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(73) Proprietor: **SHOWA ALUMINUM KABUSHIKI KAISHA
224-banchi
Kaizancho 6-cho
Sakaishi
Osaka (JP)**

(72) Inventor: **Sasaki, Hironaka
224-banchi,
Kaizancho 6-cho
Sakaishi,
Osaka (JP)**
Inventor: **Watanabe, Hirohiki
224-banchi,
Kaizancho 6-cho
Sakaishi,
Osaka (JP)**
Inventor: **Tategami, Tetsuya
224-banchi,
Kaizancho 6-cho
Sakaishi,
Osaka (JP)**

(74) Representative: **Kerr, Simonne June et al
European Patent Attorney
POTTS, KERR & CO.
P.O. Box 688
Ascot
Berkshire SL5 8YT (GB)**

EP 0 414 433 B1

Description

The invention relates to a condenser comprising two unit head exchangers.

EP-0 401 752, falling within the terms of Article 54, paragraph 3, EPC, discloses a condenser for a refrigerant of a vehicle air conditioning system having adjacent assemblies which are mechanically connected via their ribbed portion, the mean thermal conductance λ m lies, however, 20 % below the thermal conductance λ of the material of the ribbed portion of the two adjacent assemblies in a connection zone between every two adjacent assemblies.

US-A-3229760 comprises a heat exchange apparatus more particularly a radiator structure provided with cooling fins mounted in a main housing provided with a plurality of spaced apart fluid conduits and a guard structure being provided to protect the fins.

US-A-4531574 provides a mounting for attaching an oil cooler to a radiator so as to be in series therewith in the path followed by the coolant air.

A multi-flow type heat exchanger is disclosed in Japanese Patent Publication Kokai 63-34466 and has a structure such that a plurality of parallel flat tubes are connected to a pair of hollow headers at their opposite ends, respectively, with a corrugated fin interposed between one such flat tube and the next. In operation, heat exchange occurs between coolant and ambient air which flows through spaces defined between the tubes while the coolant flows through a coolant circuit composed of the flat tubes. The known multi-flow type heat exchanger can be made thinner than the other known heat exchangers in its dimension in a direction of air flow, without affecting the efficiency of heat exchange. Therefore, the multi-flow type heat exchanger has proved itself better in performance than the other known heat exchangers of some types such as the serpentine type.

An object of the invention is therefore to provide a heat exchanger which is adapted to increase the heat transfer capacity thereof without necessitating an excessively wide space.

The present invention comprises a condenser with two unit heat exchangers which respectively comprise a plurality of flat tubes arranged parallel with each other and fins each interposed between adjacent tubes with opposite ends of each tube being connected to a pair of headers in fluid connection therewith, the unit heat exchangers being closely juxtaposed to each other fore and aft in a direction of air flow so that coolant circuits of the unit heat exchangers are connected in series, each unit heat exchanger comprising partitioning means fixed inside at least one of the headers so as to divide an internal space thereof into at least two sections in a manner such that the coolant paths in the unit heat exchangers are meanders which make zigzag turns caused by the partitioning means, the coolant flowing from one unit heat exchanger lying on the leeward side, into the other unit heat exchanger standing to the windward side of the air flow, the sections including an inlet section formed in the unit heat exchanger lying on the leeward side and an outlet section formed in one of the other unit heat exchangers standing to the windward side, a coolant inlet pipe being connected to the inlet section, whilst a coolant outlet pipe is connected to the outlet section, the total cross-sectional area of the coolant paths formed through the tubes connected to the outlet section being smaller than that formed through the other tubes connected to the inlet section, wherein heat exchange capacity of the leeward unit heat exchanger is greater than that of the windward unit heat exchanger. The present invention also relates to a condenser comprising two unit heat exchangers comprising the features of claim 7.

The invention will now be described further, by way of example, with reference to the accompanying drawings, in which:

Fig. 1 is a perspective view showing a duplex heat exchanger according to a first embodiment of the invention, in a separated state of a forehand and a rearward unit heat exchangers;

Fig. 2 is a front elevation of the entirety of the duplex heat exchanger shown in Fig. 1;

Fig. 3 is a plan view of said entirety;

Fig. 4 is similarly a side elevation of said entirety;

Fig. 5 is a perspective view showing a separated state of a header, tubes and corrugated fins of the forehand or rearward unit heat exchanger;

Fig. 6 is a cross section taken along a line 6 - 6 in Fig. 2;

Fig. 7 is an enlarged cross section of the forehand or rearward unit heat exchanger, seen in the same direction as in Fig. 6;

Fig. 8 is an enlarged front elevation showing the tubes and the corrugated fins;

Fig. 9 illustrates a coolant circuit in the duplex heat exchanger shown in Fig. 1;

Fig. 10 is a perspective view showing another duplex heat exchanger according to a second embodiment of the invention, in a separated state of a forehand and a rearward unit heat exchangers;

Fig. 11 illustrates a coolant circuit in the duplex heat exchanger shown in Fig. 10;

Fig. 12 is a perspective view showing still another duplex heat exchanger according to a third embodiment of the invention, in a separated state of a forehand and a rearward unit heat exchangers;

Fig. 13 is side elevation showing an assembled state of the duplex heat exchanger shown in Fig. 12;

Fig. 14 is a perspective view of a further duplex heat exchanger according to a fourth embodiment;

Fig. 15 illustrates a coolant circuit in the duplex heat exchanger shown in Fig. 14;

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

First Embodiment

Figs. 1 to 9 shows an embodiment in which the invention is applied to a condenser made of aluminum-based alloy for us as a car cooler. The reference symbol "H" denotes a duplex heat exchanger which comprises a forehand unit heat exchanger "A" located at an upstream side, as well as a rearward unit heat exchanger "B" located at downstream side with respect to a direction "W" of air flow.

The forehand unit heat exchanger "A" is composed of a plurality of horizontally disposed tubes 1 stacked in a vertical direction, of corrugated fins 2 interposed between two of such tubes adjacent to each other, and of a left-hand header 3 and a right-hand header 4. The tubes 1 are made of an extruded profile pipe of said aluminum-based alloy. Alternatively, the tubes 1 may be porous or perforated tubes such as "harmonica tubes" or may be made of an upset-welded pipe. The corrugated fins 2 are of substantially the same width as the tubes 1 and are soldered thereto. The corrugated fins 2 are made of the same or other aluminum-based alloy, and preferably are formed with louvers cut and raised from main bodies of the fins. A cylindrical pipe made of an aluminum-based alloy and having inner and/or outer surfaces coated with a soldering agent is used to manufacture the headers 3 and 4. Tube receiving apertures 5 are formed at regular intervals in a longitudinal direction of each header so that respective ends of each tube 1 are inserted in the tube receiving apertures and securedly soldered thereto. Cover plates 6 are fixed to an upper end and a lower end of the left-hand header 3, and other cover plates 7 are similarly fixed to an upper end and a lower end of the right-hand header 4. Side plates 8 are disposed outside of the outermost corrugated fins 2.

The rearward unit heat exchanger "B" comprises tubes 21, corrugated fins 22, a left-hand header 23 and a right-hand header 24 wherein tube receiving apertures 25, cover plates 26 and 27 and side plates 28 are provided in a manner similar to that in the forehand unit heat exchanger "A". However, a distance "LB" between the left-hand and the right-hand headers in the rearward unit heat exchanger "B" is greater than a similar distance "LA" in the forehand unit heat exchanger "A". By virtue of such a difference between the distances "LA" and "LB", the forehand and rearward headers do not overlap each other and the depth of the duplex heat exchanger as a whole is reduced to a significant degree. This enhances compactness of the heat exchanger so that space occupied by it in the automobiles or the likes is advantageously decreased.

Coolant paths of the forehand unit heat exchanger "A" are connected in series to those of the rearward one "B". In detail, a coolant inlet pipe 40 is connected to an upper portion of the left-hand header 23 of the rearward unit heat exchanger "B". A coolant outlet pipe 50 is connected to an upper portion of the left-hand header 3 of the forehand unit heat exchanger "A". Said left-hand headers 3 and 23 of the forehand and rearward unit heat exchangers "A" and "B" are interconnected by a joint pipe 60. The reference numerals 71 and 72 in Figs. 2 and 3 denote brackets for fixing said unit heat exchangers one to another.

A partition plate 29 in the left-hand header 23 is located at a middle portion thereof so that said header 23 of the rearward unit heat exchanger "B" is partitioned into an upper and a lower chambers. On the other hand, other partition plates 9 in the left-hand header 3 are positioned respectively above and below a middle portion thereof, thus partitioning said header 3 of the forehand unit heat exchanger "A" into three chambers. Further, still another partition plate 10 in the right-hand header 4 at a middle portion thereof partitions same into two chambers for the forehand unit heat exchanger "A". Due to the partitions 29, 9 and 10, coolant flows in such a manner as illustrated in Fig. 9 wherein the coolant enters the left-hand header 23 of the rearward heat exchanger "B" through the coolant inlet pipe 40 so that it makes a U-turn within said heat exchanger before flowing into the lower chamber of said header 23. The coolant then advances through the joint pipe 60 into the lower chamber of the left-hand header 3 of the forehand heat exchanger "A". Subsequently, the coolant makes U-turn three times from one group of tubes to the next group of tubes within the forehand heat exchanger "A" so as to rise from a bottom thereof. Upon arrival at the upper chamber of said lefthand header 3, the coolant leaves it through the coolant outlet pipe 50. Heat transfer occurs between an air flow indicated by an arrow "W" and the coolant flowing through the tubes of said unit heat exchangers "A" and "B". A sufficient difference is assured between the coolant temperature and the air flow temperature in the embodiment because the coolant is flowed from the rearward heat exchanger lying leeward to the forehand one standing to the windward. It is also an important feature that the times of coolant U-turn between the groups of tubes in the forehand heat exchanger "A" is more than that in the rearward heat exchanger "B". Such a structure makes less a total cross-sectional area of coolant paths in the forehand heat exchanger "A" than that in the rearward one "B" incorrespondence with a change in volume of the coolant flowing through the duplex heat exchanger employed as a condenser. It is to be noted in this connection that the coolant flowing into the rearward heat exchanger "B" is still in its gas state of a larger volume but it is gradually cooled down therein into its liquid state of a smaller volume. Therefore, the larger cross-sectional area of coolant paths in the rearward heat exchanger "B" is useful for sufficient heat transfar of the coolant in its gas state in said heat exchanger. At the same time, undesirable pressure loss is diminished to a minimum though the cross-sectional area in the forehand heat exchanger "A" is decreased corresponding to shrinkage of the coolant therein, thereby improving heat transfer efficiency of the duplex heat exchanger as a whole. The cross-sectional area in the forehand heat exchanger "A" is set at 30 to 60 % of that in the rearward heat exchanger "B". In a case wherein the area in forehand heat exchanger "A" adapted to the supercooling of coolant is below 30 %, this excessively decreased area brings about an undesirable great pressure loss in said heat exchanger "A" on one hand, and a superfluously large area of coolant paths in the rearward heat exchanger "B" adapted to condense the coolant will on the other hand undesirably decrease a flow rate of coolant to lower the heat transfer efficiency. In another case wherein the coolant path area in the forehand heat exchanger "A" is above 60 % of that in the rearward heat exchanger "B", such a small area in "B" will increase the pressure loss of coolant therein and lower the heat transfer capacity due to insufficient area of heat transfer surfaces. Therefore, it is desirable to set the cross-sectional area of coolant paths in the forehand heat exchanger "A" to be 30 to 60 %, and more preferably 35 to 50 % of that in the rearward heat exchanger "B" in order for the duplex heat exchanger to perform efficient heat transfer under a lower pressure loss.

Other parameters for better performances of the forehand and rearward heat exchangers "A" and "B" are as follows.

The aforementioned tubes 1 and 21 may preferably be 6 to 20 mm in width "Wt", 1.5 to 7 mm in height "Ht", and 1.0 mm or more in an inner height "Hp" of coolant path. The corrugated fins 2 and 22 may preferably be 6 to 16 mm in height "Hf" (that is, a distance between two adjacent tubes 1 or 21), and 1.6 to 4.0 mm in fin pitch "Fp". Reasons for such dimensions will be given below.

Tube width "Wt" less than 6 mm will make too narrow the width of the corrugated fins 2 and 22 which are interposed respectively between the two adjacent tubes 1 or 21. A larger tube width above 20 mm will cause an excessively large width of said fins 2 and 22, which in turn causes an increased resistance against air flow therethrough in addition to an overweight of the condenser. Thus, the range of 6 to 20 mm is desirable, and a range of 10 to 20 mm is more desirable.

Tube height "Ht" above 7 mm will increase the resistance of the tubes against air flow, and said height below 1.5 mm will make it difficult to obtain the inner height "Hp" of coolant path greater than 1.0 mm with a sufficient wall thickness of the tubes. The range of 1.5 to 5 mm, or more particularly a range 2 to 4 mm is preferable.

If said inner height "Hp" of coolant path were less than 1.0 mm, then the loss in coolant pressure would undesirably increase lowering the heat transfer efficiency. A range of 1.0 to 3.0 is preferable.

Fin height "Hf" less than 6 mm will bring about an increased pressure loss of air flow penetrating through the fins, though fin height of 16 mm or more will reduce the number of mounted fins, reducing the "fin effect" and making worse the heat transfer performance. Therefore, fin height is selected from the aforementioned range of 6 to 16 mm, or more preferably from a range of 8 to 12 mm is selected.

As for fin pitch "Fp", the air flow pressure loss increases with its value below 1.6 mm whereas heat transfer performance becomes worse with its value above 4.0 mm. The most preferable range is from 2.0 to 3.6 mm.

As described above, the most adequate dimensions are selected as to the shapes of tubes 1 and 21 and the corrugated fins 2 and 22 which give important influences on the performance of the condenser. Selection of the dimensions of tube width, tube height, inner height of coolant path, fin height and fin pitch respectively from the ranges referred to above will provide the condenser which can be operated efficiently in an optimal manner wherein a good balance is realized between the pressure loss of coolant or airflow and the heat transfer characteristics, without being accompanied by any significant increase in the weight of condenser.

Second Embodiment

A second embodiment of the invention is illustrated in Fig. 10 and 11. The same reference numerals are allotted to the same parts or elements as those in the first embodiment, and description of such parts or elements is not repeated here. The second embodiment also is applied to a condenser and comprises a forehand heat exchanger "A" connected in series to a rearward heat exchanger "B". However, coolant flows through the forehand heat exchanger "A" from its upper region toward its lower region, contrary to the flow direction in the first embodiment. A lower chamber of a left-hand header 23 of the rearward heat exchanger "B" is brought by a joint pipe 60 into fluid communication with an upper chamber of a left-hand header 3 of the forehand heat exchanger "A". Consequently, coolant which enters the rearward heat heat exchanger "B" through a coolant inlet pipe 40 will make then a U-turn before it arrives at the lower chamber of said lower chamber of the left-hand header 23 and hence moves into said upper chamber of the left-hand header 3. The coolant descends zigzag to the lower region of the forehand heat exchanger "A" while making U-turns therein, and is discharged from said heat exchanger "A".

It is an important feature peculiar to the second embodiment that positions of partition plates 9, 10 and 29 are determined such that cross-sectional areas of tube groups arranged in a serpentine pattern throughout the forehand and rearward heat exchangers "A" and "B" decrease gradually and stepwise in a direction from inlet toward outlet of coolant. Said cross-sectional areas depend on the numbers of tubes in the groups thereof. Said partition plates 9, 10 and 29 are disposed so as to form the tube groups consisting of thirteen, ten, eight, six, five and four pipes in said direction. Such a gradual change in the number of tubes in the second embodiment corresponds more exactly to the change in the specific volume of coolant, thus providing more efficient condenser.

Fin pitch "Fp_A" in the forehand heat exchanger "A" is greater than fin pitch "Fp_B" in the rearward heat exchanger so that. This means that heat transfer surface per unit area seen in the direction of air flow is narrower in the forehand heat exchanger "A" than that in the rearward heat exchanger "B". Such different fin pitches "Fp_A" and "Fp_B" is effective to improve heat transfer efficiency, without increasing air flow pressure loss. It is recommended to adopt a value of 1.07 to 1.8 as a ratio of "Fp_A" to "Fp_B". A ratio less than 1.07 will result in a greater pressure loss of air flow and a lower efficiency of heat radiation. A ratio higher than 1.8 however will likewise result in an insufficient heat radiation efficiency, apart from an enough decrease in the pressure loss. A narrower range of the ratio from 1.1 to 1.6 is more preferable. Even in a case wherein the core sizes of the forehand and rearward heat exchangers are the same, the ratio should fall within the range of 1.07 to 1.8, and more desirably within the range of 1.1 to 1.6 for the same reason as mentioned above.

Third Embodiment

A third embodiment also is applied to a condenser shown in Figs. 12 and 13 and comprising a forehand heat exchanger "A" which is connected in series to a rearward heat exchanger "B" and is of the same size as the former. Headers, tubes and corrugated fins in the third embodiment are respectively of the same structures and are given the same reference numerals as those in the first embodiment, so that description thereof is not repeated.

In this embodiment, joint blocks are utilized to connect the heat exchangers "A" and "B" into fluid communication with each other. A male joint block 80 is welded or otherwise attached to a lowermost portion of a left-hand header 3 in the forehand heat exchanger "A". The male joint block 80 has a lug 81 protruding from an inner side, and a coolant passage 82 is formed through the lug 81 so as to be in fluid connection with the left-hand header 3. On the other hand, a female joint block 90 is fixed to a lowermost portion of the left-hand header 23 in the rearward heat exchanger "B". An aperture 91 is formed at inner side of and through the female joint block so as to be likewise in fluid connection with the left-hand header 23. To combine the male joint block with the female joint block, the lug 81 is engaged with the aperture 91, the inner sides of the blocks thereby being brought into close contact with each other. Then a bolt 100 is inserted through a hole 83 of the male block 80 into an internally-threaded hole 92 of the female block 90. In this way, the coolant paths of the forehand and rearward heat exchangers "A" and "B" are connected in series. Further, fixed to an uppermost portion of the rearward heat exchanger "B" is an inlet block 110 having a hole. A pipe attaching block 120 which has a lug 121 and an attached inlet pipe 130 is mounted to the inlet block 110, by engaging the lug 121 with the hole of the inlet block, and is fastened thereto by means of a bolt 140. Similarly, an outlet block 150 having a hole 151 is fixed to an uppermost portion of the left-hand header 3 in the forehand heat exchanger "A". A pipe attaching block 160 which has a lug 161 and an attached outlet pipe 170 is mounted to the outlet block 150, by engaging the lug 161 with the hole 151 of the outlet block, and is fastened thereto by means of a bolt 180.

Such a connection system as using the joint blocks and other blocks as in this embodiment is advantageous in that the forehand and rearward unit heat exchangers can be separately manufactured and inspected for coolant leakage. Assembly of the two unit heat exchangers into a duplex heat exchanger is easily carried out at a final step in the manufacture process whereby workshop operations and productivity are improved.

Fourth Embodiment

Figs. 14 and 15 show a fourth embodiment also applied to a condenser. This embodiment is different from the first embodiment in that a forehand and a rearward heat exchangers "A" and "B" of the same shape and the same dimension are connected parallel with each other. A bifurcate inlet pipe 190 for coolant is attached to upper portions of left-hand headers 3 and 23 of the respective heat exchangers "A" and "B". A bifurcate outlet pipe 200 is connected to bottoms of said left-hand headers 3 and 23. A partition plate 9 is secured in the left-hand header 3 at its middle portion, for the forehand heat exchanger, while two partition plates 29 are secured in the left-hand header 23 respectively at its upper and lower portions. Further, another partition plate (not shown) is similarly secured in a right-hand header 24 at its middle portion. Those partition plates cause coolant which enter the heat exchangers "A" and "B" through the bifurcate inlet pipe 190 to make a U-turn within the forehand heat exchanger "A" and to make three U-turns within the rearward heat exchanger "B". Finally, the coolant is collected into lower chambers of the left-hand headers 3 and 23 before it leaves this duplex heat exchanger through the bifurcate outlet pipe 200. The coolant is caused to make U-turn more times in the rearward heat exchanger "B" than in the forehand heat exchanger "A" because a total length of coolant flow paths is to be greater for the rearward heat exchanger "B" which lies leeward and is thus of a lesser efficiency of heat transfer. A good balance between the heat transfer efficiency of the two heat exchangers is assured in this manner in the fourth embodiment.

Claims

1. A condenser comprising unit heat exchangers (A and B) which respectively comprise a plurality of flat tubes (1, 21) arranged parallel with each other and fins (2, 22) each interposed between adjacent tubes with opposite ends of each tube being connected to a pair of headers (3, 4 and 23, 24) in fluid connection therewith, the unit heat exchangers (A, B) being closely juxtaposed to each other fore and aft in a direction of air flow so that coolant circuits of the unit heat exchangers are connected in series, each unit heat exchanger comprising partitioning means (9, 29) fixed inside at least one of the headers so as to divide an internal space thereof into at least two sections in a manner such that the coolant paths in the unit heat exchangers are meanders which make zigzag turns caused by the partitioning means, the coolant flowing from one unit heat exchanger lying on the leeward side, into the other unit heat exchanger standing to the windward side of the air flow, the sections including an inlet section formed in the unit heat exchanger lying on the leeward side and an outlet section formed in one of the other unit heat exchangers standing to the windward side, a coolant inlet pipe being connected to the inlet section, whilst a coolant outlet pipe is connected to the outlet section, the total cross-sectional area of the coolant paths formed through the tubes connected to the outlet section being smaller than that formed through the other tubes connected to the inlet section, wherein heat exchange capacity of the leeward unit heat exchanger is greater than that of the windward unit heat exchanger.

2. A condenser according to claim 1, characterized in that the fin pitch of the leeward unit heat exchanger is smaller than that of the windward unit heat exchanger.

3. A condenser according to claim 1, characterized in that the number of U-turns within the coolant circuit in the windward unit heat exchanger is greater than that in the leeward unit heat exchanger.

4. A condenser according to claim 1, characterized in that the total cross-sectional area of the coolant paths connected to the outlet section is about 30% to 60% of that connected to the inlet section.

5. A condenser according to claim 1, characterized in that the cross-sectional area of the coolant flow paths gradually decreases from the inlet section towards the outlet section.

6. A condenser according to claim 1, characterized in that each unit heat exchanger is designed such that the tube width is 6 - 20 mm, the tube height is 1.5 - 7.0 mm, the height of coolant path formed through the tube is 1.0 mm or more, the fin height is 6 - 16 mm and the fin pitch is 1.6 - 4.0 mm.

7. A condenser comprising two unit heat exchangers which respectively comprise a plurality of flat tubes arranged parallel with each other and fins each interposed between one of

such tubes and the next, with opposite ends of each tube being respectively connected to a pair of hollow headers in fluid connection therewith, the unit heat exchangers being closely juxtaposed to each other fore and aft in the direction of air flow wherein coolant circuits of the unit heat exchangers are connected in parallel with each other so that the coolant flows simultaneously through the unit heat exchangers and the coolant circuit of each heat exchanger is formed in a meandering pattern to make U-turns, by means of at least one partition means secured in at least one header and wherein the number of U-turns of the coolant within the leeward unit heat exchanger is greater than that within the windward unit heat exchanger, so that a total length of the flow paths in the former is greater than that in the latter.

Patentansprüche

1. Duplex-Wärmetauscher, bestehend aus Wärmetauschern (A und B) mit einem Rohrregister aus parallelen flachen, an ihren Enden mit einem Paar von Sammelkanälen (3, 4 und 23, 24) verbundenen Rohren (1, 21) und Kühlrippen zwischen benachbarten Rohren, wobei die Kühlräume der in geringem Abstand nebeneinander in einem Luftstrom liegenden Wärmetauschern (A, B) in Reihe hintereinander geschaltet und in den Sammelkanälen mindestens eines der Wärmetauscher Trennwände (9, 29) vorgesehen sind zur Zwecke der Teilung des Kühlraumes des Wärmetauschers in mindestens zwei sich mäanderartig durch die Trennwände gebildeten zick-zack-förmigen Umlenkungen hintereinander liegende Kühlmittelkanäle, durch die das Kühlmittel im Gegenstrom zum Luftstrom durch die hintereinandergeschalteten Wärmetauscher geleitet wird, von denen der auf der Luftabströmseite (leewärts) gelegene Wärmetauscher einen mit einer Zuflußleitung verbundenen Kühlmitteleinlaß und der auf der Luftzuströmseite (luvwärts) gelegene Wärmetauscher einen mit einer Kühlmittelabflußleitung verbundenen Kühlmittelauslaß aufweist und die Querschnitte der von den verbundenen Rohren gebildeten Kühlmittelkanäle auf der Abflußseite kleiner als auf der Zuflußseite sind, so daß die Wärmetauscherleistung des auf der Luftabströmseite gelegenen Wärmetauschers größer als die des auf der Luftzuströmseite gelegenen Wärmetauschers ist.

2. Duplex-Wärmetauscher nach Anspruch 1, dadurch gekennzeichnet, daß der Abstand der Rippen des leewärts gelegenen Wärmetauschers kleiner als der Abstand der Rippen des luvwärts gelegenen Wärmetauschers ist.

3. Duplex-Wärmetauscher nach Anspruch 1, dadurch gekennzeichnet, daß die Zahl der Umlenkungen im Kühlmittelkreis des luvwärtigen Wärmetauschers größer als im leewärtigen Wärmetauscher.

4. Duplex-Wärmetauscher nach Anspruch 1, dadurch gekennzeichnet, daß der Querschnitt der einzelnen Kühlmittelkanäle auf der Seite des Kühlmittelabflusses etwa 30% bis 60% des Querschnittes der Kühlmittelkanäle auf der Seite Kühlmittelzuflusses beträgt.

5. Duplex-Wärmetauscher nach Anspruch 1, dadurch gekennzeichnet, daß sich der Querschnitt der Kühlmittelkanäle von der Einlaufseite zur Abflußseite kontinuierlich verringert.

6. Duplex-Wärmetauscher nach Anspruch 1, dadurch gekennzeichnet, daß die Breite der Rohre 6 - 20 mm, ihre Höhe 1,5 - 7.0 mm, die Höhe der durch die Rohre gebildeten Kühlmittelkanäle 1,0 mm oder mehr, die Rippenhöhe 6 - 16 mm und der Rippenabstand 1,6 - 4,0 mm betragen.

7. Duplex-Wärmetauscher aus zwei Wärmetauschern mit je einem Rohrregister aus parallelen flachen, an ihren Enden mit einem Paar von Sammelkanälen verbundenen Rohren und Kühlrippen zwischen benachbarten Rohren, wobei die Enden jedes der Rohre mit einem Paar von Sammelkanälen flüssigkeitsleitend verbunden sind und die Kühlräume der in geringem Abstand parallel nebeneinander in einem Luftstrom liegenden Wärmetauschern in Reihe hintereinander geschaltet sind, so daß das Kühlmittel gleichzeitig durch die Kühlmittelkanäle des Wärmetauschers fließt, die mäanderartig in U-förmigen Umlenkungen durch den Wärmetauscher geführt sind mit Hilfe von mindestens einer Trennwand in mindestens einem der Sammelkanäle, wobei die Zahl der U-förmigen Umlenkungen des Kühlmittels im leewärtigen, dh. auf der Luftabströmseite gelegenen Wärmetauscher größer als die im luvwärtigen Wärmetauscher ist, so daß die Gesamtlänge der Kühlmittelkanäle im ersteren größer als im letzteren ist.

Revendications

1. Condenseur comprenant des échangeurs de chaleur unitaires (A et B) qui comprennent,

respectivement, une pluralité de tubes plats (1, 21) disposés parallèles l'un à l'autre et des ailettes (2, 22) interposées, chacune, entre des tubes adjacents, les extrémités opposées de chaque tube étant connectées à une paire de collecteurs (3, 4 et 23, 24), en connexion de fluide avec ceux-ci, les échangeurs de chaleur unitaires (A, B) étant étroitement juxtapposés l'un à l'autre, de bout en bout, dans le sens de la circulation de l'air, de sorte que les circuits de réfrigérant des échangeurs de chaleur unitaires sont connectés en série, chaque échangeur de chaleur unitaire comprenant un moyen de cloisonnement (9, 29) fixé à l'intérieur d'au moins l'un des collecteurs de manière à diviser l'espace intérieur de celui-ci en au moins deux sections de telle manière que les trajets de réfrigérant dans les échangeurs de chaleur unitaires sont des méandres qui créent des retours en zigzag provoqués par le moyen de cloisonnement, le réfrigérant circulant d'un échangeur de chaleur unitaire situé du côté sous le vent dans l'autre échangeur de chaleur unitaire se trouvant du côté au vent de la circulation de l'air, les sections comportant une section d'entrée formée dans l'échangeur de chaleur unitaire situé du côté sous le vent et une section de sortie dans l'un des autres échangeurs de chaleur unitaires se trouvant du côté au vent, un tuyau d'admission de réfrigérant étant connecté à la section d'entrée tandis qu'un tuyau de sortie de réfrigérant est connecté à la section de sortie, la surface de section transversale totale des trajets de réfrigérant formés dans les tubes connectés à la section de sortie étant plus petite que celle formée dans les autres tubes connectés à la section d'entrée, dans lequel la capacité d'échange de chaleur de l'échangeur de chaleur unitaire sous le vent est supérieure à celle de l'échangeur de chaleur unitaire au vent.

2. Condenseur suivant la revendication 1, caractérisé en ce que le pas d'ailettes de l'échangeur de chaleur unitaire sous le vent est plus petit que celui de l'échangeur de chaleur unitaire au vent.

3. Condenseur suivant la revendication 1, caractérisé en ce que le nombre de retours en "U" dans le circuit de réfrigérant dans l'échangeur de chaleur unitaire au vent est supérieur à celui de l'échangeur de chaleur unitaire sous le vent.

4. Condenseur suivant la revendication 1, caractérisé en ce que la surface de section transversale totale des trajets de réfrigérant connectés à la section de sortie est d'environ 30% à 60% de celle connectée à la section d'entrée.

5. Condenseur suivant la revendication 1, caractérisé en ce que la surface de section transversale des trajets de circulation de réfrigérant diminue progressivement de la section d'entrée vers la section de sortie.

6. Condenseur suivant la revendication 1, caractérisé en ce que chaque échangeur de chaleur unitaire est conçu de telle manière que la largeur de tube est de 6 à 20 mm, que la hauteur de tube est de 1,5 à 7,0 mm, que la hauteur de trajet de réfrigérant formé dans le tube est de 1,0 mm ou plus, que la hauteur d'ailette est de 6 à 16 mm et que le pas d'ailettes est de 1,6 à 4,0 mm.

7. Condenseur comprenant deux échangeurs de chaleur unitaires qui comprennent, respectivement, une pluralité de tubes plats disposés parallèles l'un à l'autre et des ailettes interposées, chacune, entre l'un de ces tubes et le suivant, les extrémités opposées de chaque tube étant connectées, respectivement, à une paire de collecteurs creux en connexion de fluide avec ceux-ci, les échangeurs de chaleur unitaires étant étroitement juxtapposés l'un à l'autre, de bout en bout, dans le sens de la circulation de l'air, dans lequel les circuits de réfrigérant des échangeurs de chaleur unitaires sont connectés en parallèle entre eux, de sorte que le réfrigérant circule simultanément dans les échangeurs de chaleur unitaires, et le circuit de réfrigérant de chaque échangeur de chaleur est formé en forme de méandre pour créer des retours en "U" à l'aide d'au moins un moyen de cloisonnement fixé dans au moins un collecteur et dans lequel le nombre de retours en "U" du réfrigérant dans l'échangeur de chaleur unitaire sous le vent est supérieur à celui dans l'échangeur de chaleur unitaire au vent, de sorte que la longueur totale des trajets de circulation dans le premier est supérieure à celle dans le dernier.

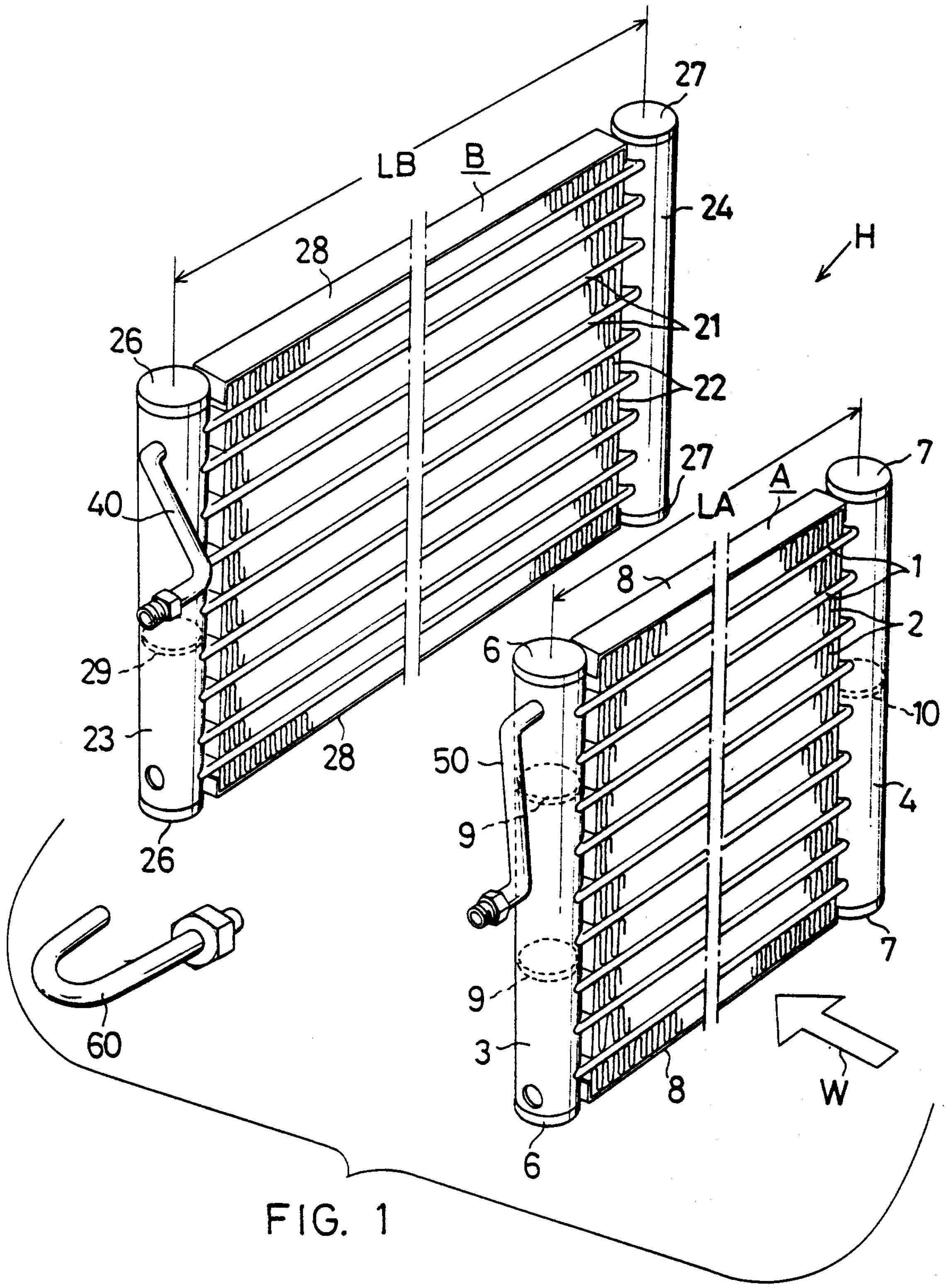

27
LB
B
24
H
28
21
26
22
27
7
40
LA
A
8
1
2
6
10
29
23
28
50
4
26
9
9
7
60
3
8
W
6

FIG. 1

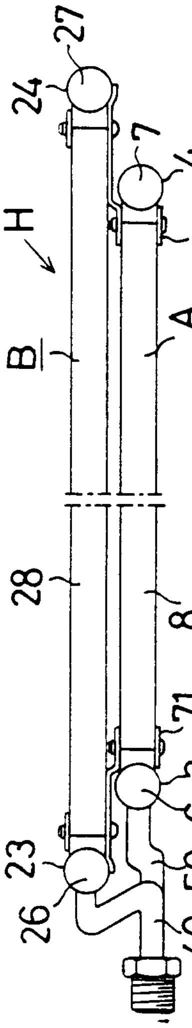


FIG. 3

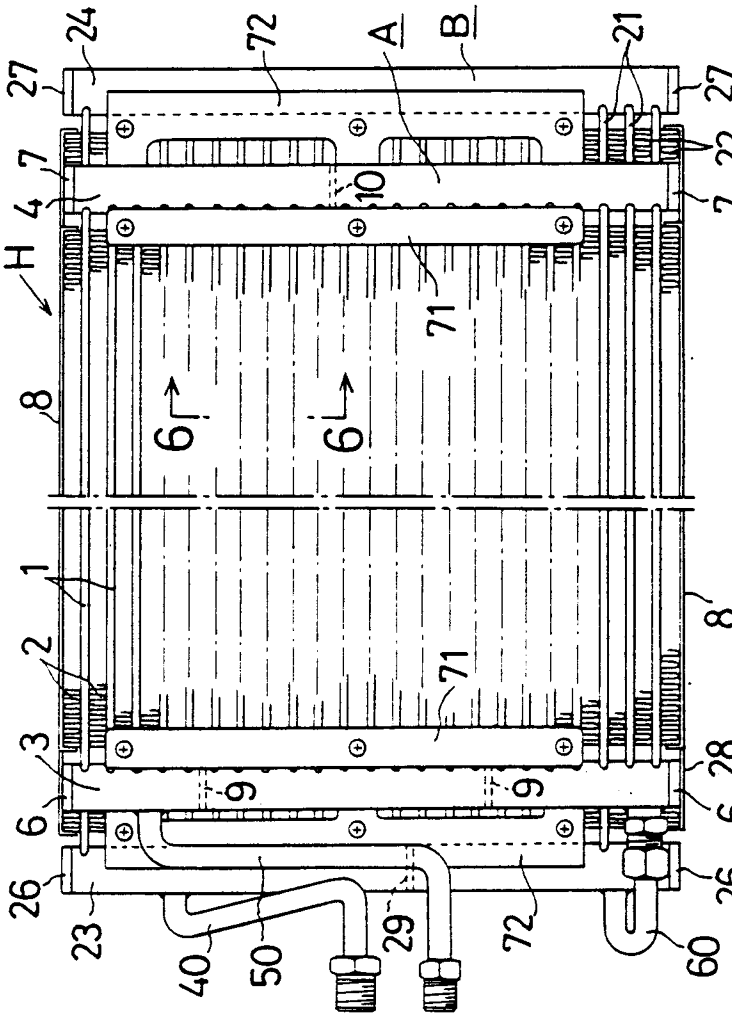


FIG. 2

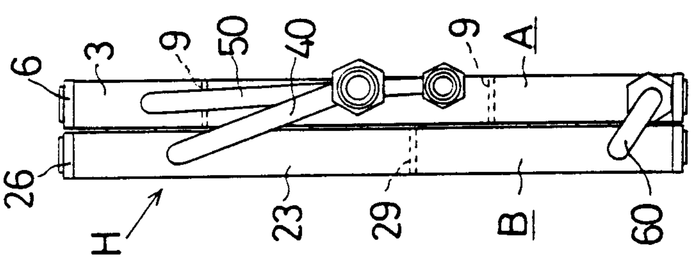


FIG. 4

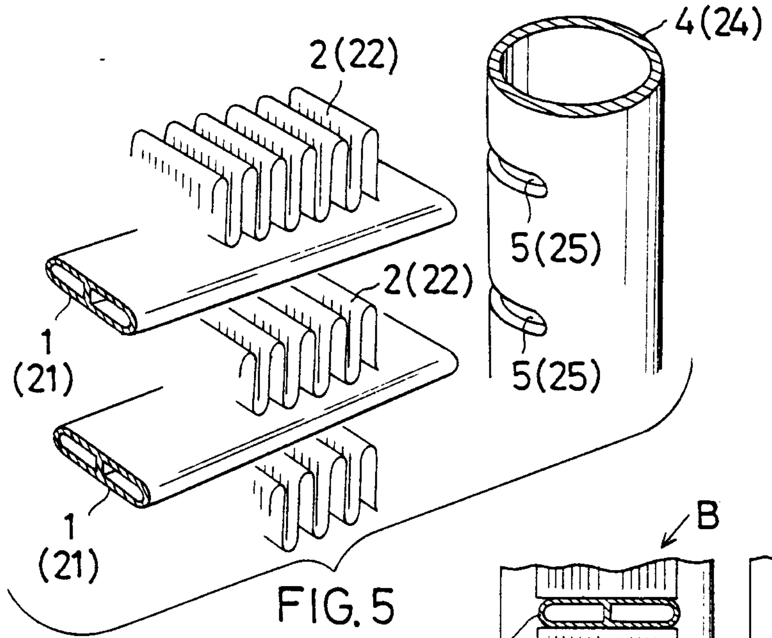


FIG. 5

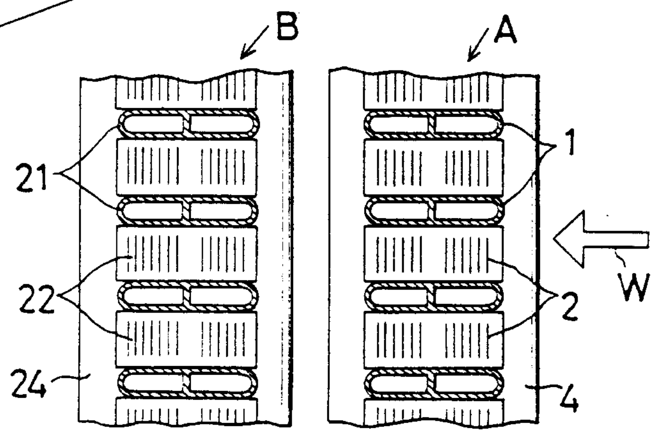


FIG. 6

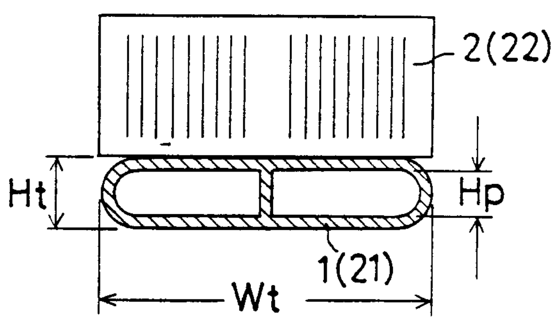


FIG. 7

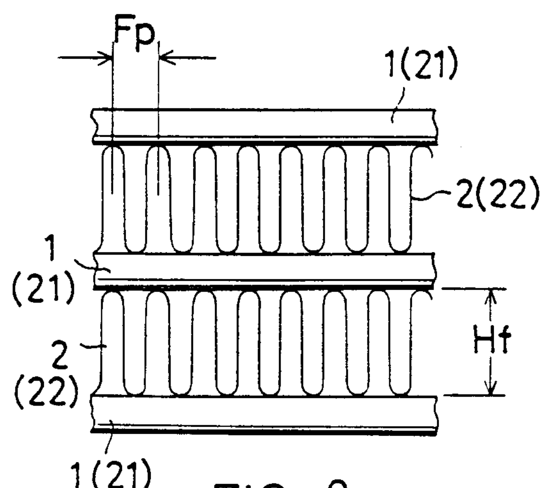


FIG. 8

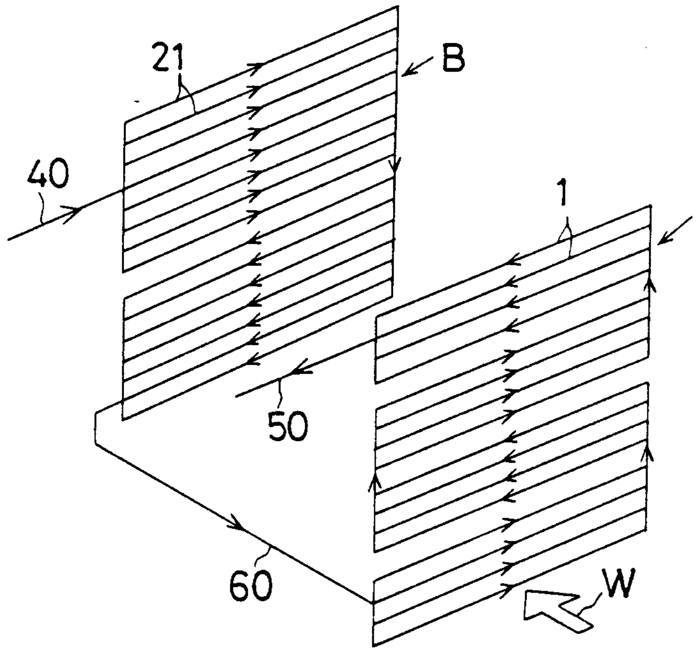

21
B
40
1
A
50
60
W

FIG. 9

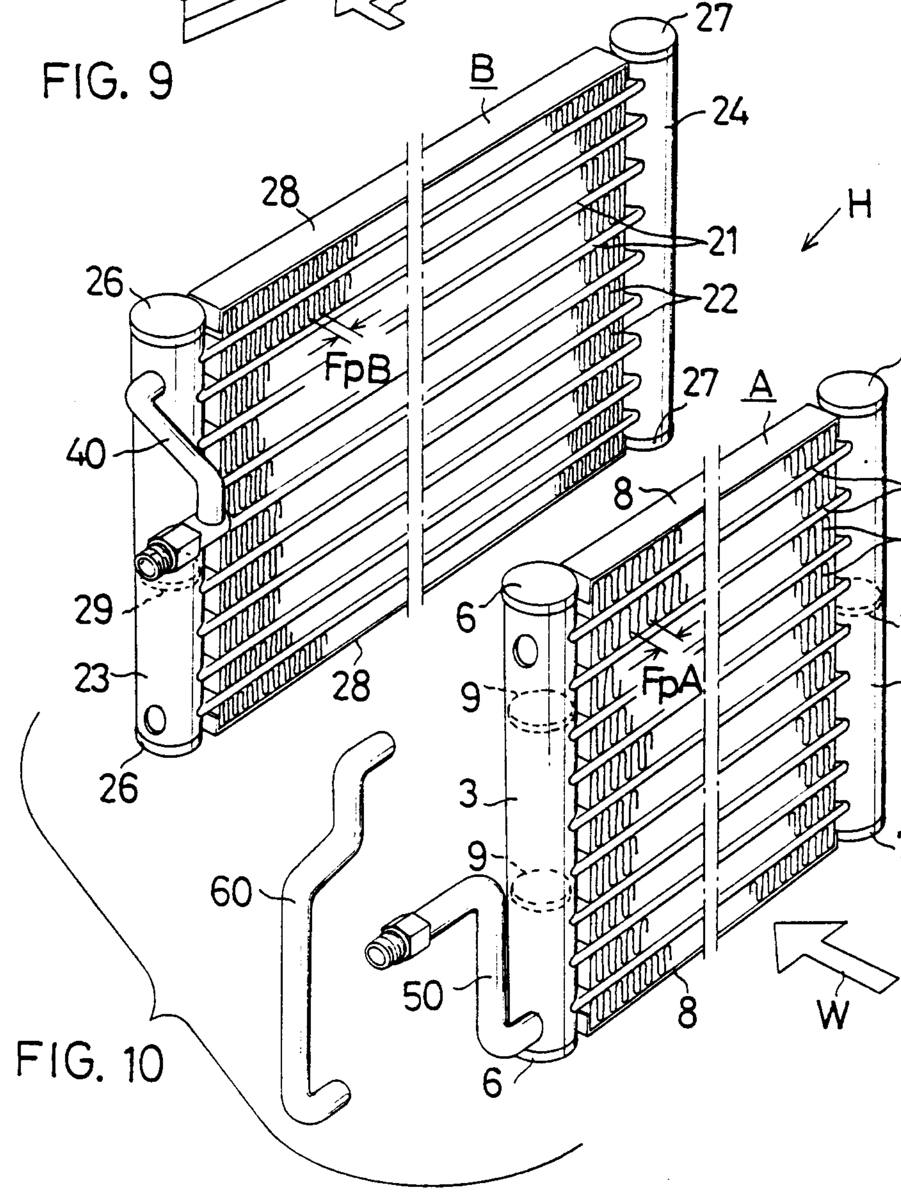

27
B
24
28
H
21
26
22
FpB
27
A
7
40
8
1
2
29
6
10
23
28
9
FpA
4
26
3
9
7
60
50
8
W
6

FIG. 10

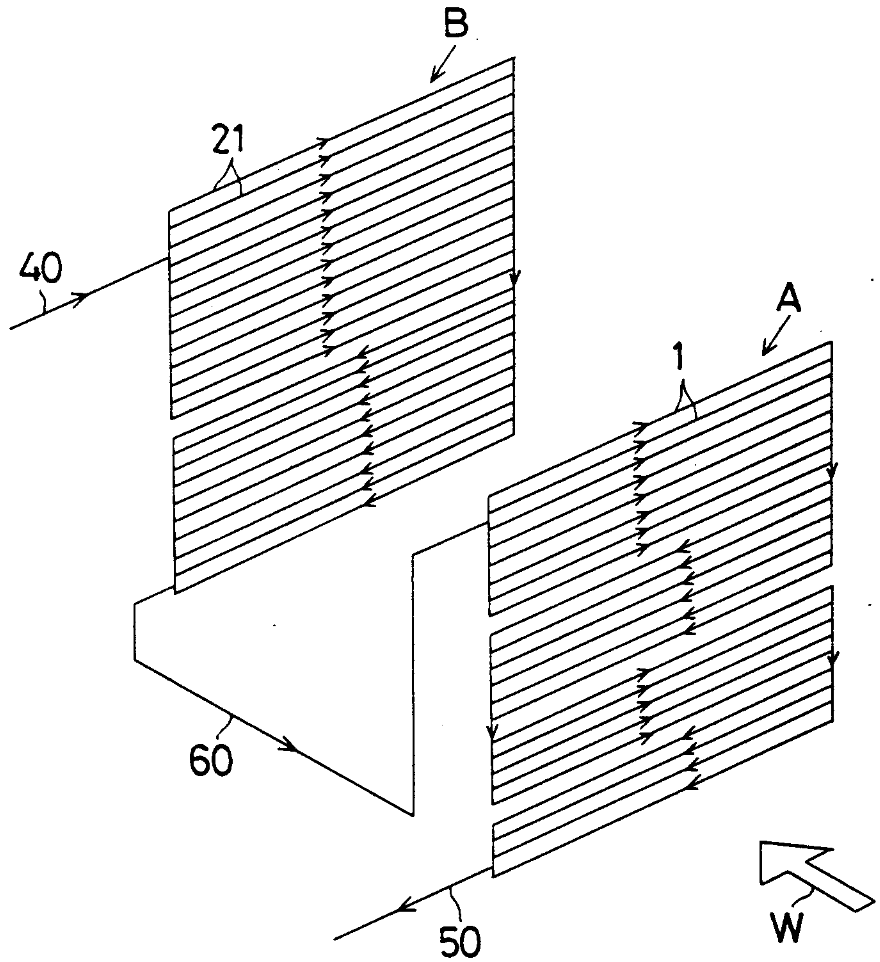


FIG. 11

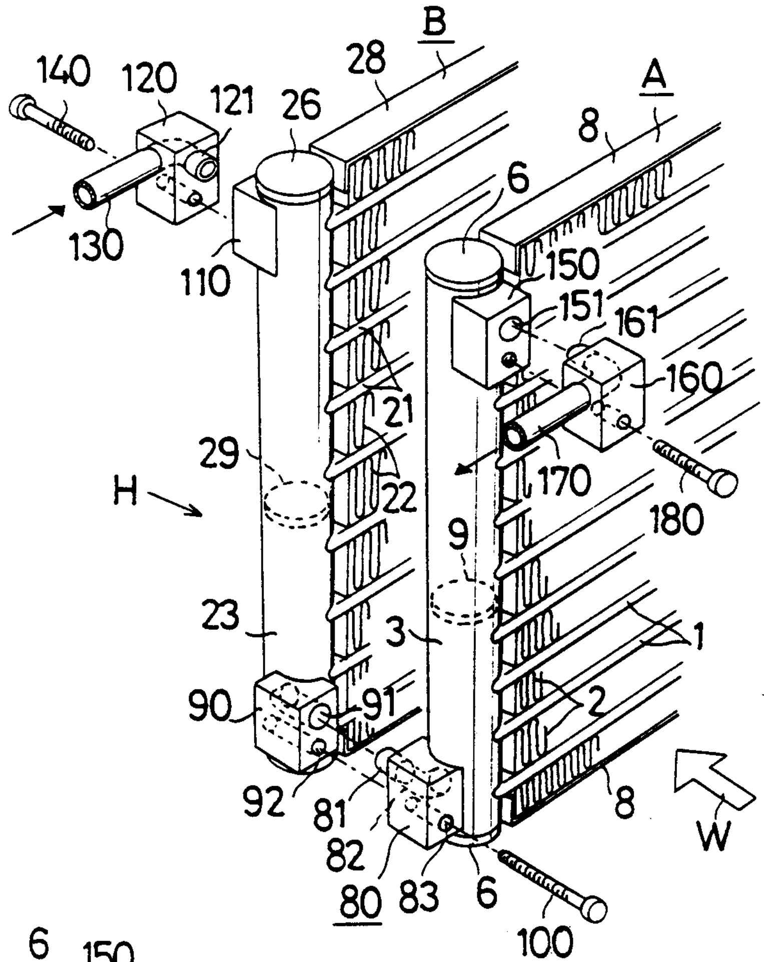
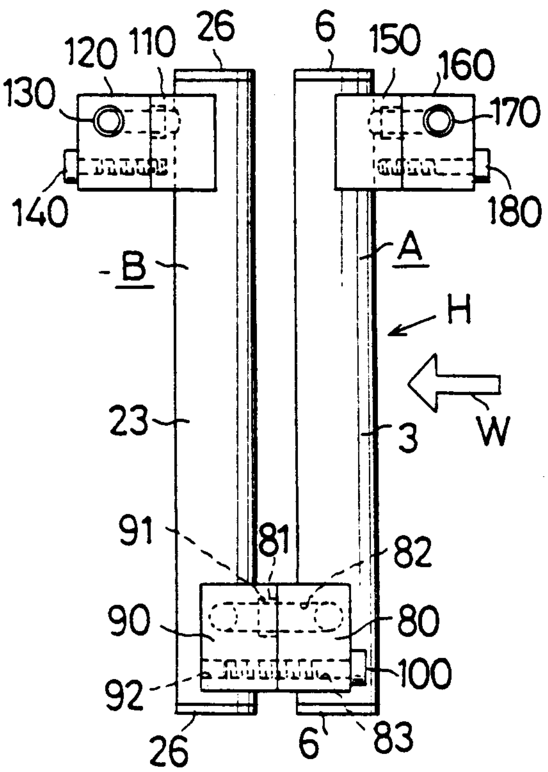

140
120
121
26
28
B
A
8
6
130
110
150
151
161
160
21
29
22
H
170
180
9
23
3
1
90
91
2
92
81
82
8
W
80
83
6
100

FIG. 12


120
110
26
6
150
160
130
170
140
180
B
A
H
W
23
3
91
81
82
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80
92
100
26
6
83

FIG. 13

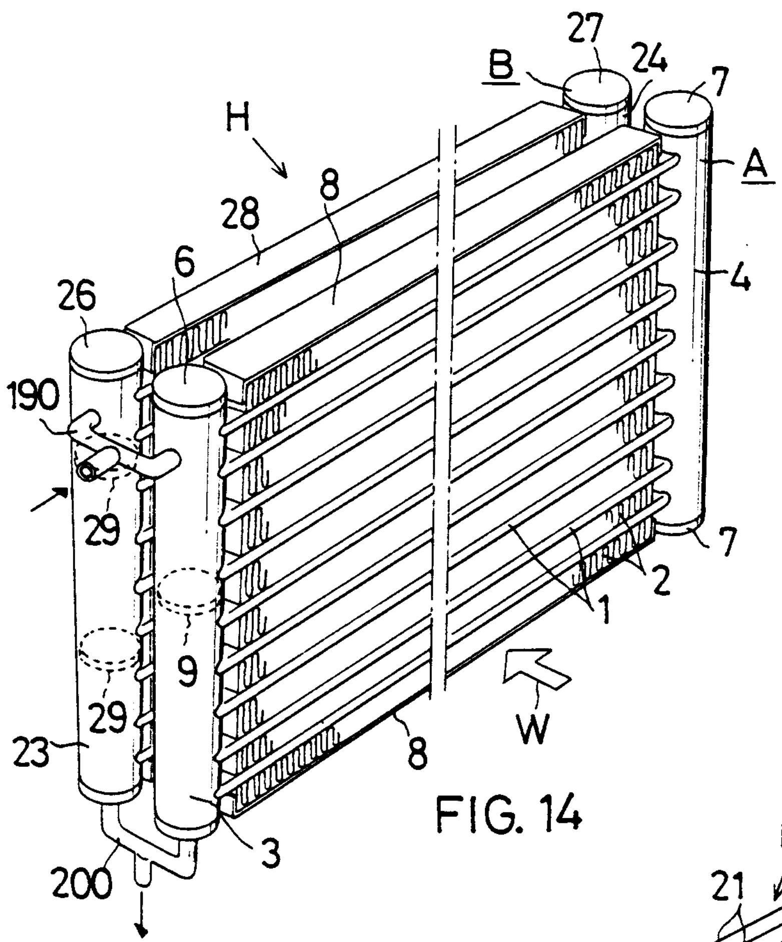

FIG. 14

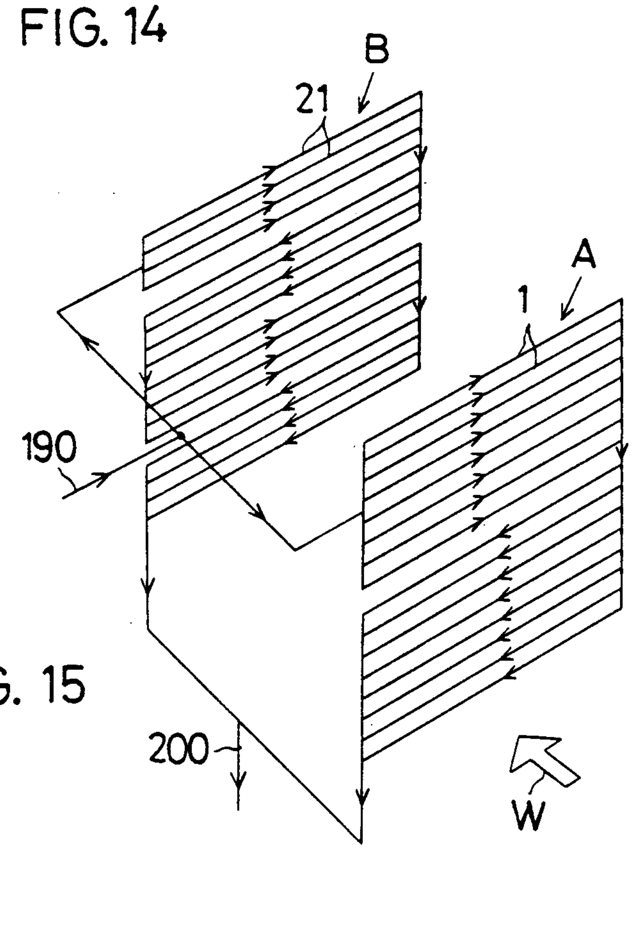

FIG. 15