Office de la Propriété Intellectuelle du Canada
Un organisme d'Industrie Canada

Canadian Intellectual Property Office
An agency of Industry Canada

CA 2023499 C 2002/10/29

(11)(21) 2 023 499

(12) BREVET CANADIEN
CANADIAN PATENT

(13) C

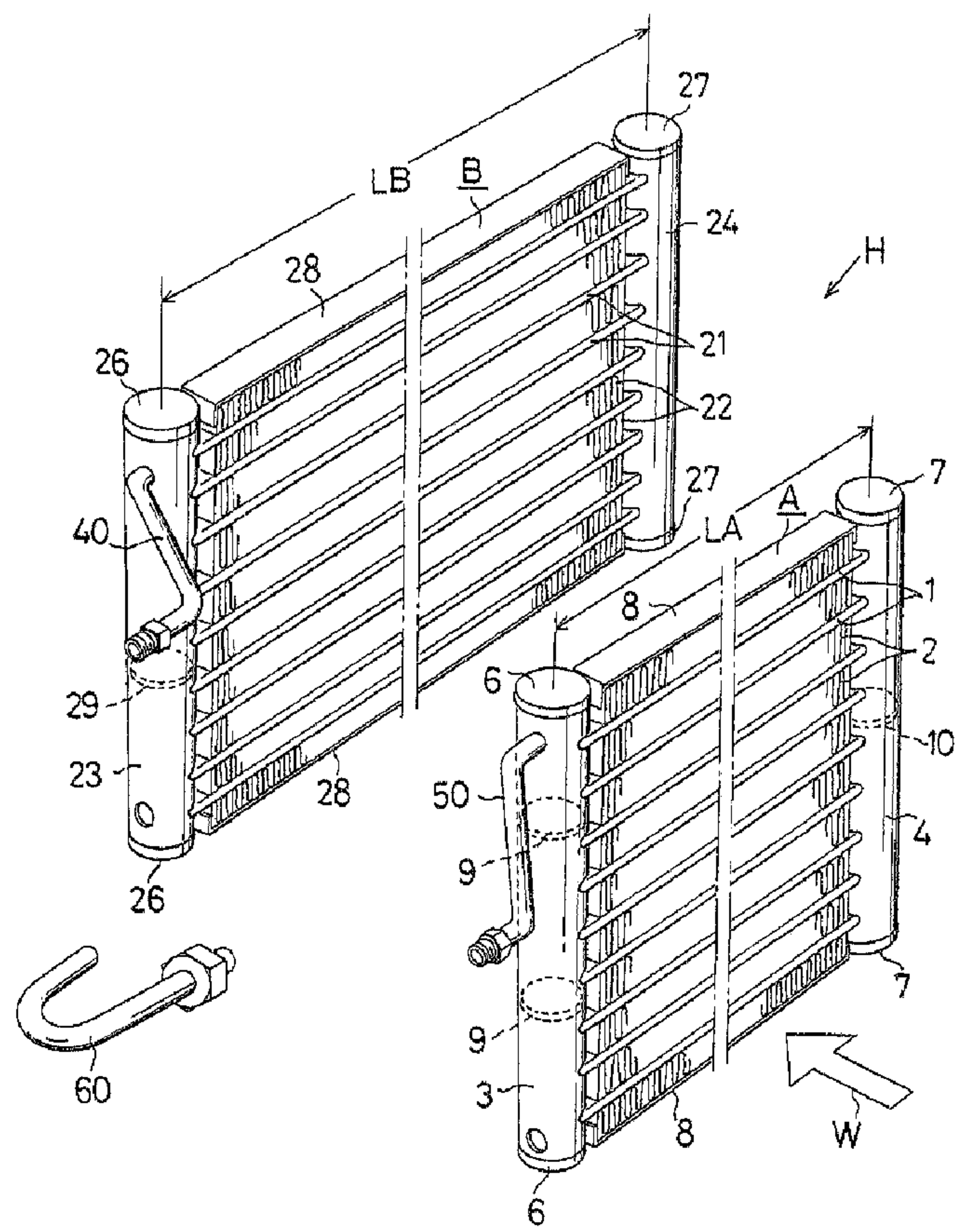
(22) Date de dépôt/Filing Date: 1990/08/17
(41) Mise à la disp. pub./Open to Public Insp.: 1991/02/24
(45) Date de délivrance/Issue Date: 2002/10/29
(30) Priorité/Priority: 1989/08/23 (1-217959) JP

(51) Cl.Int.^{5}/Int.Cl.^{5} F28F 7/00
(72) Inventeurs/Inventors:
Sasaki, Hironaka, JP;
Watanabe, Hirohiko, JP;
Tategami, Tetsuya, JP

(73) Propriétaire/Owner:
SHOWA DENKO K.K., JP

(74) Agent: MARKS & CLERK

(54) Titre : ECHANGEURS DE CHALEUR EN TANDEM
(54) Title: DUPLEX HEAT EXCHANGER



(57) Abrégé/Abstract:
A duplex heat exchanger comprises unit heat exchangers which have a plurality of tubes arranged parallel with each other and comprise fins each interposed between two adjacent ones of such tubes, opposite ends of each tube being connected to a pair of headers in fluid connection therewith. The unit heat exchangers are closely juxtaposed to each other fore and aft in a direction of air flow. Coolant circuits of said unit heat exchangers are connected either in series or in parallel with each other.

ABSTRACT OF THE DISCLOSURE

A duplex heat exchanger comprises unit heat exchangers which have a plurality of tubes arranged parallel with each other and comprise fins each interposed between two adjacent ones of such tubes, opposite ends of each tube being connected to a pair of headers in fluid connection therewith. The unit heat exchangers are closely juxtaposed to each other fore and aft in a direction of air flow. Coolant circuits of said unit heat exchangers are connected either in series or in parallel with each other.

DUPLEX HEAT EXCHANGER

The invention relates to a heat exchanger, and more particularly to a duplex heat exchanger comprising two unit heat exchangers and adapted for use as the condensers or evaporators in car coolers, or for use as the oil coolers for automobiles or the likes.

The so-called multi-flow type heat exchanger has attracted public attention in the uses mentioned above. This heat exchanger has a structure, as disclosed in Japanese Patent Publication Kokai 63-34466, 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 said 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, said 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.

In a case where a higher capacity of heat exchange is needed for the multi-flow type heat exchanger, vertical and/or horizontal dimensions thereof may be restricted by a given space for installation of said heat exchanger. In detail, length and the number of the tubes are generally delimited by the spatial condition. It has thus been necessary for the width of said tubes, i.e., the depth of said heat exchanger, to be increased to meet the required greater capacity.

However, a larger width of the tubes inevitably causes an outer diameter of the headers to be increased resulting in decrease of the tubes' length effective to heat transfer. This problem has been a bottleneck in increasing the heat transfer capacity to a satisfactory degree.

An object of the invention is therefore to provide a heat exchanger which is adapted to increase the heat transfer capacity thereof.

Another object is provide a heat exchanger whose heat transfer capacity can be increased without necessitating an excessively wide space.

A further object of the invention is to provide a heat exchanger which is so beneficially constructed that an optimal design is readily chosen for a higher efficiency of heat transfer and also for a lower loss of fluid pressure.

The invention aims to provide a duplex heat exchanger which, in order to achieve the abovementioned objects, comprises unit heat exchangers of the multi-flow type wherein these unit heat exchangers are closely juxtaposed to each other fore and aft in a direction of air flow.

In more detail, the duplex heat exchanger in accordance with the present invention is characterized in that the unit heat exchangers which respectively comprise a plurality of tubes arranged parallel with each other and comprise fins each interposed between one of such tubes and the next, opposite ends of each tube being connected to a pair of headers in fluid connection therewith. Coolant paths of the unit heat exchangers are connected either in series or in parallel with each other. When the coolant paths are connected in series with each other the coolant flows from one of the unit heat exchangers lying on the leeward into another unit heat exchanger standing to the windward of the air flow. When the coolant paths are connected in parallel with each other the coolant flows simultaneously through the unit heat exchangers. The coolant paths in each of the unit heat exchangers are meanders which make zigzag turns defined by partition plates fixed inside of the headers. In such a configuration, heat transfer between coolant and air flow takes place in the forehand unit heat exchanger as well as in the rearward one so that an amount of transferred heat increases as a whole.

Other objects and advantages of the invention will become apparent from the description given hereinunder referring 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;

Fig. 16 is a cross section, taken in a horizontal direction, of a still further duplex heat exchanger according to a fifth embodiment;

Fig. 17 is a cross section taken along a line 17 - 17 in Fig. 16;

Fig. 18 is a perspective view of a yet further duplex heat exchanger according to a sixth embodiment;

Fig. 19 is a perspective view illustrating an essential part of the duplex heat exchanger shown in Fig. 18; and

Fig. 20 is a schematic plan view illustrating still another duplex heat exchanger according to a seventh embodiment of the invention.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

First Embodiment

Figs. 1 to 9 shows an embodiment in which the invention is applied to an evaporator 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" in correspondence 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 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.

Fifth Embodiment

Figs. 16 and 17 show a fifth embodiment in which wide corrugated fins 210 extend from a forehand heat exchanger "A" to a rearward heat exchanger "B" so as to span them. This structure provides direct connection between cores of said heat exchangers "A" and "B", thereby improving their heat transfer efficiencies as a whole. Mechanical strength of connection is also enhanced so that only one of the heat exchangers need be secured to a body of automobile. This reduces the number of parts necessary for mounting this duplex heat exchanger to automobiles or the likes, and thereby improves productivity of the heat exchanger.

Sixth Embodiment

Figs. 18 and 19 show a sixth embodiment which is applied to an evaporator for car coolers. In the sixth embodiment which is suited to reduce coolant pressure loss in evaporators, tubes of a forehand heat exchanger "A" as well as tubes of a rearward heat exchanger "B" are vertical and parallelly arranged in a right-to-left direction. Upper headers 3 and 23 and lower headers 4 and 24 disposed horizontally. A bifurcate inlet pipe 220 for coolant is connected to left ends of the upper headers 3 and 23. Likewise, a bifurcate outlet pipe 230 is connected to right ends of the lower headers 4 and 24, coolant paths of the two heat exchangers "A" and "B" thereby running parallel with each other. Coolant flows into the heat exchangers "A" and "B" through the inlet pipe 220, descends to the lower headers 4 and 24, and then flows out of the heat exchangers through the outlet pipe 230. As shown in Fig. 19, fin pitch "Fp_B" of corrugated fins 22 in the rearward heat exchanger "B" is made greater than that "Fp_A" in the forehand heat exchanger "A". Such a greater fin pitch "Fp_B" in the rearward heat exchanger prevents the so-called water-drop-flying which would otherwise be caused by air flow forcing toward a cabin of the automobile such condensed water that is retained between the fins in the rearward heat exchanger due to the capillary phenomenon. Some partition plates may be fixed inside the upper and lower headers to cause coolant to meander along zigzag paths.

Three unit heat exchangers "A", "B" and "C" as shown in Fig. 20 may be combined though two unit heat exchangers are arranged fore and aft in the first to sixth embodiments. Further, four or more unit heat exchangers may be combined in the invention.

As will be apparent from the above description, the duplex heat exchanger in invention is constructed such that the fins are each interposed between two adjacent tubes each having ends respectively connected to the hollow headers in fluid connection therewith. A plurality of the unit heat exchangers are aligned with each other in the direction of air flow, and the coolant paths of said heat exchangers are connected in series or parallel with each other. Therefore, the capacity of heat transfer can be increased for the duplex heat exchanger as a whole because each unit heat exchanger contributes to the heat transfer therein. Such a combination of two or more unit heat exchangers provides a higher degree of freedom in selecting the number and/or location of the partition plates in order to form a desired coolant flow circuit. Thus, an optimal design of the duplex heat exchanger can be employed for a higher heat transfer efficiency and for a lower pressure loss which are indispensable to a good heat exchanger.

The embodiments of the invention in which an exclusive property or privilege is claimed are defined as follows:

1. A duplex heat exchanger comprising unit heat exchangers which respectively comprise a plurality of tubes arranged parallel with each other and which include fins, each interposed between one of such tubes and the next, opposite ends of each tube being connected to a pair of headers in fluid connection therewith; wherein the unit heat exchangers are closely juxtaposed to each other fore and aft in a direction of air flow, coolant paths of said unit heat exchangers are connected in series with each other so that coolant flows from one of the unit heat exchangers lying on the leeward into another unit heat exchanger standing to the windward of said air flow, and wherein partition plates are fixed inside at least one of the headers so as to divide an internal space thereof into at least two sections, so that the coolant paths in the unit heat exchangers are meanders which make zigzag turns caused by the partition plates.

2. The duplex heat exchanger according to claim 1, adapted for use as a condenser, wherein the number of zigzag turns made within the unit heat exchanger standing to the windward is greater than that made within the other

unit heat exchanger lying on the leeward, so that a total cross-sectional area of the former heat exchanger is smaller than that of the latter heat exchanger.

3. The duplex heat exchanger according to claim 2, wherein the total cross-sectional area of the coolant paths in the unit heat exchanger standing to the windward is 30% to 60% of that in the unit heat exchanger lying on the leeward.

4. The duplex heat exchanger according to claim 1, wherein the total cross-sectional area of the coolant paths gradually decreases from an inlet toward an outlet of the coolant whereby the duplex heat exchanger is adapted for use as a condenser.

5. The duplex heat exchanger according to any one of claims 1 to 4, wherein a surface area in contact with the air flow in one of two unit heat exchangers which stands to the windward is smaller than that in the other unit heat exchanger.

6. The duplex heat exchanger according to any one of claims 1 to 5, wherein the tubes are 6 to 20 mm in width and 1.5 to 7.0 mm in height, coolant paths of tubes are 1.0

mm or more in height, and the fins are 6 to 16 mm in height, with a fin pitch of 1.6 to 4.0 mm.

7. The duplex heat exchanger according to any one of claims 1 to 6, wherein a distance between the headers in one unit heat exchanger is different from those in the other unit heat exchangers.

8. A duplex heat exchanger comprising unit heat exchangers which respectively comprise a plurality of tubes arranged parallel with each other and which include fins, each interposed between one of such tubes and the next, opposite ends of each tube being connected to a pair of headers in fluid connection therewith; wherein the unit heat exchangers are closely juxtaposed to each other fore and aft in a direction of air flow, coolant paths of said unit heat exchangers are connected in parallel with each other so that coolant flows simultaneously through said unit heat exchangers; and wherein partition plates are fixed inside at least one of the headers so as to divide an internal space thereof into at least two sections, so that the coolant paths in the unit heat exchangers are meanders which make zigzag turns caused by the partition plates

9. The duplex heat exchanger according to claim 8, wherein the number of zigzag turns within the coolant path in the unit heat exchanger lying on the leeward is greater than that in other unit heat exchangers standing to the windward.

10. The duplex heat exchanger according to claim 8 or 9, wherein the fins are each spanned between two adjacent unit heat exchangers.

11. The duplex heat exchanger according to claim 8, 9 or 10, and adapted for use as an evaporator, wherein the headers in said unit heat exchangers are horizontal and are connected with the tubes disposed vertically, coolant paths of said unit heat exchangers are connected in parallel with each other so that coolant flows simultaneously through said unit heat exchangers, and wherein the pitch of the fins in the unit heat exchanger lying on the leeward is greater than that in other unit heat exchangers standing to the windward.

12. A condenser comprising unit heat exchangers which respectively comprise a plurality of flat tubes arranged parallel with each other and which include 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 paths 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, said 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 and a coolant outlet pipe being 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, and wherein heat exchange capacity of the leeward unit heat exchanger is greater than that of the windward unit heat exchanger.

13. A condenser according to claim 12, wherein the pitch of the fins of the leeward unit heat exchanger is smaller than that of the windward unit heat exchanger.

14. A condenser according to claim 12 or 13, wherein the number of zigzag turns within the coolant path in the windward unit heat exchanger is greater than that in the leeward unit heat exchanger.

15. A condenser according to claim 12, 13 or 14, wherein 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.

16. A condenser according to any one of claims 12 to 15, wherein the cross-sectional area of the coolant flow paths gradually decreases from the inlet section towards the outlet section.

17. A condenser according to any one of claims 12 to 16, wherein 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 pitch of the fins is 1.6 - 4.0 mm.

18. A condenser comprising two unit heat exchangers which respectively comprise a plurality of flat tubes arranged parallel with each other and which include 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 a direction of air flow, wherein coolant paths of the unit heat exchangers are connected in parallel with each other so that the coolant flows simultaneously through the unit heat exchangers, the coolant path of each heat exchanger is formed in a meandering pattern to make zigzag turns by means of at least one partition means secured in at least one header, and wherein the number of zigzag 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.

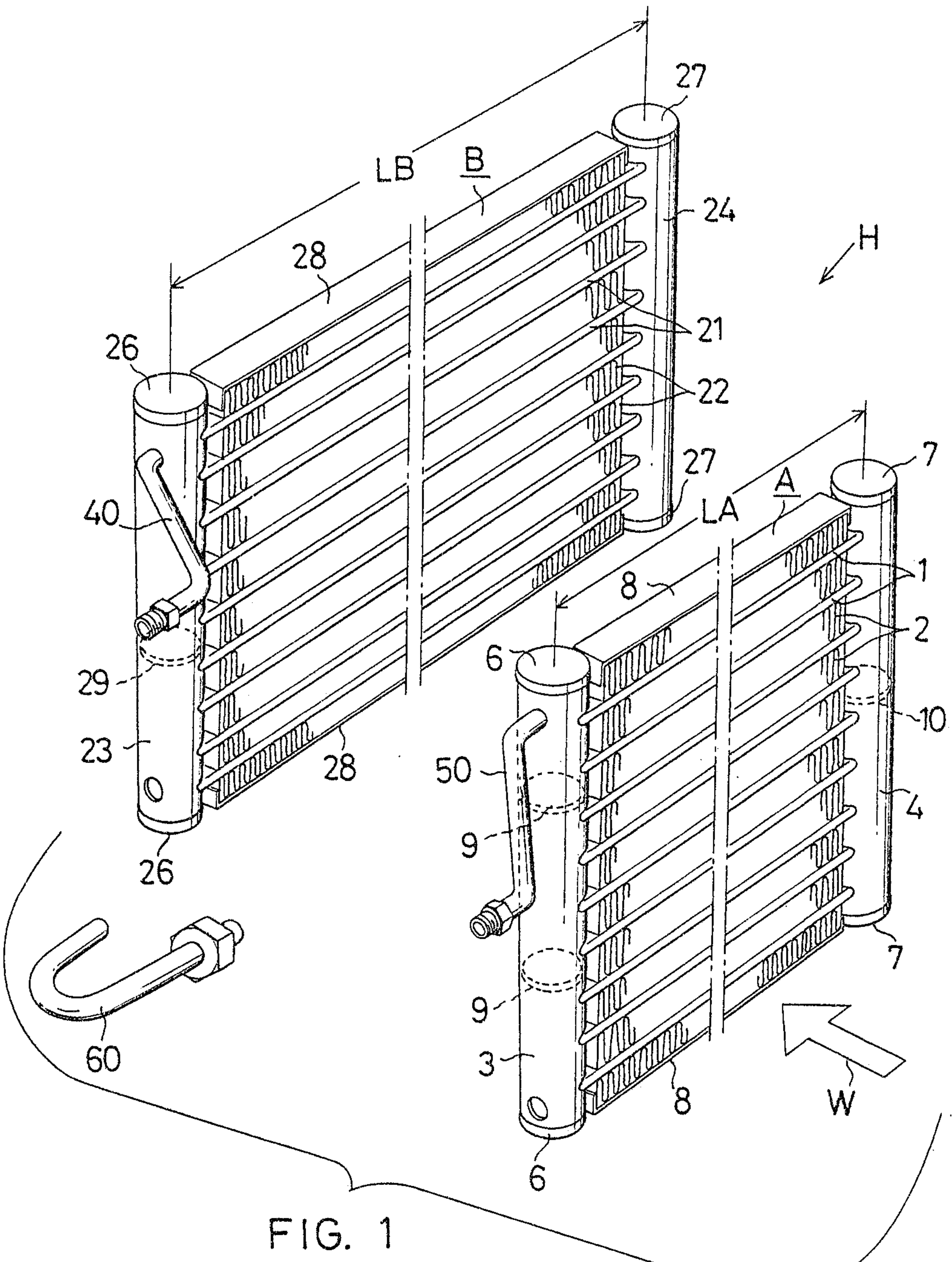


FIG. 1

Marks & Clerk

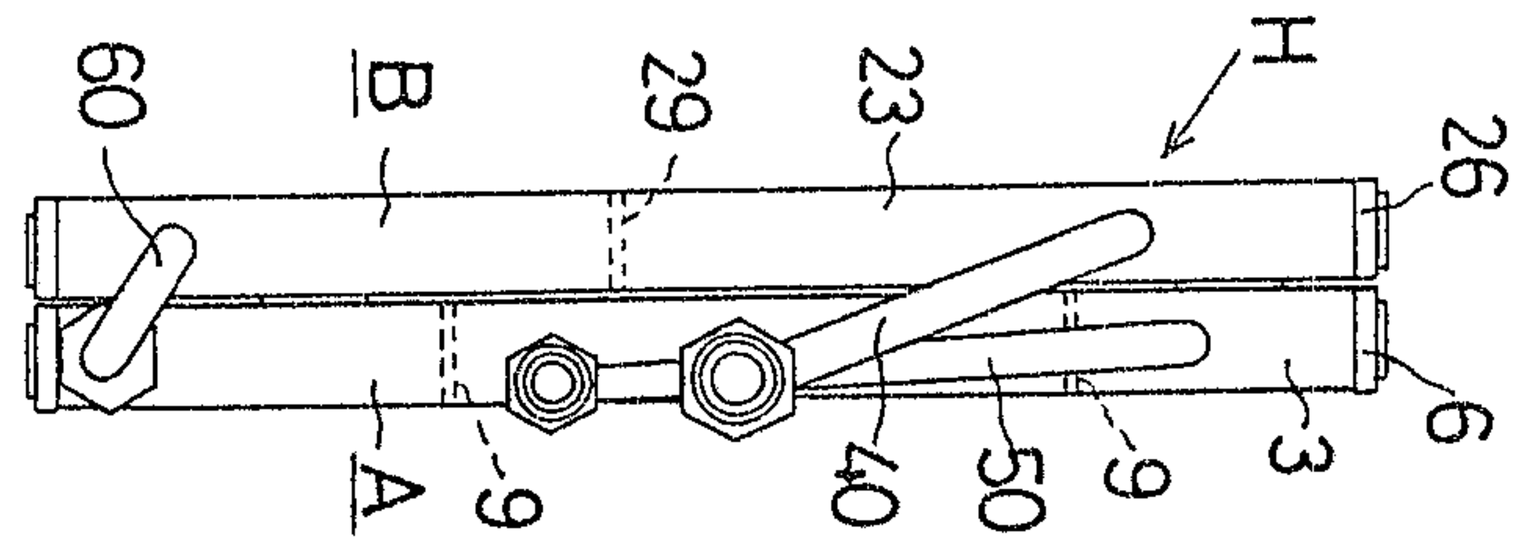


FIG. 4

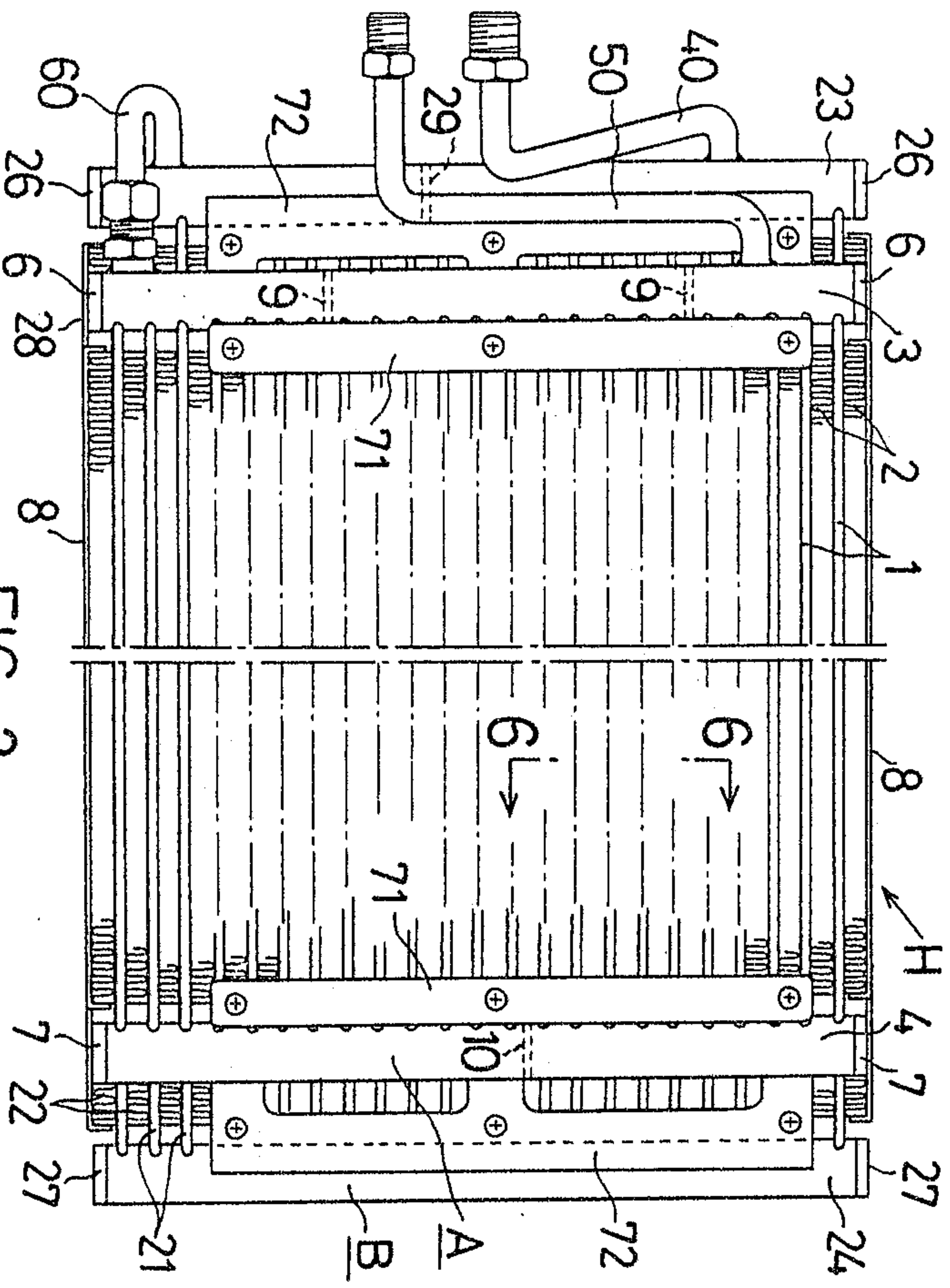


FIG. 2

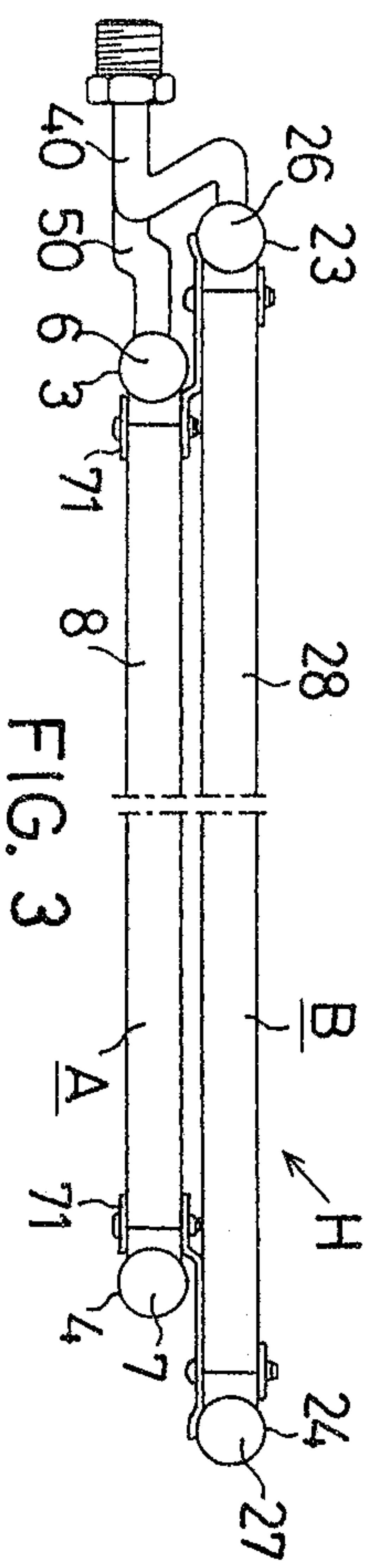


FIG. 3

Marks & Clerk

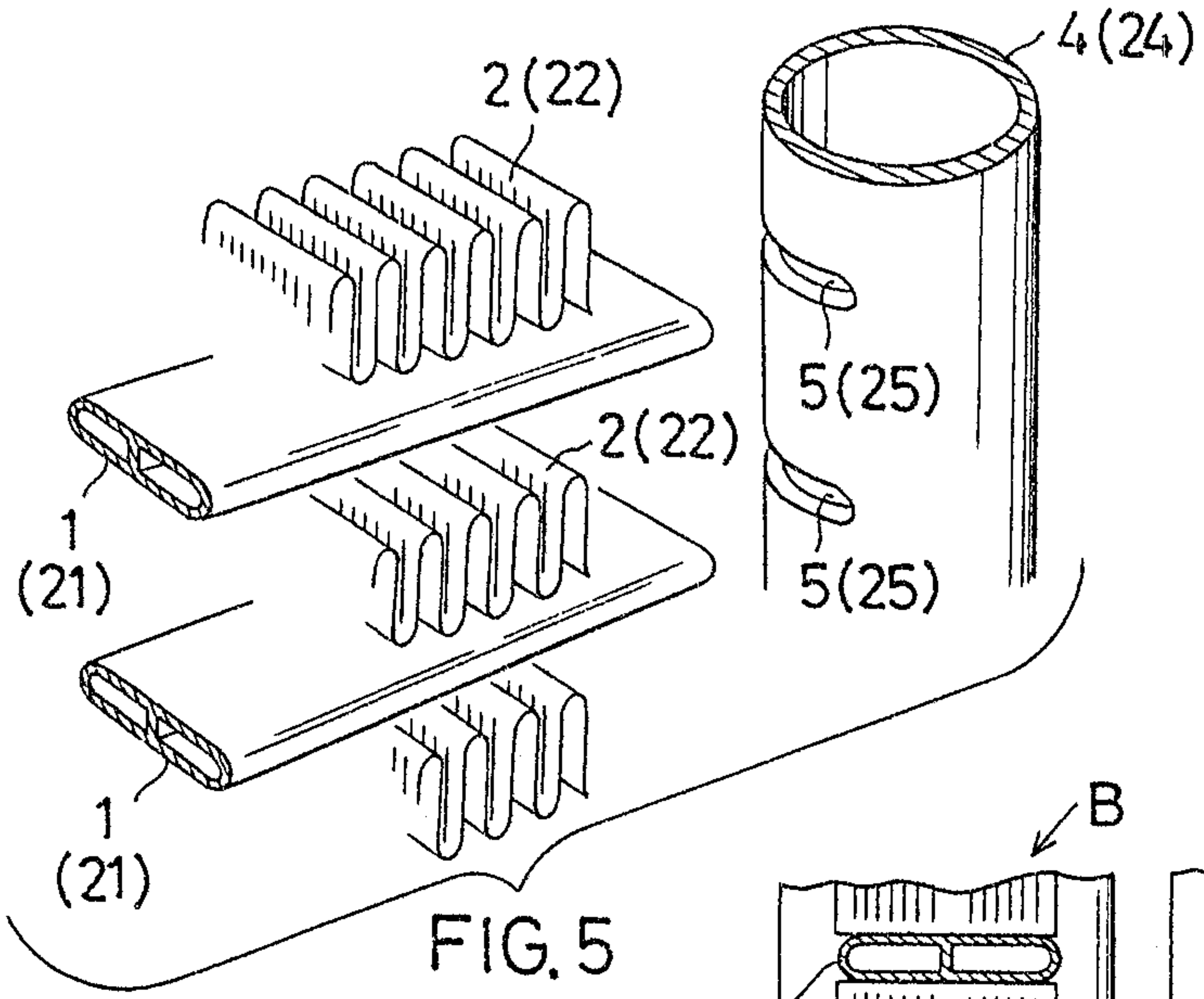


FIG. 5

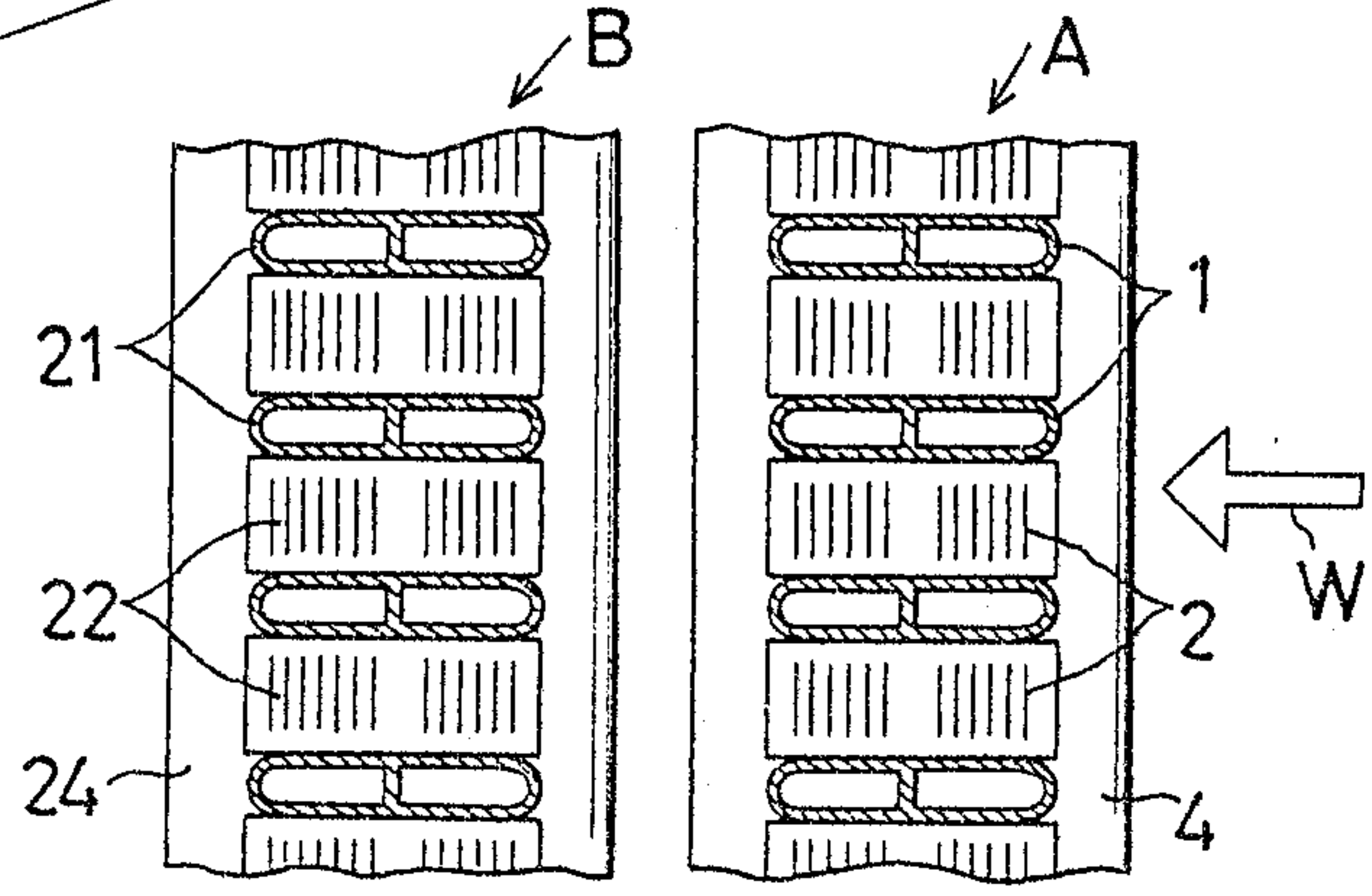


FIG. 6

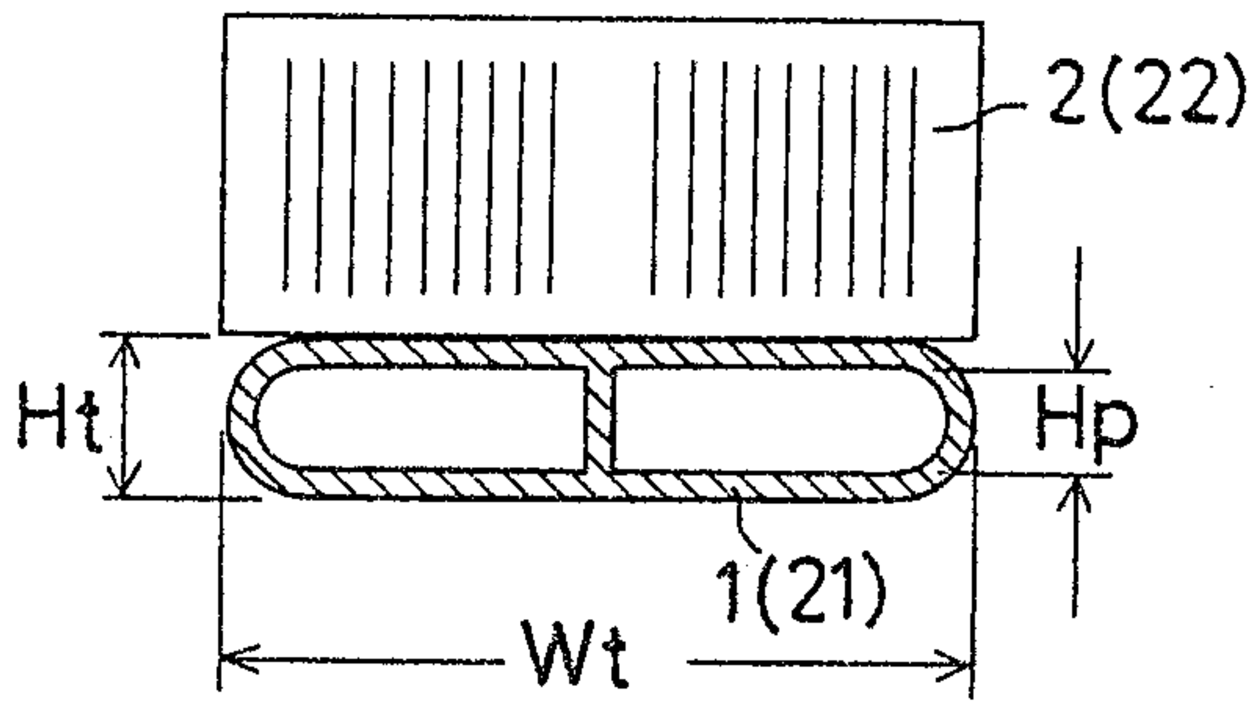


FIG. 7

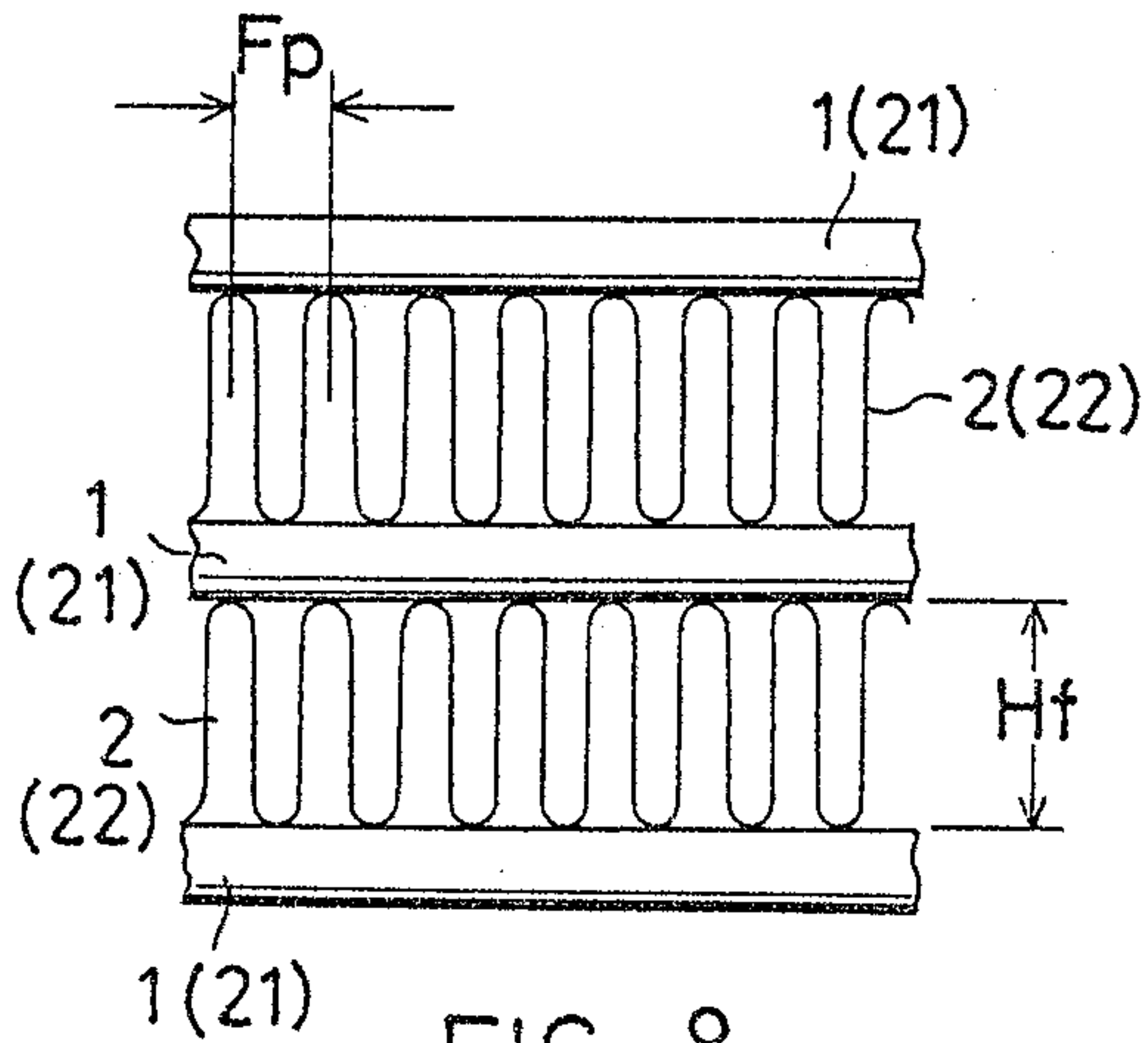


FIG. 8

Marks & Clerk

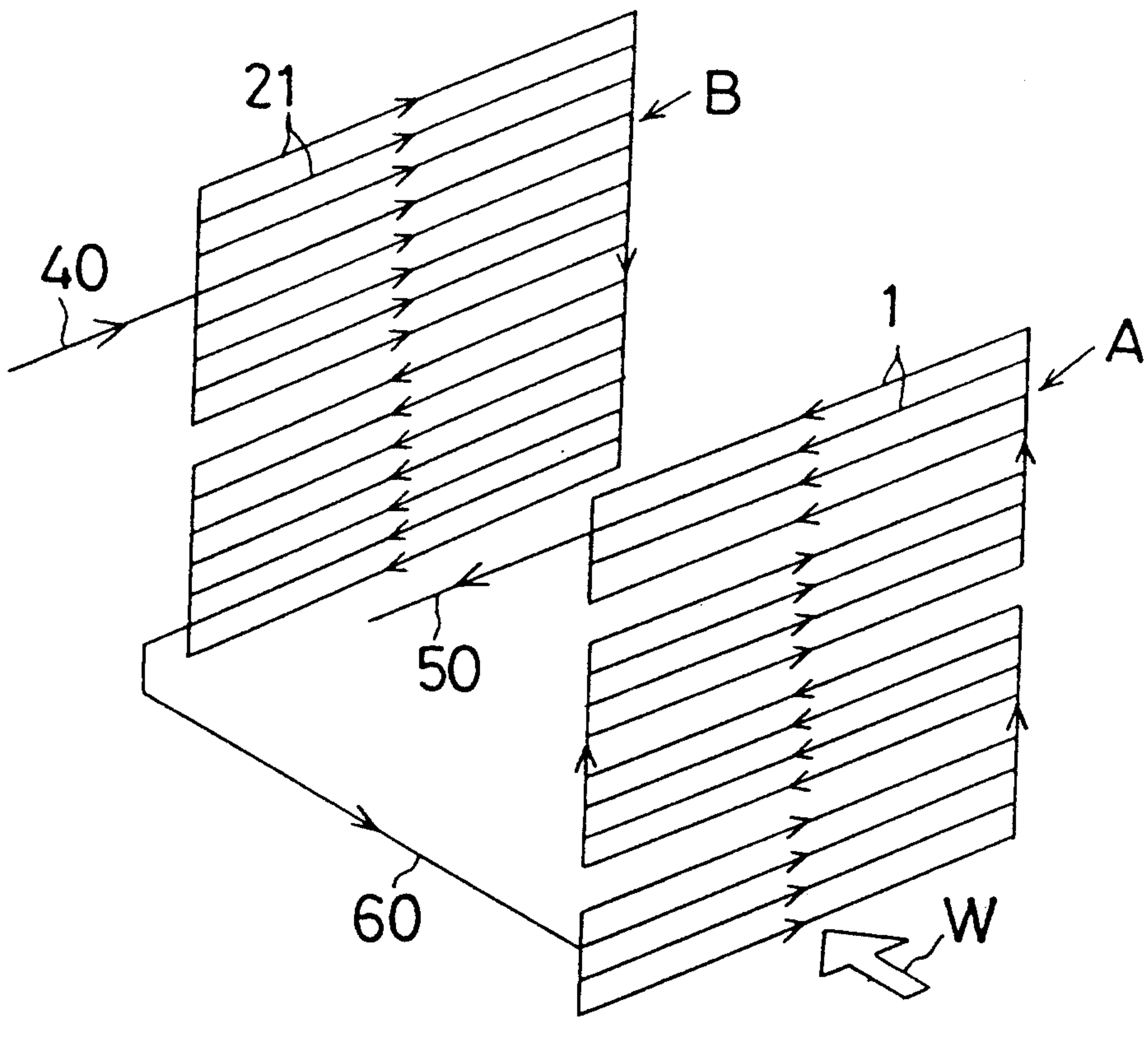


FIG. 9

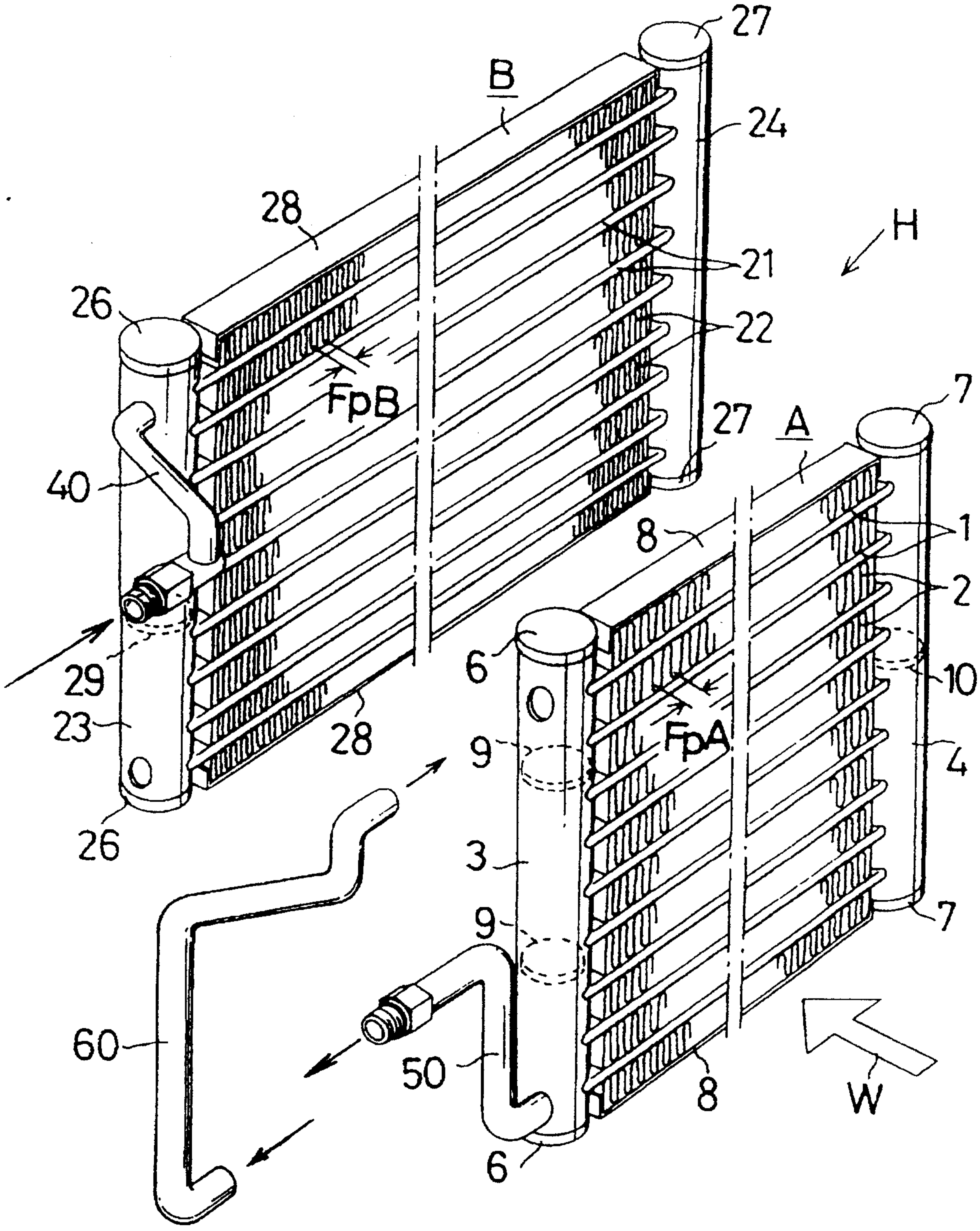

27
B
24
28
H
21
26
22
FpB
27
A
7
40
8
1
2
6
10
29
FpA
23
9
4
28
26
3
9
7
60
50
8
W
6

FIG. 10

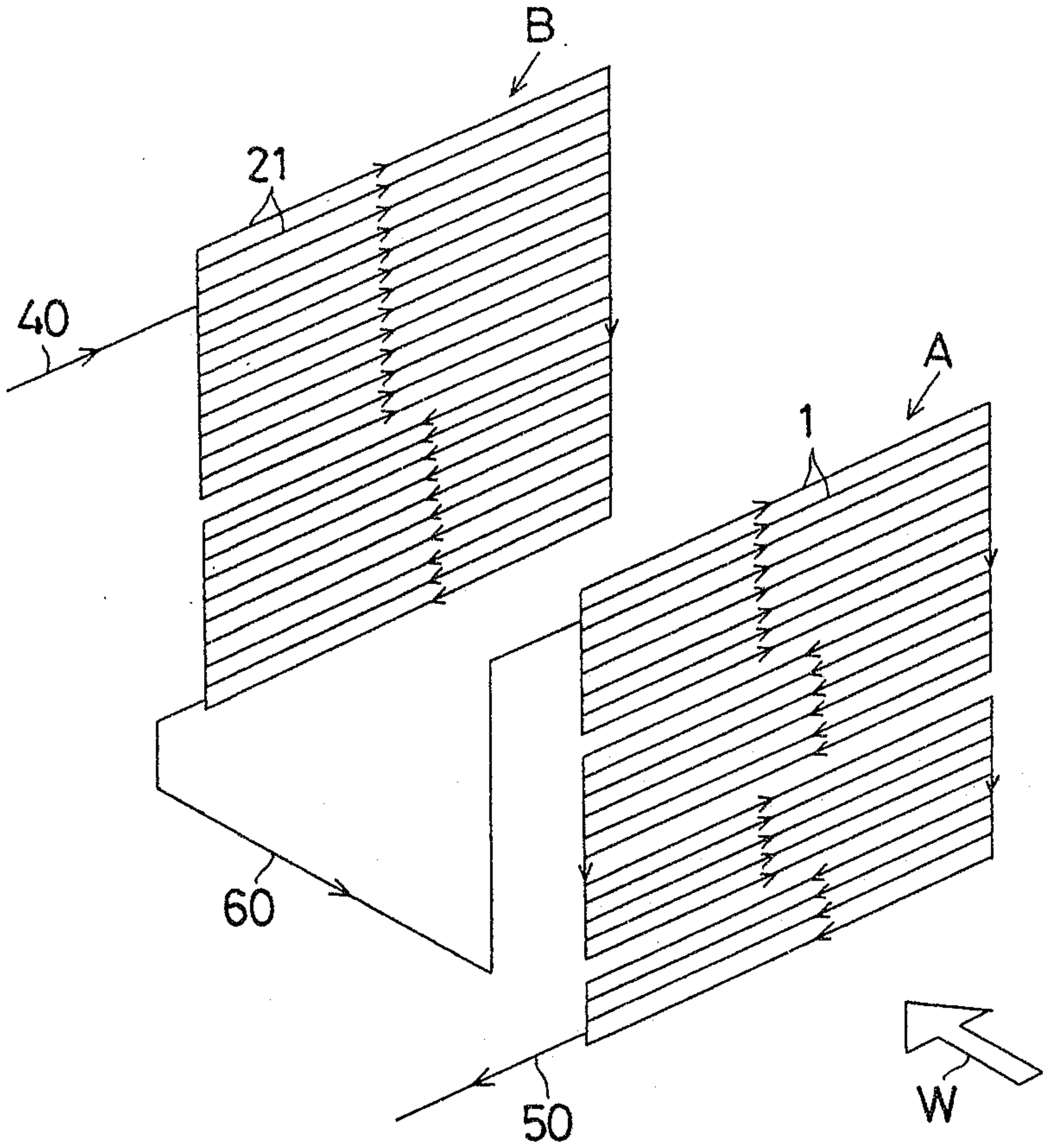


FIG. 11

Marks & Clerk

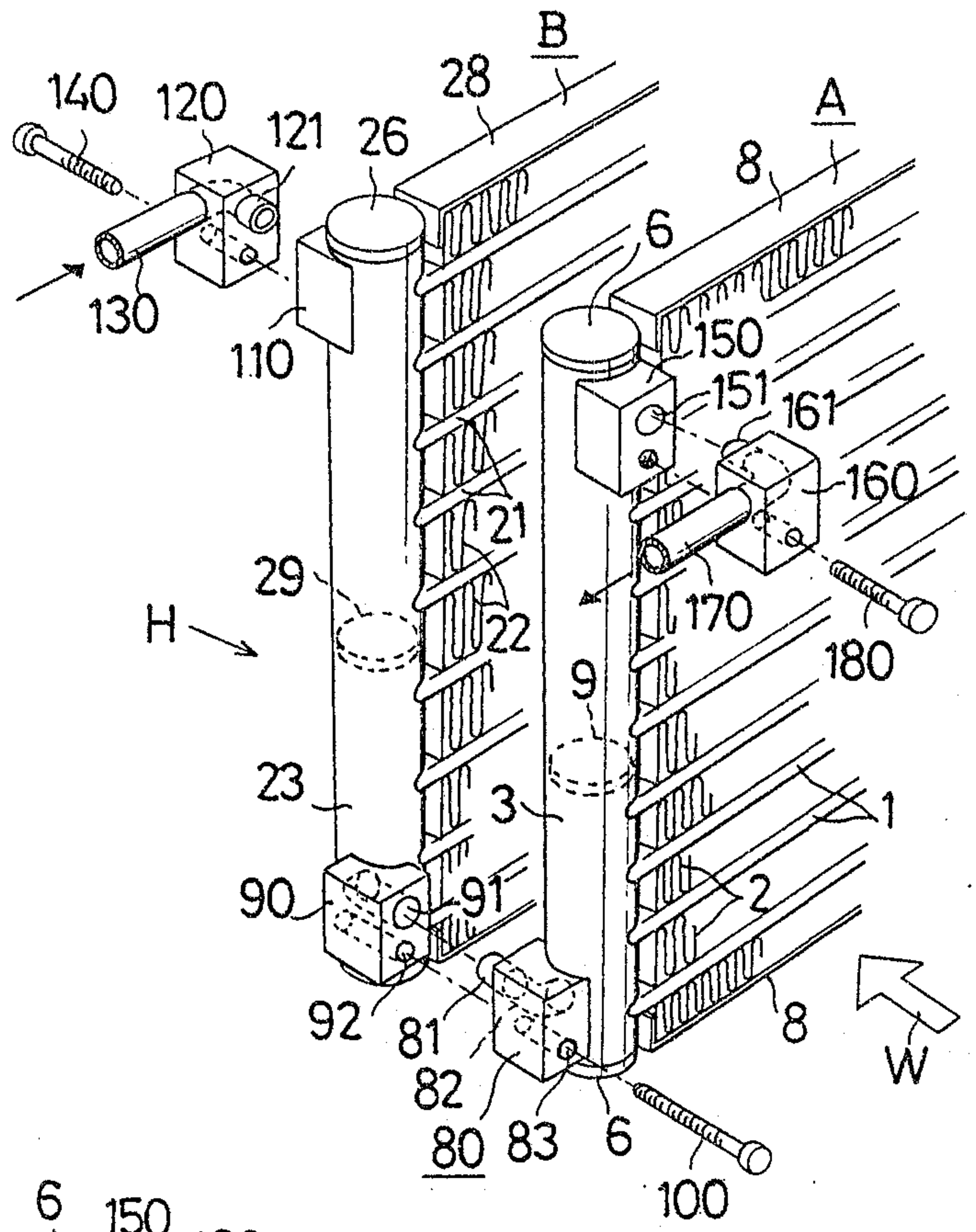


FIG. 12

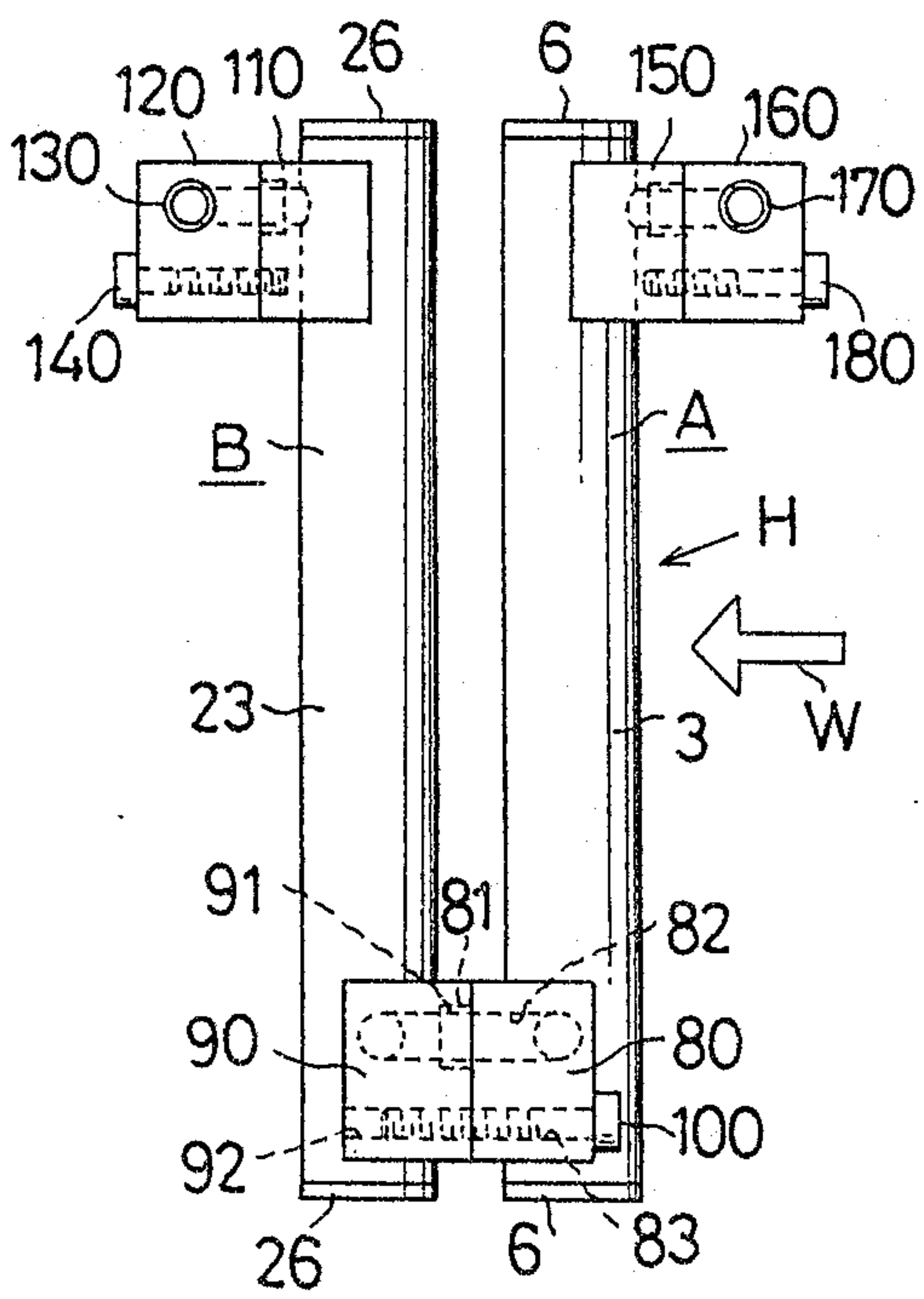


FIG. 13

Marks & Clerk

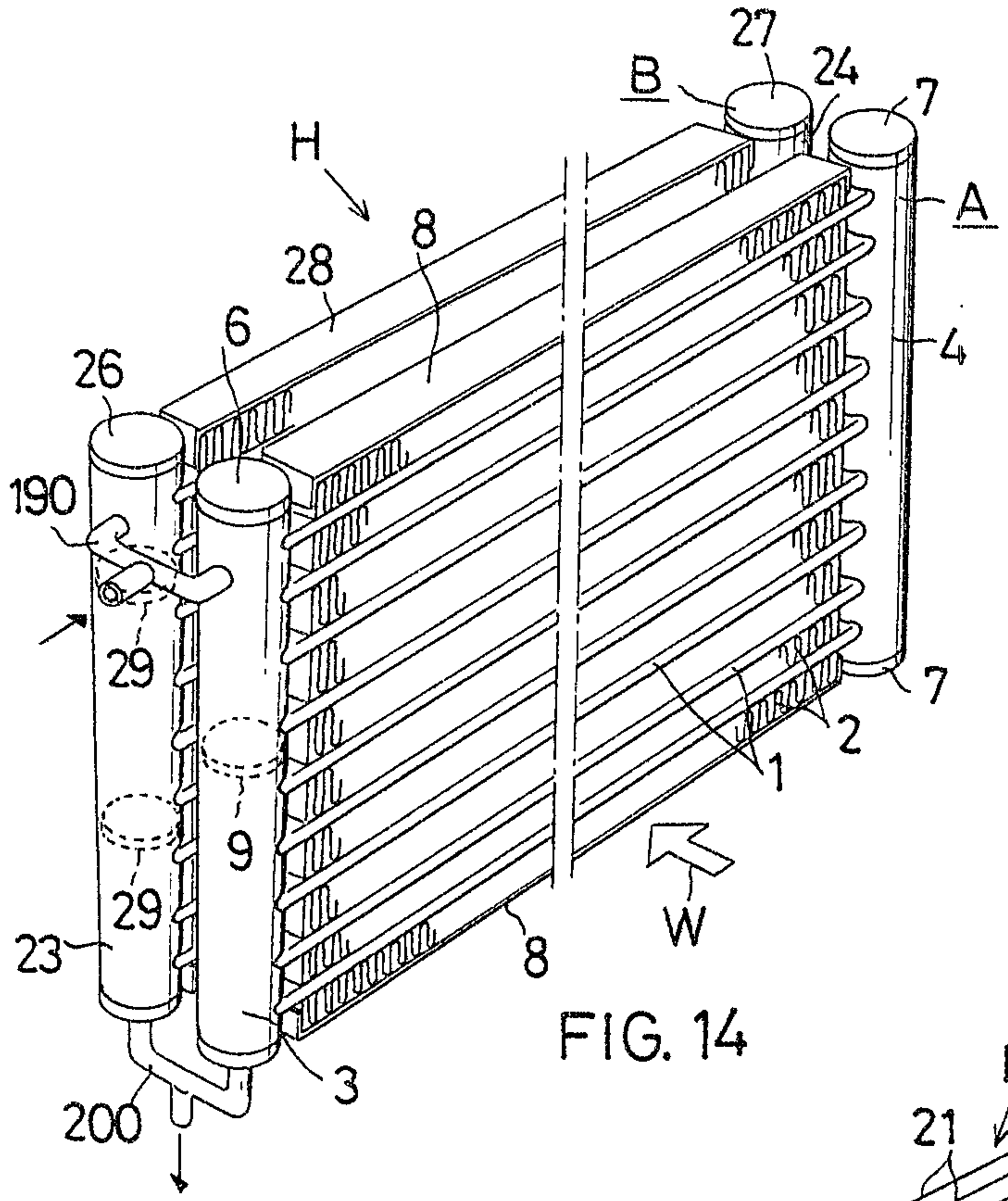


FIG. 14

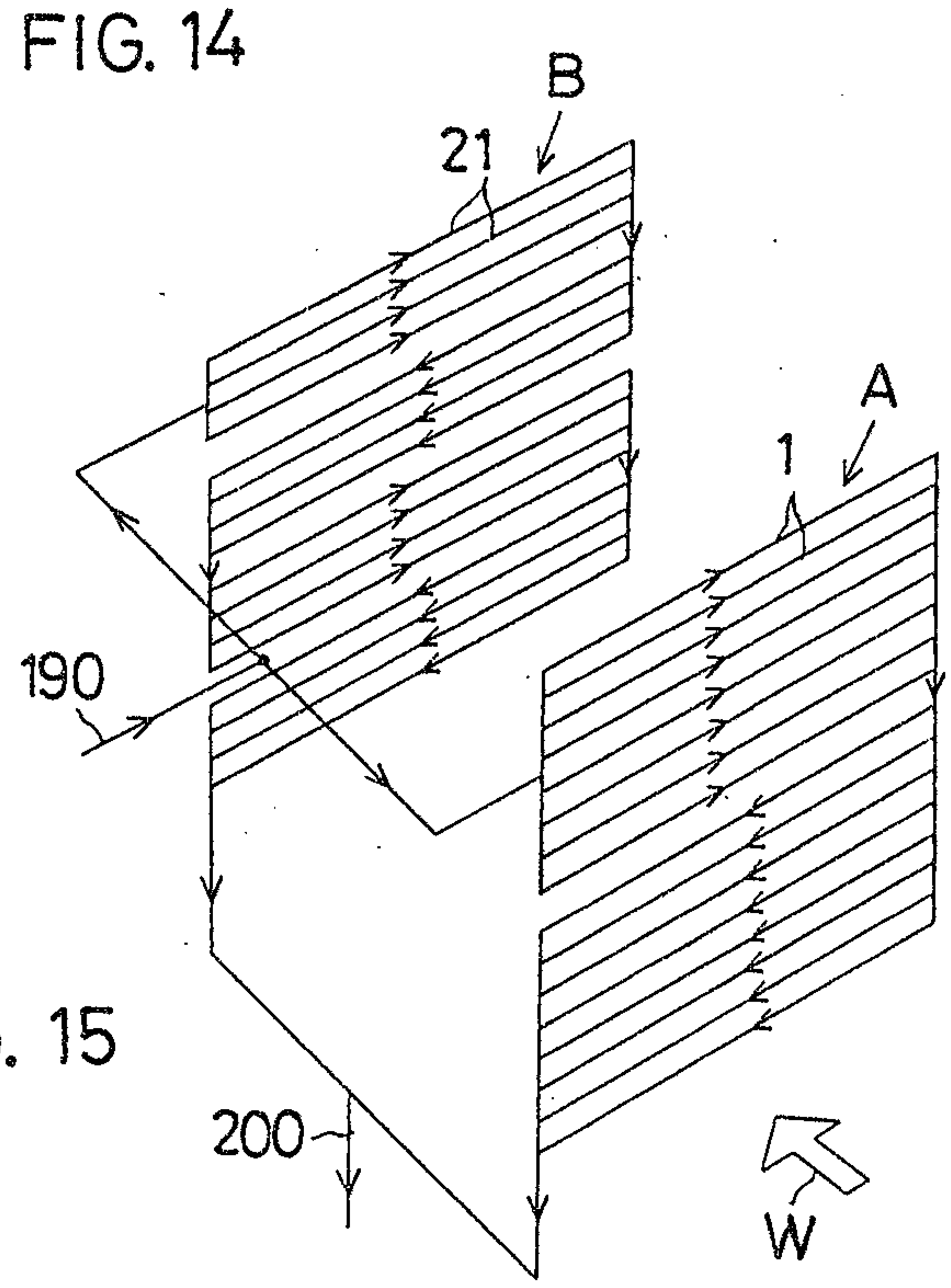


FIG. 15

Marks & Clerk

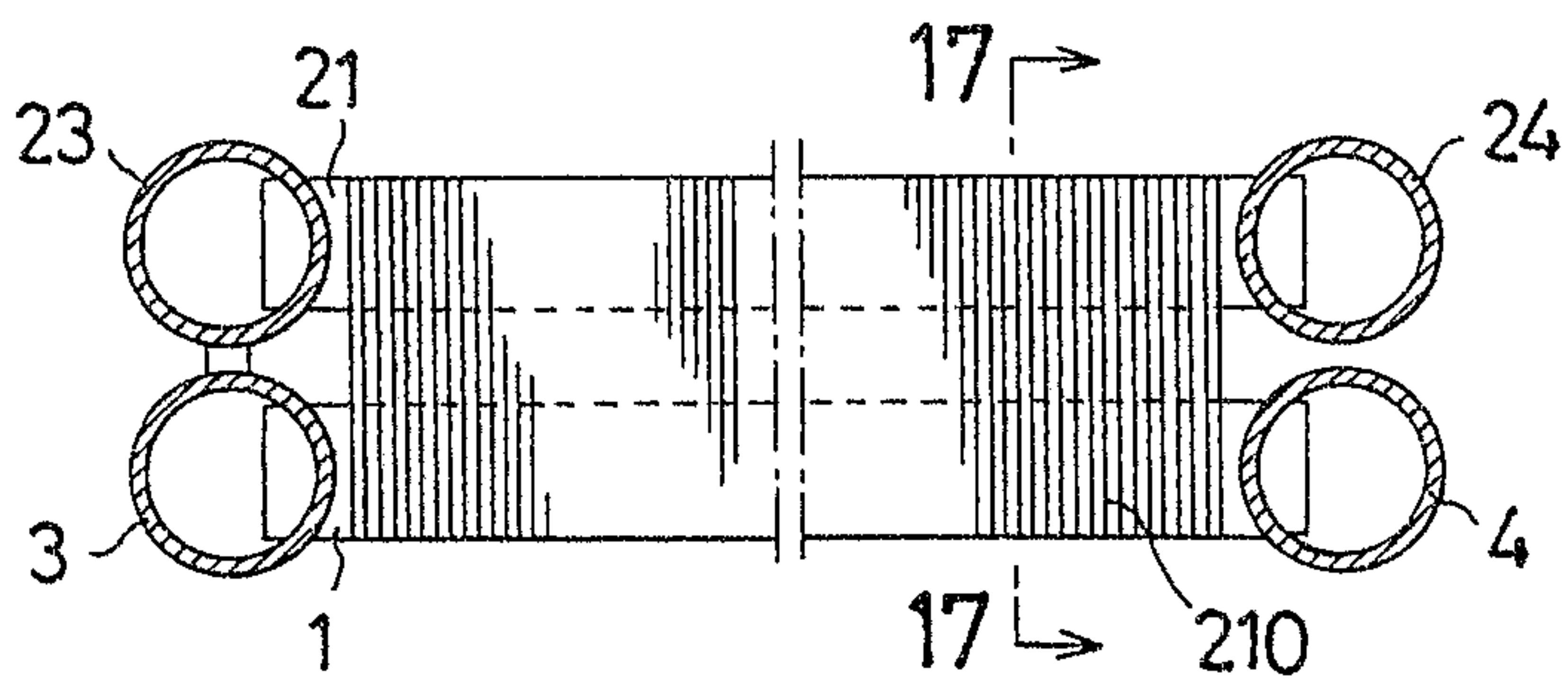


FIG. 16

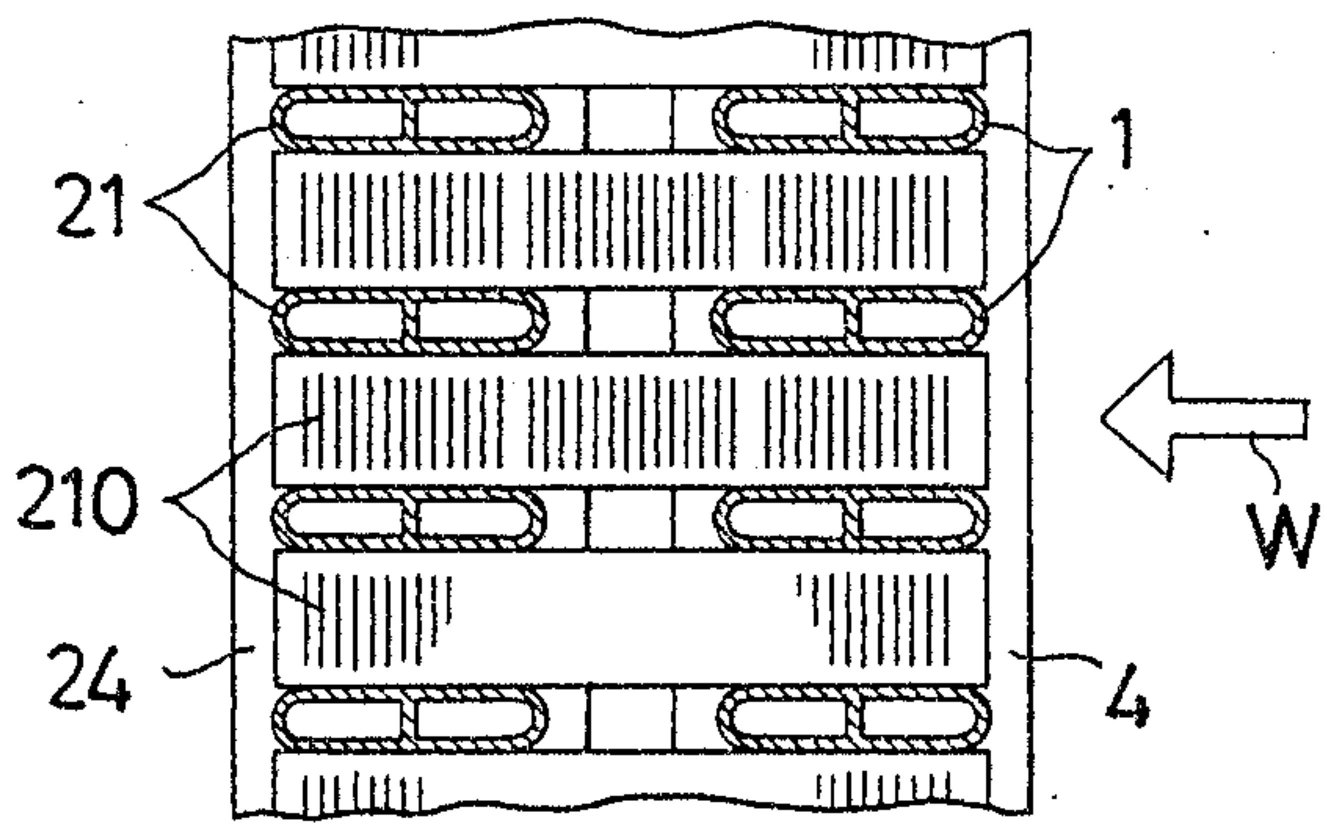


FIG. 17

Marks & Clerk

2023499

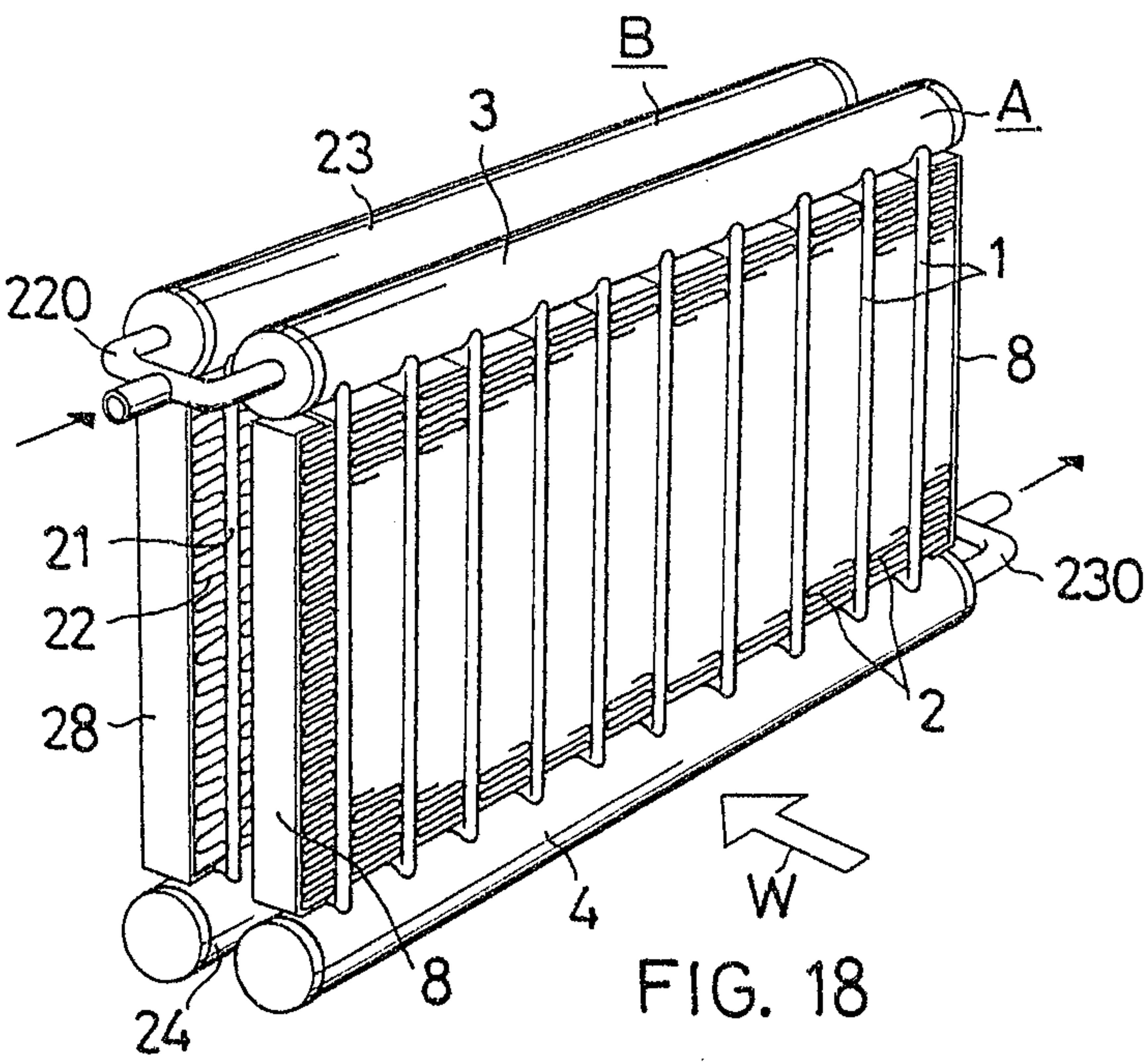


FIG. 18

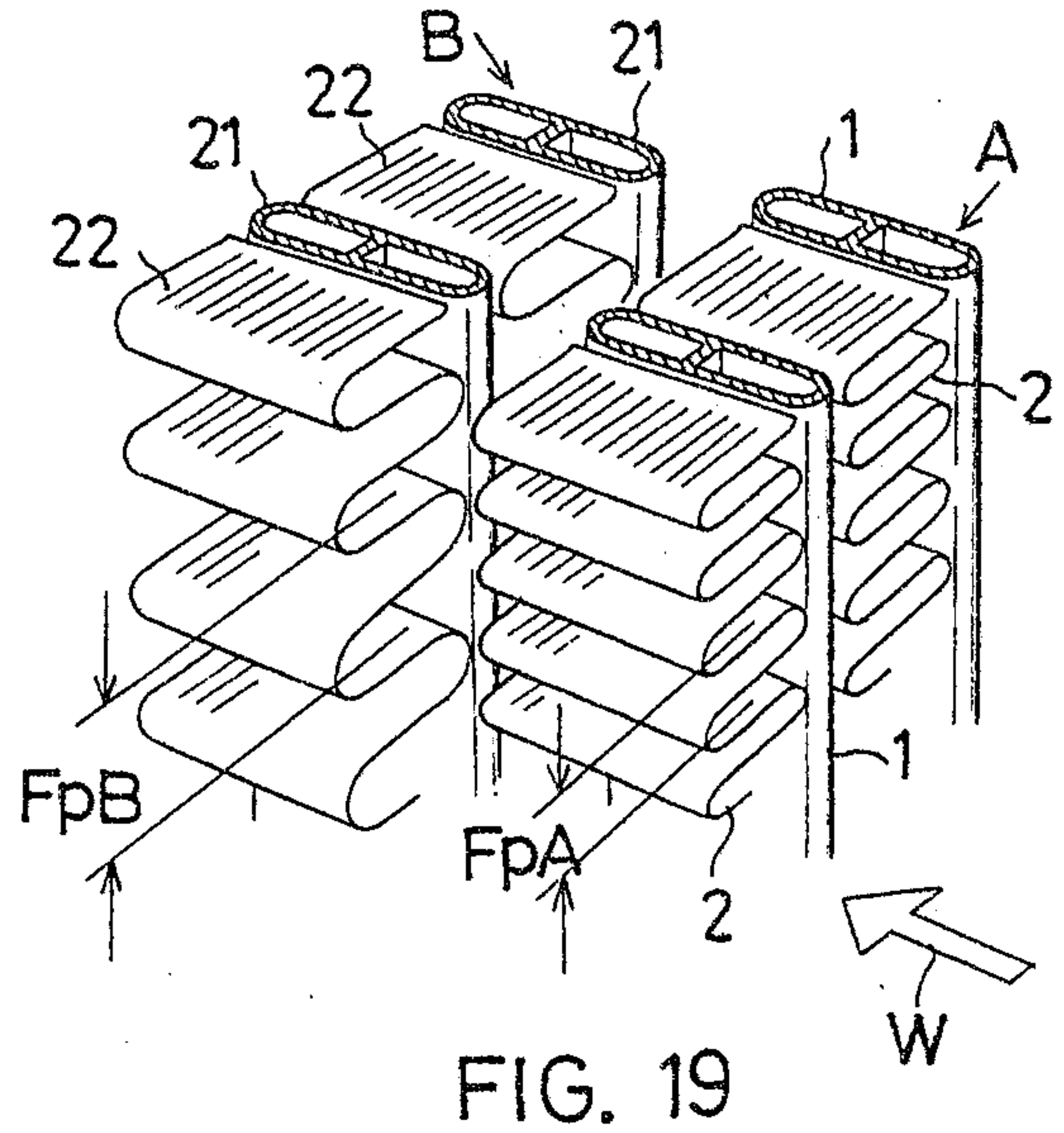


FIG. 19

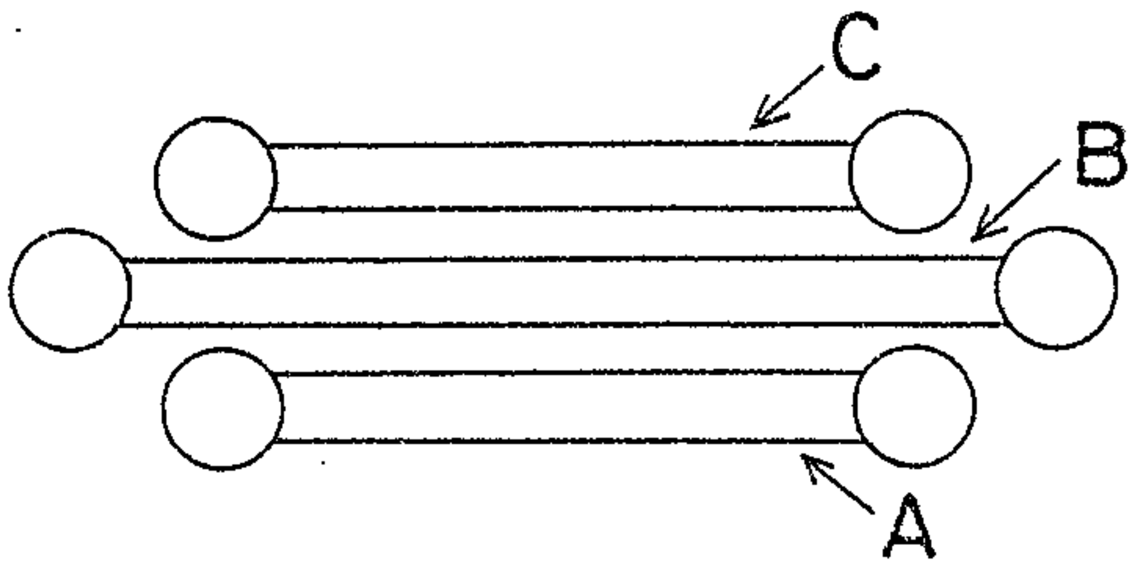
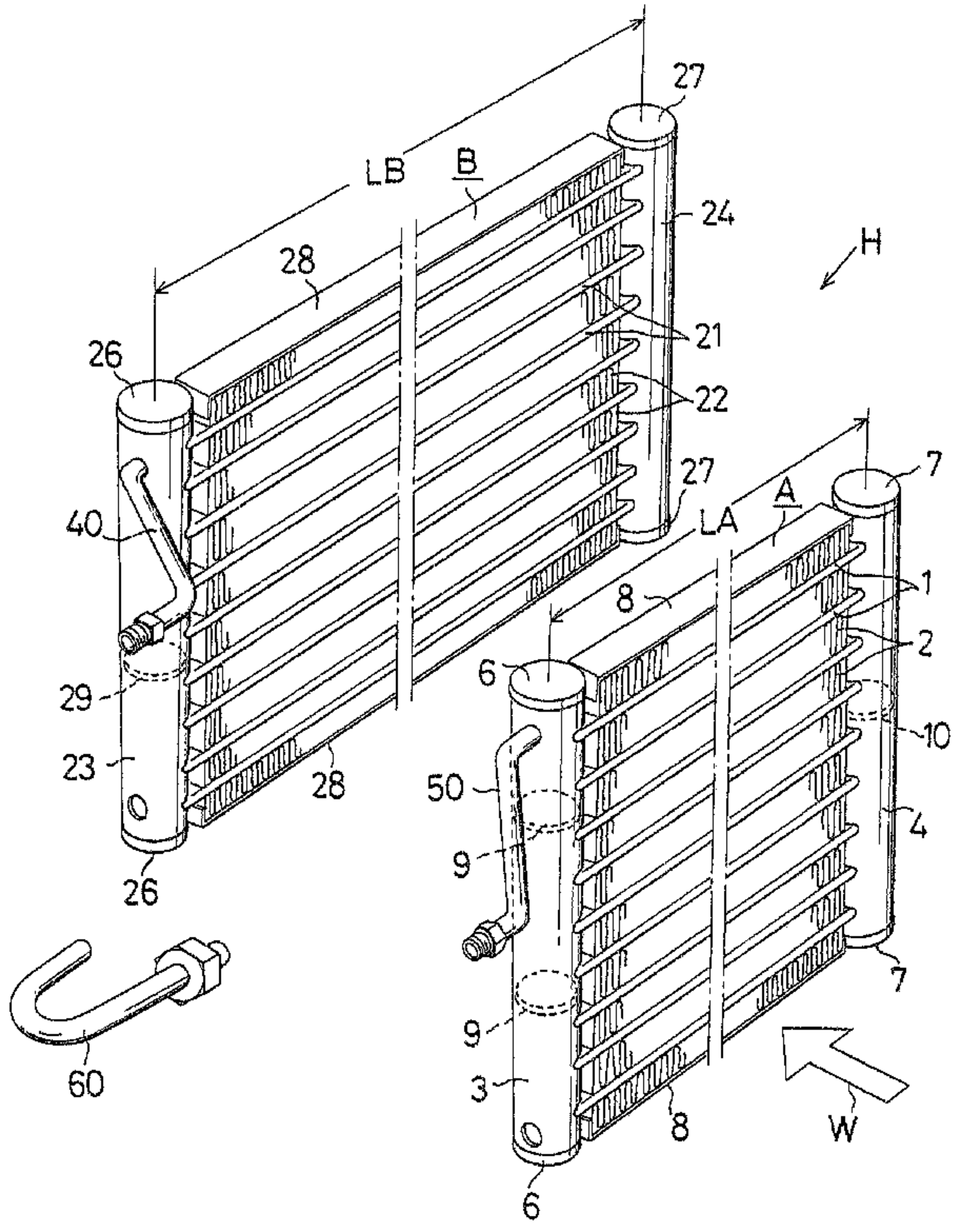


FIG. 20

Marks & Clerk



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