensing a refrigerant.

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 resistance to the flow of vapour and/or condensate. A condenser of this type is shown in US-A-2136641.

The large tube size of such condensers 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.

An alternative design of condenser for use in refrigeration apparatus is shown in US-A-1958226. In this design a multiplicity of individual tubes extend in a matrix arrangement between spaced apart headers. This design produces a heat exchanger which is both bulky and heavy and is accordingly unsuitable for applications where a high degree of efficiency (measured in terms of cooling capacity per unit volume or per unit weight) is required.

The present invention is directed to overcoming the above problems.

In accordance with the present invention an air cooled condenser suitable for use in a refrigeration or air conditioning system to condense a refrigerant vapour into a refrigerant liquid, comprises a pair of spaced headers for receiving refrigerant vapour and collecting condensed refrigerant; and a plurality of tubes extending in hydraulic parallel between said headers, each tube being in fluid communication with each said header and being elongate in transverse cross-section with the minor dimension of the cross-section aligned substantially perpendicular to the direction of air flow through the condenser is characterised in that each said tube defines a plurality of discrete hydraulically parallel fluid flow paths, each said fluid flow path having a hydraulic diameter in the range of 0.381 to 1.778mm (0.015 to 0.070 inches).

The preferred embodiment of the invention provides a condenser which 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 vapour and/or condensate side pressure drop.

In the preferred embodiment of the invention the tubes are flat tubes.

In a highly preferred embodiment, the plurality of flow paths in each tube are defined by an undulating spacer contained within the tube.

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

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

The invention will become apparent from the following specification, taken in connection with the accompanying drawings, wherein:

FIGURE 1 is an exploded, perspective view of an embodiment of condenser made according to the invention;
FIGURE 2 is a fragmentary, enlarged, cross-sectional view of a condenser tube that may be employed in the invention;
FIGURE 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;
FIGURE 4 is a graph comparing an embodiment of the present invention with a 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;
FIGURE 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 unit mass of material employed in making up the core of each; and
FIGURE 6 is a further graph comparing the prior art construction with an embodiment of the present invention by plotting air velocity versus pressure drop across the air side of the condenser.

An exemplary embodiment of a condenser made according to the invention is illustrated in Figure 1 and is seen to include opposed, spaced, generally parallel headers 10 and 12. Preferably, the headers 10 and 12 are 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 US-A-4615385 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 vapour inlet while the other serves as a condensate outlet. For the orientation shown in Figure 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 parallel to 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 Figure 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 Figure 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 described embodiment of the invention, however, the desired support in a serpentine fin heat exchanger is accomplished by the fact that the spacer 40 and the crests thereof are bonded along its entire length to 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.

A highly preferred means by which the tubes 20 with accompanying inserts 40 may be formed is disclosed in US-A-4688311 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 0.381 to 1.778mm (0.015 to 0.070 inches). Given current assembly techniques known in the art, a hydraulic diameter of approximately 0.889mm (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 Figure 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 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 vapour and/or conduit side.

A number of advantages of the invention will be apparent from the data illustrated in Figures 3-6 inclusive and from the following discussion. Figure 3 for example, on the right-hand side, plots the heat transfer rate against the cavity or hydraulic diameter at air flows varying from 12.74 to 90.61m^3 (450 to 3200 Standard Cubic Feet) per minute for production condenser cores made by the applicant. Heat transfer rate is plotted in kW (thousands of BTU per hour) and the hydraulic diameter is plotted in mm (inches).

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 Figure 1 having a frontal area of 0.186m- (two square feet) utilizing tubes approximately 0.61m (24 inches) long and having a 0.381mm (0.015 inch) tube wall thickness, a 13.51mm (0.532 inch) tube major dimension, 43.3°C (110°F) inlet air, 82.2°C (180°F) inlet temperature and 1.619 MPa (235 psig) pressure for R-12 and assuming 1.1 degree C (2 degree F) of subcooling of the exiting refrigerant after condensation. The core was provided with 18 fins per 25.4mm (inch) between tubes and the fins were 15.88mm (0.625 inches) by 13.72mm (0.540 inches by 0.152mm (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 Figure 3, heat transfer is advantageously and substantially increased in the range of hydraulic diameters of about 0.381 to 1.778mm (0.015 to 0.070 inches) through the use of the invention with some variance depending upon air flow.

Turning now to Figure 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 applicant as "1E2803". The data for the conventional core is likewise listed in Table 1 below. In Figure 4: heat transfer rate is plotted in kW (thousands of BTU per hour); air flow rate is plotted in m^3 (Standard Cubic Feet) per minute; refrigerant flow is plotted in kg (pounds) per hour; and refrigerant pressure drop is plotted in kPa (PSI).

Both the core made according to the invention and the conventional core have the same design point which is, as shown in Figure 4, a heat transfer rate of 7.62kW (26,000 BTU per hour) at an air flow of 50.97m^3 (1800 Standard Cubic Feet) per minute. The actual observed equivalence of the two cores occurred at 8.21kW (28,000 BTU per hour) and 56.63m^3 (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 Figures 4-6, R-12 was applied to the condenser inlet at 1.619MPa (235 psig) at 82.2°C (180°F). The exiting refrigerant was subcooled 1.1 degrees C (2 degrees F). Inlet air temperature to the condenser was 43.3°C (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 an embodiment of 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 an embodiment 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 vapour 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 an embodiment of 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 an embodiment of the invention, when compared with the conventional core, holds less refrigerant. Thus, the core of embodiment 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 Figure 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, Figure 5 compares, at various air velocities, the heat transfer rate per unit mass of core of the conventional condenser (curve

"K") versus heat transfer per unit mass of core of a condenser made according to the invention (curve "L"). In Figure 5 heat transfer rate per unit mass is plotted in W kg^{-1} (BTU per pound) and air flow is plotted in m^3 (Standard Cubic Feet) per minute. Thus Figure 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. 4-6		
CORE PROPERTIES	CURRENT PRODUCTION 1E2803	PRESENT INVENTION
Depth mm (in.)	24.97 (.938)	13.72 (.540)
Heights mm (in.)	311.81 (12.276)	304.8 (12.00)
Length mm (in.)	612.90 (24.13)	599.19 (23.259)
Face Area m^2 ($ft.^2$)	0.191 (2.057)	0.18 (1.938)
Weight kg (lbs.)	2.577 (5.682)	0.933 (2.057)
Ratio $\frac{\text{outside surface}}{\text{inside surface}}$	4.478	5.391
FIN PROPERTIES		
Fins per 25.4mm	12	18
Fin Rows	13	21
Fin Thickness mm (in.)	0.203 (.008)	0.102(.004)
Fin Height mm (in.)	19.06 (.7502)	12.75 (.5018)
Free Flow Area m^2 ($ft.^2$)	0.134 (1.444)	0.144 (1.554)
Surface Area m^2 ($ft.^2$)	3.45 (37.110)	3.102 (33.389)
Hydraulic Diameter mm (in.)	3.312 (.1304)	2.311 (.0910)
Fin Weight kg (lbs.)	0.981 (2.163)	0.450 (.993)
TUBE PROPERTIES		
No. Circuits	2	20
Tube Rows	14	20
Tube Thickness mm (in.)	4.75 (.187)	1.91 (.075)
Tube Wall mm (in.)	0.686 (.027)	0.381 (.015)
Tube Length mm (ft.)	385.27 (15.168)	51.99 (2.047)
Free Flow Area mm^2 ($in.^2$)	100.39 (.1556)	206.45 (.3200)
Hydraulic Diameter mm (in.)	2.0 (.07871)	0.767 (.0302)
Outside Tube Surface m^2 ($ft.^2$)	0.412 (4.431)	0.325 (3.494)
Inside Tube Surface m^2 (ft^2)	0.862 (9.276)	0.636 (6.842)
Tube Weight kg (lbs.)	1.596 (3.519)	0.483 (1.064)

Figure 6, in curve "M" thereon, illustrates the air side pressure drop, plotted in Pa (inches of water), for a conventional core and for a core according to the invention for various air flows plotted in m^3 (Standard Cubic Feet) per minute. 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. An air cooled condenser suitable for use in a refrigeration or air conditioning system to condense a refrigerant vapour into a refrigerant liquid, the condenser comprising a pair of spaced headers (10,12) for receiving refrigerant vapour and collecting condensed refrigerant; and a plurality of tubes (20) extending in hydraulic parallel between said headers, each tube being in fluid communication with each said header and being elongate in transverse cross-section with the minor dimension of the cross-section aligned substantially perpendicular to the direction of air flow through the condenser, characterised in that each said tube defines a plurality of discrete hydraulically parallel fluid flow paths, each said fluid flow path having a hydraulic diameter in the range of 0.381 to 1.778mm (0.015 to 0.070 inches).

2. A condenser according to claim 1 wherein said tubes are flattened tubes and the plurality of flow paths in each tube is defined by an undulating spacer (40) contained within the tube.

3. A condenser according to claim 1 or claim 2 further including fins (34) on the exteriors of said condenser tubes.

4. A condenser according to any preceding claim including fins (34) extending between the exteriors of adjacent ones of said condenser tubes.

5. A condenser according to any preceding claim wherein said headers are defined by generally cylindrical tubes and have facing openings (14) for receiving respective ends 16,18) of said condenser tubes.

6. A condenser according to claim 5 wherein said openings are a series of elongated slots (14), the slots on one header tube facing the slots on the other header tube; and wherein the condenser tubes are flattened tubes (20) having opposed ends (16,18) disposed in corresponding ones of said slots.

Patentansprüche

1. Luftgekühlter Kondensator, der sich zum Einsatz in einem Kühl- oder Klimasystem zum Kondensieren eines Kühlmitteldampfes zu einer Kühlmittelflüssigkeit eignet, wobei der Kondensator ein Paar beabstandeter Sammelrohre (10,12) zur Aufnahme von Kühlmitteldampf und zum Sammeln von kondensiertem Kühlmittel umfaßt; sowie eine Vielzahl von Röhren (20), die hydraulisch parallel zwischen den Sammelrohren verlaufen, wobei jede Röhre mit jedem der Sammelrohre in Fluidverbindung steht und im Querschnitt länglich ist, wobei die kleinere Abmessung des Querschnitts im wesentlichen senkrecht zur Richtung des Luftstroms durch den Kondensator ausgerichtet ist, **dadurch gekennzeichnet,** daß jede der Röhren eine Vielzahl getrennter, hydraulisch paralleler Fluidströmungswege aufweist, wobei jeder der Fluidströmungswege einen hydraulischen Durchmesser im Bereich von 0,381 bis 1,778 mm (0,015 bis 0,071 inch) aufweist.

2. Kondensator nach Anspruch 1, wobei die Röhren abgeflachte Röhren sind und die Vielzahl von Strömungswegen in jeder Röhre durch einen wellenförmigen Abstandhalter (40) gebildet wird, der in der Röhre enthalten ist.

3. Kondensator nach Anspruch 1 oder Anspruch 2, der des weiteren Rippen (34) an den Außenseiten der Kondensatorröhren enthält.

4. Kondensator nach einem der vorangehenden Ansprüche, der Rippen (34) enthält, die sich zwischen den Außenseiten benachbarter Kondensatorröhren erstrecken.

5. Kondensator nach einem der vorangehenden Ansprüche, wobei die Sammelrohre durch im allgemeinen zylindrische Röhren gebildet werden und einander zugewandte Öffnungen (14) aufweisen, die entsprechende Enden (16, 18) der Kondensatorröhren aufnehmen.

6. Kondensator nach Anspruch 5, wobei die Öffnungen eine Reihe länglicher Schlitze (14) sind, wobei die Schlitze an einem Sammelrohr den Schlitzen an dem anderen Sammelrohr zugewandt sind; und wobei die Kondensatorröhren abgeflachte Röhren (20) mit einander gegenüberliegenden Enden (16, 18) sind, die den Schlitzen entsprechend angeordnet sind.

Revendications

1. Condenseur refroidi par air qui peut être utilisé dans un système de réfrigération ou de conditionnement d'air pour la condensation de vapeur d'un fluide réfrigérant en un liquide réfrigérant, le condenseur comprenant deux collecteurs distants (10, 12) destinés à recevoir la vapeur du fluide réfrigérant et à collecter le fluide réfrigérant condensé, et plusieurs tubes (20) disposés hydrauliquement en parallèle entre les collecteurs, chaque tube communiquant avec chaque collecteur et étant allongé en direction transversale avec une petite dimension en coupe alignée en direction pratiquement perpendiculaire à la direction de la circulation de l'air dans le compresseur, caractérisé en ce que chaque tube délimite plusieurs trajets séparés hydrauliquement et parallèles de circulation de fluide, chaque trajet de circulation de fluide ayant un diamètre hydraulique compris entre 0,381 et 1,778 mm (0,015 à 0,70 pouce).

2. Condenseur selon la revendication 1, dans lequel les tubes sont des tubes aplatis, et les trajets de circulation de chaque tube sont délimités par une entretoise ondulée (40) contenue dans le tube.

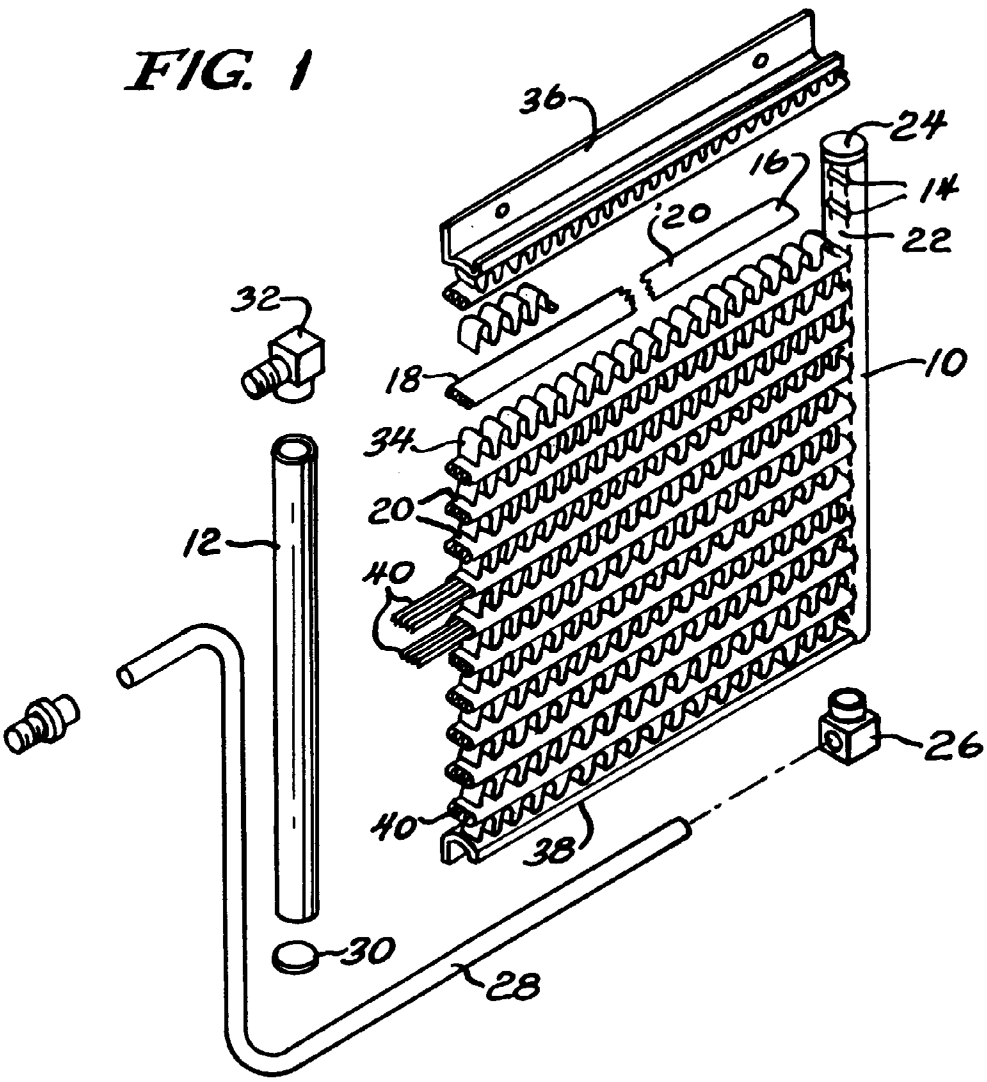
3. Condenseur selon la revendication 1 ou 2, comprenant en outre des ailettes (34) placées à l'extérieur des tubes du condenseur.

4. Condenseur selon l'une quelconque des revendications précédentes, comprenant des ailettes (34) disposées entre les parties extérieures de tubes adjacents du condenseur.

5. Condenseur selon l'une quelconque des revendications précédentes, dans lequel les collecteurs sont délimités par des tubes de forme générale cylindrique et ayant des ouvertures en regard (14) pour le logement des extrémités respectives (16, 18) des tubes du condenseur.

6. Condenseur selon la revendication 5, dans lequel les ouvertures sont formées d'une série de fentes allongées (14), les fentes d'un tube collecteur étant tournées vers les fentes de l'autre tube collecteur, et les tubes du condenseur sont des tubes aplatis (20) ayant des extrémités opposées (16, 18) disposées dans des fentes correspondantes.

FIG. 1


36
24
16
14
20
22
32
18
10
34
12
20
40
26
40
38
30
28

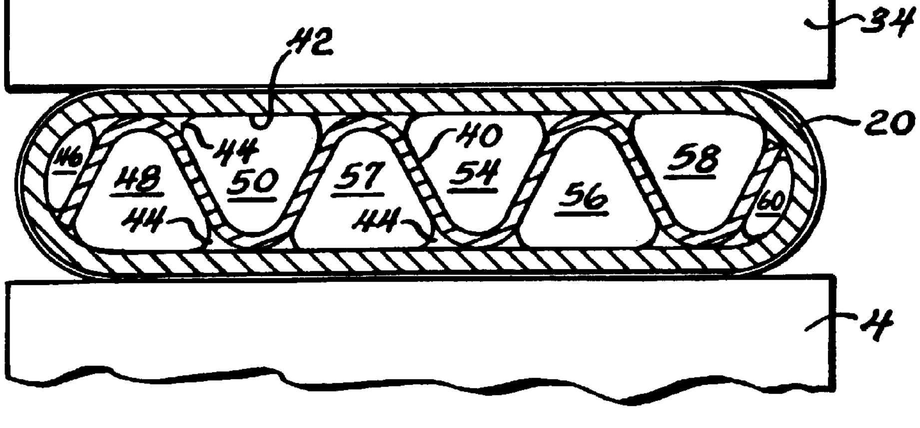

34
42
20
44
40
46
48
50
57
54
58
56
60
44
44
4

FIG. 2

Fig.3.

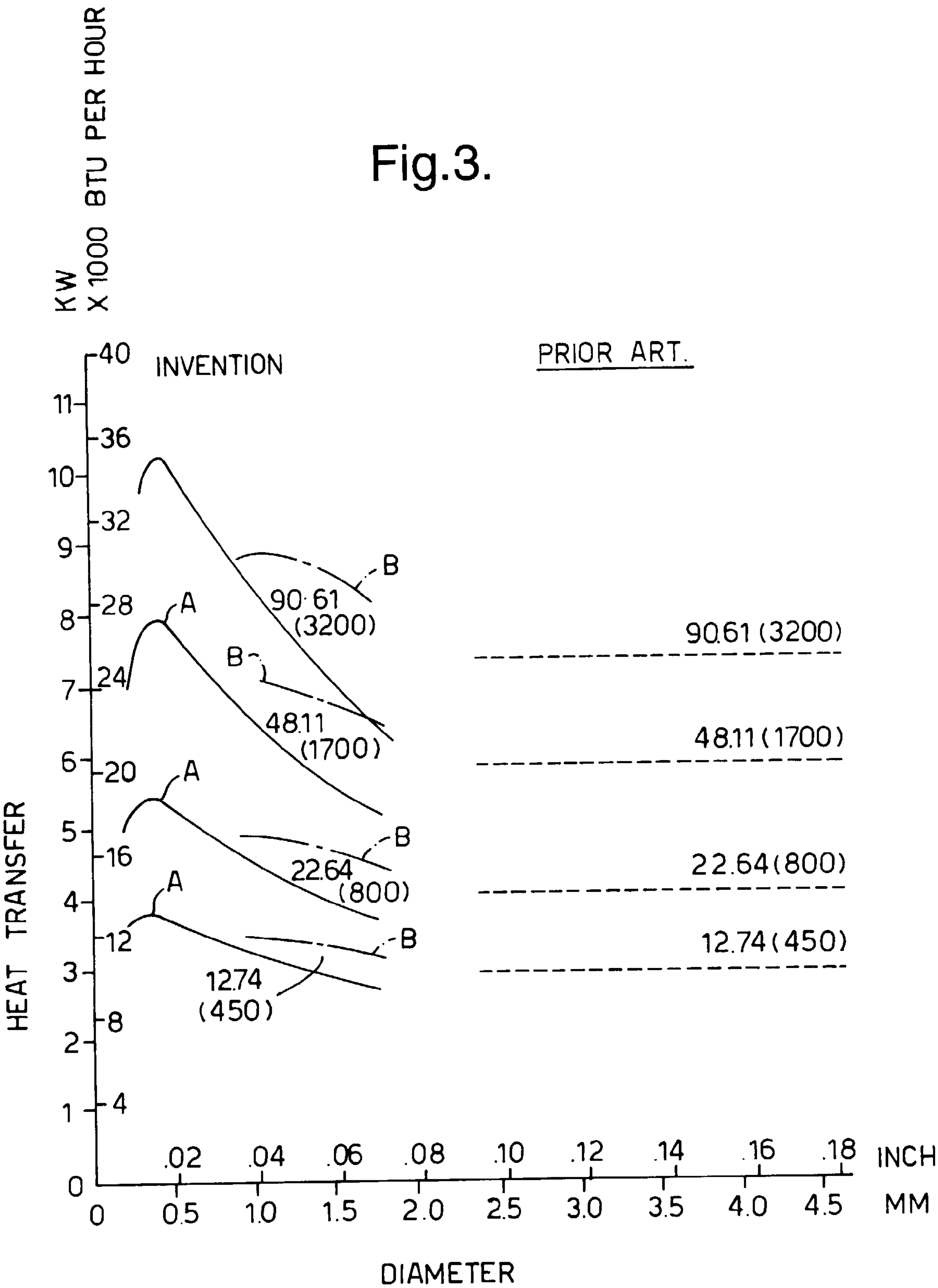

KW
X 1000 BTU PER HOUR
INVENTION
PRIOR ART.
HEAT TRANSFER
A
B
90·61 (3200)
48.11 (1700)
22.64 (800)
12.74 (450)
90.61 (3200)
48.11 (1700)
22.64 (800)
12.74 (450)
.02 .04 .06 .08 .10 .12 .14 .16 .18 INCH
0 0.5 1.0 1.5 2.0 2.5 3.0 3.5 4.0 4.5 MM
DIAMETER

Fig.4.

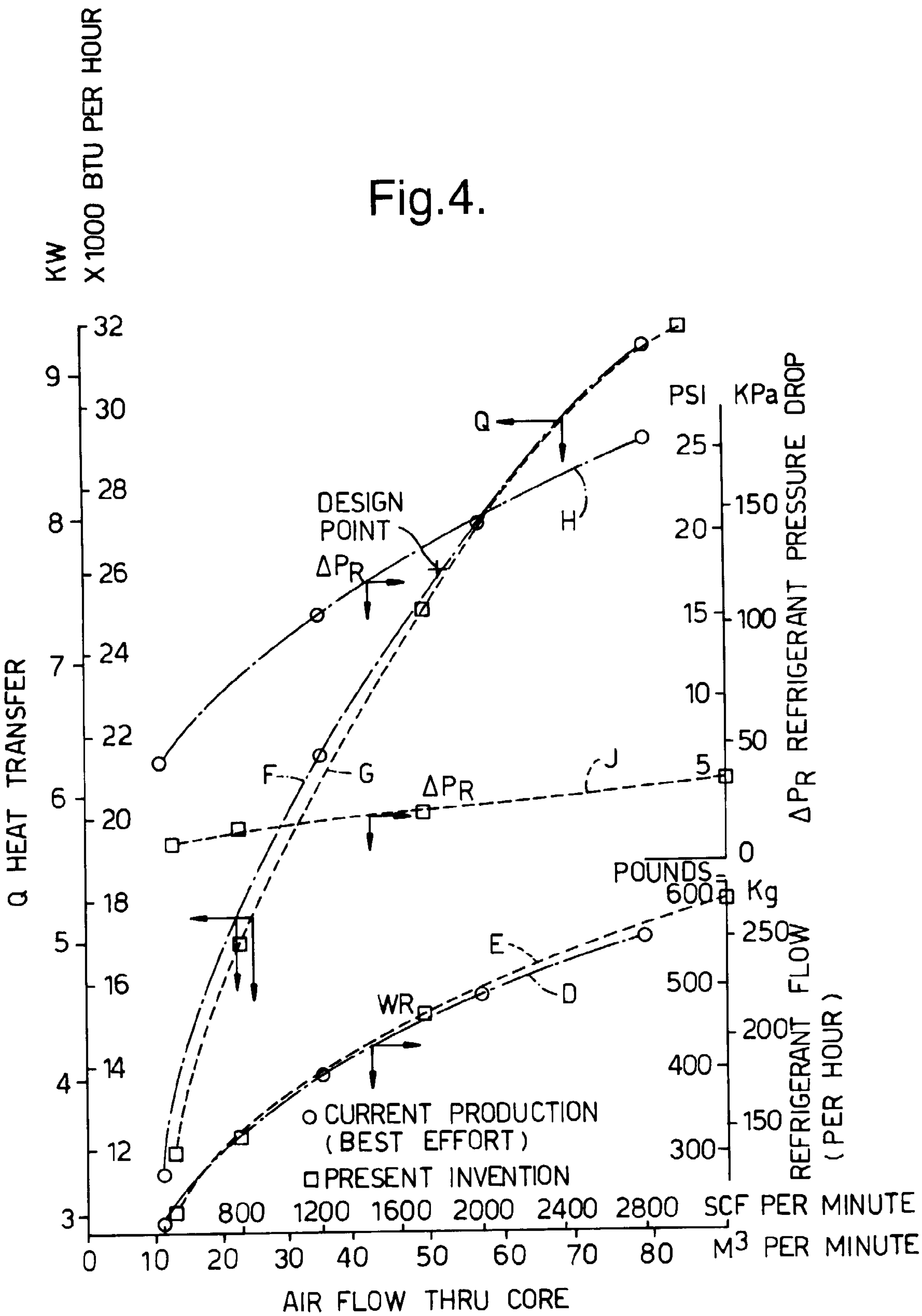

KW
X 1000 BTU PER HOUR
Q HEAT TRANSFER
PSI
KPa
ΔPR REFRIGERANT PRESSURE DROP
Q
DESIGN POINT
ΔPR
H
F
G
J
POUNDS
Kg
E
D
WR
REFRIGERANT FLOW (PER HOUR)
o CURRENT PRODUCTION (BEST EFFORT)
□ PRESENT INVENTION
SCF PER MINUTE
M3 PER MINUTE
AIR FLOW THRU CORE

Fig.5.

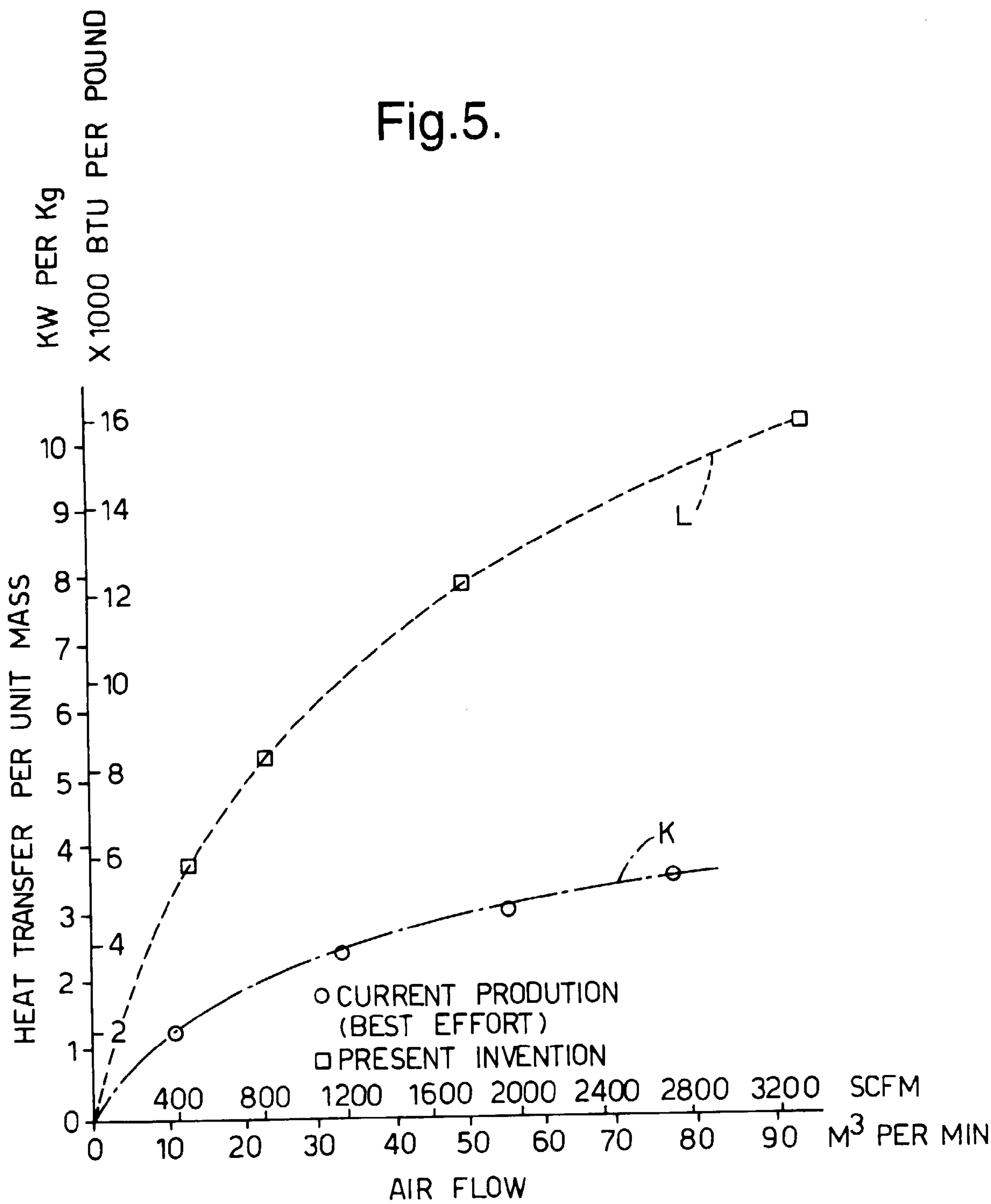
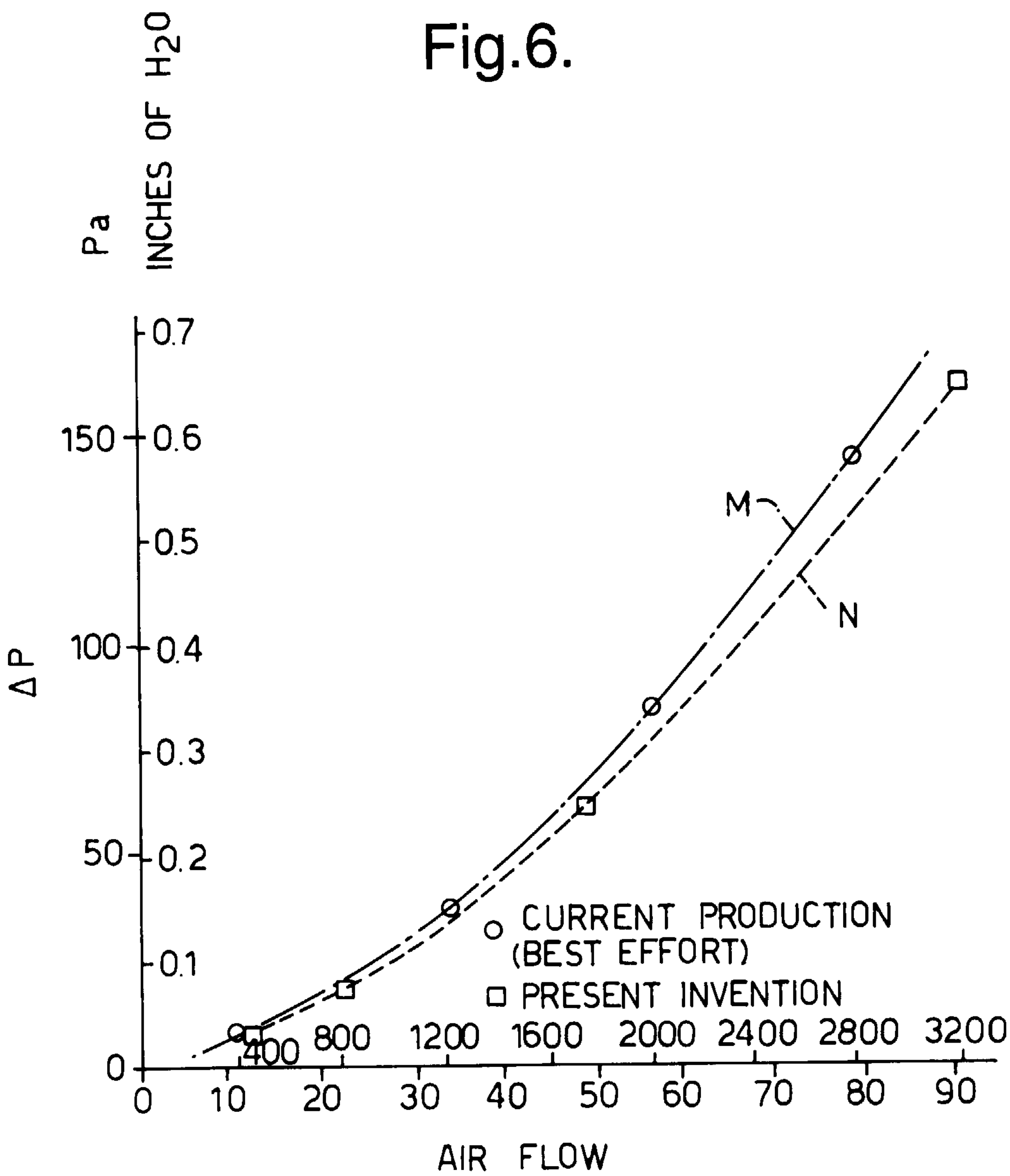

KW PER Kg
X 1000 BTU PER POUND
HEAT TRANSFER PER UNIT MASS
0
1
2
3
4
5
6
7
8
9
10
2
4
6
8
10
12
14
16
L
K
O CURRENT PRODUTION
(BEST EFFORT)
□ PRESENT INVENTION
400
800
1200
1600
2000
2400
2800
3200
SCFM
0
10
20
30
40
50
60
70
80
90
M3 PER MIN
AIR FLOW

Fig.6.


Pa
INCHES OF H_2O
ΔP
0.7
150
0.6
0.5
100
0.4
0.3
50
0.2
0.1
0
M
N
CURRENT PRODUCTION
(BEST EFFORT)
PRESENT INVENTION
400 800 1200 1600 2000 2400 2800 3200
0 10 20 30 40 50 60 70 80 90
AIR FLOW