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(71) Applicant: Denso Corporation
Kariya-city, Aichi-pref., 448-8661 (JP)

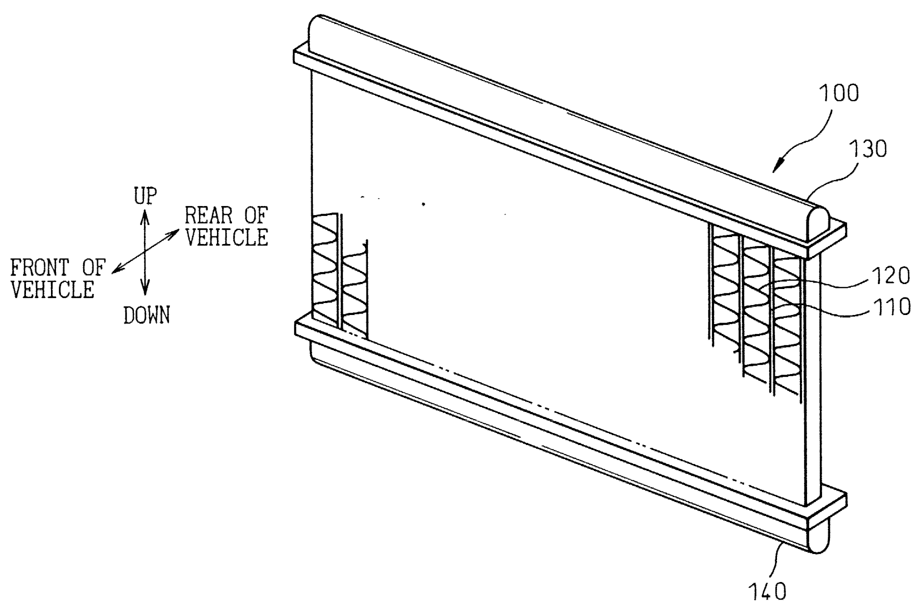
(72) Inventor: Kokubunji, Hiroshi
Kariya-shi, Aichi 448-8661 (JP)

(74) Representative:
Klingseisen, Franz, Dipl.-Ing. et al
Patentanwälte,
Dr. F. Zumstein,
Dipl.-Ing. F. Klingseisen,
Postfach 10 15 61
80089 München (DE)

(54) Heat exchanger

(57) A heat exchanger, in which the space (110a) in each of a plurality of tubes (110) is segmented by a partitioning wall (111), and dimples (protrusions) (112) are formed on the inner wall of the tubes (110), is disclosed. The substantial sectional area of the flow path, which has a large effect on the flow velocity in the tubes (110), can thus be reduced without increasing the total number of the tubes (110). Even in the case where the radiation area is enlarged by increasing the long diameter of the tube (110), therefore, a turbulent flow of the cooling water can be obtained in collaboration with the agitating action of the dimples (112). In this way, the reduction in heat transfer coefficient between the cooling water and the tubes (110) can be prevented. As a result, the radiation capacity of the radiator (100) can be improved while suppressing the increase in the number of the parts and the production cost of the radiator (100), thereby producing a radiator suitable for large automotive vehicles.

Fig.1



EP 1 128 148 A2

Description

BACKGROUND OF THE INVENTION

1. Field of the Invention

[0001] The present invention relates to a heat exchanger for cooling a coolant (the cooling water) of a liquid-cooled internal combustion engine, such as a water-cooled engine, by exchanging heat between the coolant and the air, or in particular to a heat exchanger effectively applicable to a large vehicle such as a truck or a bus having a large heat release requirement.

2. Description of the Related Art

[0002] As is well known, the radiator has a structure including a plurality of flat tubes with the cooling water flowing therein, a header tank arranged at each of the longitudinal ends of the tube and cooling fins. The cooling water flowing in the tubes and the cooling air exchange heat through the tubes. The heat exchange capacity of the radiator is determined substantially by the following parameters.

1. The heat transfer coefficient between the cooling water and the tubes (hereinafter referred to as the first heat transfer coefficient)
2. The contact area between the cooling water and the tubes (hereinafter referred to as the first radiation area)
3. The heat transfer coefficient between the tube and the cooling fins on the one hand and the air on the other (hereinafter referred to as the second heat transfer coefficient)
4. The contact area between the tubes and the cooling fins on the one hand and the air on the other (hereinafter referred to as the second radiation area)
5. The heat conductivity of the tubes and the cooling fins

[0003] The engine of a large vehicle has a high heating value, and therefore requires a radiator of high radiation capacity. This requirement is generally met by increasing the size of the radiator and thereby increasing the first and second radiation areas, as obvious from the parameters described above. In an automotive radiator, the limited equipment space generally makes it difficult to increase the radiation areas by increasing the number of the tubes and the cooling fins. In many cases, therefore, the first and second radiation areas are increased by increasing the long diameter of the tubes.

[0004] In the case where the sectional area of the tube is increased by increasing the long diameter thereof, however, the flow velocity of the cooling water flowing in the tube decreases. The resulting decreased Reynolds number causes a laminar flow of the cooling water in the tube, which in turn reduces the first heat transfer coefficient, thereby decreasing the heat exchange capacity of the radiator.

[0005] This problem can be solved by a radiator structure in which a multiplicity of tubes having a small sectional area are arranged along the long diameter thereof to assure a turbulent flow of the cooling water in the tubes. An increased number of tubes, however, increases the number of the required parts and hence the production cost.

SUMMARY OF THE INVENTION

[0006] In view of the problems mentioned above, the object of the present invention is to provide a heat exchanger not increased in production cost and suitable for the large vehicles.

[0007] In order to achieve the object described above, according to a first aspect of the invention, there is provided a heat exchanger comprising a plurality of tubes (110) having a flat section with a coolant flowing therein, a plurality of cooling fins (120) arranged between the tubes (110) for increasing the radiation area by contacting the outer surface of the tubes (110), a first header tank (130) arranged at a longitudinal end of the tubes (110) for supplying by distributing the coolant to the tubes (110), and a second header tank (140) arranged at the other longitudinal end of the tubes (110) for recovering by collecting the coolant flowing out of the tubes (110), wherein the tubes (110) each have therein a partitioning wall (111) for partitioning the internal space (110a) of the tube (110) into a plurality of spaces (110b, 110c) along the long diameter thereof, and the inner wall of the tubes (110) is formed with a plurality of protrusions (112) extending inward of the tube (110).

[0008] With this configuration, the substantial sectional area of the flow path having a great effect on the flow velocity in the tubes (110) can be reduced without increasing the total number of the tubes (110). Even in the case where the radiation area is increased by increasing the size of the tubes (110) along the long diameter thereof, therefore, a turbulent coolant flow can be produced in collaboration with the agitating action of the protrusions (112).

[0009] As a result, the heat transfer coefficient between the coolant and the tubes (110) can be prevented from decreasing. Therefore, the radiation capacity of the heat exchanger (100) can be increased while suppressing the increase in the number of the parts and the production cost of the heat exchanger, and a heat exchanger suitable for large vehicles can be produced.

[0010] According to a second aspect of the invention, there is provided a heat exchanger wherein, desirably, the ratio of the short diameter (H) to the long diameter (L) of the tubes (110) is not less than 0.035 but not more than 0.1, the ratio of the distance (d1) between the partitioning wall (111) and a given one of the protrusions (112) to the long diameter (L) of the tubes (110) is not less than 0.15 but not more than 0.3, the ratio of the size (A) of the portion of the protrusions (112) parallel to the long diameter (L) of the tubes (110) to the long diameter (L) of the tubes (110) is not less than 0.05 but not more than 0.15, the ratio of the distance (d2) between the protrusions (112) to the long diameter (L) of the tubes (110) is not less than 0.15 but not more than 0.25, and the ratio of the extension (h) of the protrusions (112) to the short diameter (H) of the tubes (110) is not less than 0.15 but not more than 0.25.

[0011] According to a third aspect of the invention, there is provided a heat exchanger wherein, desirably, the ratio of the short diameter (H) to the long diameter (L) of the tubes (110) is not less than 0.05 but not more than 0.09, the ratio of the distance (d1) between the partitioning wall (111) and the protrusions (112) to the long diameter (L) of the tubes (110) is not less than 0.2 but not more than 0.25, the ratio of the size (A) of the portion of the protrusions (112) parallel to the long diameter (L) of the tubes (110) to the long diameter (L) of the tubes (110) is not less than 0.07 but not more than 0.12, the ratio of the distance (d2) between the protrusions (112) to the long diameter (L) of the tubes (110) is not less than 0.2 but not more than 0.23, and the ratio of the extension (h) of the protrusions (112) to the short diameter (H) of the tubes (110) is not less than 0.18 but not more than 0.2.

[0012] Incidentally, the reference numerals indicated in the parentheses following the various means described above represent an example of correspondence with the specific means described later with reference to embodiments.

[0013] The present invention may be more fully understood from the description of preferred embodiments of the invention set forth below, together with the accompanying drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

[0014] In the drawings:

Fig. 1 is a perspective view of a radiator according to an embodiment of the invention.

Fig. 2 is a sectional view of a tube employed for the radiator according to an embodiment of the invention.

Fig. 3 is a diagram for explaining the steps (a) to (g) of the process for fabricating the tube employed for the radiator according to an embodiment of the invention.

Fig. 4 is a model diagram showing the state of a temperature boundary layer and a velocity boundary layer employed for the radiator according to an embodiment of the invention.

Fig. 5 is a graph showing the relation between the flow velocity of the cooling water and the heat transfer coefficient between the cooling water and the tube 110.

Figs. 6A to 6D are graphs showing the relation between the heat release and the specifications of the tubes employed for the radiator according to an embodiment of the invention.

Fig. 7 is a graph showing the relation between the heat release of the tubes employed for the radiator and the flow velocity according to an embodiment of the invention.

Fig. 8 is a sectional view of the tube employed for the radiator according to a modification of the invention.

Fig. 9 is a diagram for explaining the steps (a) to (e) of the process for fabricating the tube employed for the radiator according to another embodiment of the invention.

Fig. 10 is a diagram for explaining the steps (a) to (d) of the process for fabricating the tube employed for the radiator according to still another embodiment of the invention.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

[0015] A heat exchanger according to an embodiment of the invention is used for the radiator of a large vehicle. Fig. 1 is a perspective view of a radiator 100 according to this embodiment. The large vehicle is defined as a truck or an omnibus having mounted thereon a large engine of not less than 7000 cc in displacement or not less than 200 thousand W in heating value.

[0016] In Fig. 1, reference numeral 110 designates a plurality of tubes in which the cooling water flows. A corrugated cooling fin (hereinafter referred to simply as the fin) 120 for increasing the radiation area by contacting the outer surface of each tube 110 is arranged between the tubes 110. Both the tubes 110 and the fins 120 are made of aluminum, and integrated with each other by brazing, thereby constituting a heat exchange core for exchanging heat between the cooling water and the air.

[0017] A first header tank 130 for supplying, by distributing, the cooling water to the tubes 110 is arranged at a longitudinal'end (the upper end in this embodiment) of the tubes 110, and a second header tank 140 for recovering the cooling water flowing out of the tubes 110 is arranged at the other longitudinal end (the lower end in this embodiment) of the tubes 110.

[0018] Fig. 2 shows a section (the section at right angles to the length of the tube 110) of the tube 110. This tube 110 is formed by bending a plate material, made of aluminum, covered with a brazing material on the surface thereof. The brazing material is lower in melting point than aluminum (core member) and corresponds to Japanese Industrial Standards (JIS) A4045 according to this embodiment.

[0019] The tube 110 has a flat cross section, and partitioning walls 111a to 111c protruded by bending a part or an end of the plate material inward of the tube 110 are formed at the substantial central portion along the long diameter of the tube 110. These partitioning walls 111a to 111c are laid one on another along the long diameter of the tube 110 and brazed into a single wall with the surfaces thereof parallel to the short diameter of the tube 110 kept in contact with each other.

[0020] Therefore, the space 110a in the tube 110 is segmented into two spaces 110b, 110c along the length of the tube 110 by the partitioning walls 111a to 111c (which are collectively called the partitioning wall 111).

[0021] The internal wall of the tube 110 is formed with a plurality of inward protrusions (dimples) 112 over the whole area along the length of the tube 110. According to this embodiment, as described later, the dimples 112 are formed by press machining (plastic machining). The portions of the outer wall of the tubes 110 which are formed with the dimples 112a, therefore, are formed with recesses 112a which are depressed inward on the tube 110.

[0022] Now, a method of fabricating the tube 110 will be briefly described.

[0023] Fig. 3 is a diagram for briefly explaining the steps of the process for fabricating the tube 110. The production process includes a blanking step shown in Fig. 3(a), a U-bending step shown in Fig. 3(b), a bending step shown in Fig. 3(c), a dimple shaping step shown in Fig. 3(d), a U-shaping step shown in Fig. 3(e), a bending step shown in Fig. 3(f) and the final step shown in Fig. 3(g), which steps are carried out in the above order.

[0024] Specifically, the portion corresponding to the extend dimension of the tube 110 is cut out of a plate material (see Fig. 3(a)), and a portion corresponding to the partitioning wall 111b is formed by U-bending. Then, the portions corresponding to the partitioning walls 111a, 111c are formed by bending (see Fig. 3(c)), after which the dimples 112 are formed by pressing (see Fig. 3(d)). The resulting plate material is bent as shown in Fig. 3(e), 3(f) and 3(g) and in that order thereby to form the spaces 110b, 110c.

[0025] Now, the features of this embodiment will be described.

[0026] According to this embodiment, the space 110a in one tube 110 is segmented into a plurality of (two, in this embodiment) spaces by the partitioning walls 111. Therefore, the substantial sectional area of the flow path having a great effect on the flow velocity in each tube 110 can be reduced without increasing the total number of the tubes 110.

[0027] Thus, even in the case where the radiation area is enlarged by increasing the long diameter of the tubes 110, a turbulent flow of the cooling water can be created in collaboration with the agitating action of the dimples 112. In this way, it is possible to prevent the heat transfer coefficient between the cooling water and the tubes 110 from decreasing.

[0028] Further, the radiation capacity of the radiator 100 can be increased while at the same time suppressing the increase in the number of the parts and the production cost of the radiator 100 and, therefore, a radiator suitable for large vehicles can be produced.

[0029] In the tube 110, as shown in Fig. 4, a temperature boundary layer and a velocity boundary layer are generated originating from the ends of the partitioning wall 111 along the length of the tube 110 and each of the dimples 112. In the neighborhood of the partitioning wall 111, however, the two types of boundary layers, the temperature foundary layer and the velocity foundary layer, (which hereinafter will be referred to as the first boundary layer) generated by the partitioning wall 111 and the two types of boundary layers (which hereinafter will be referred to as the second boundary layer) generated by the dimples 112 interfere with each other. Therefore, the thickness of the first and second boundary layers can be prevented from increasing (growing).

[0030] As a result, the cooling water flow can be positively agitated and therefore the heat transfer coefficient between the cooling water and the tubes 110 can be increased for a higher radiation capacity.

[0031] Incidentally, in Fig. 5, the solid line indicates the heat transfer coefficient between the cooling water in the tube 110 and the tube 110, the dashed line indicates the heat transfer coefficient between the cooling water in the tube 110 assumed to lack the partitioning wall 111 and the particular tube 110, and the one-dot chain indicates the heat transfer coefficient between the cooling water assumed to lack the partitioning wall 111 and the dimples 112 and the particular tube 110. As apparent from Fig. 5, with the tube 110 according to this embodiment, the heat transfer coefficient between the cooling water and the tube 110 increases for an improved radiation capacity. In Fig. 5, characters Rew designates the Raynolds number for the water, Nuw the Nusselt number for the water, and Prw the Prandtl number for the water.

[0032] The graphs of Figs. 6A to 6D show the relation between the radiation capacity and the ratio of the distance dl between the partitioning wall 111 and a given dimple 112 to the long diameter L of the tube 110 (which ratio will hereinafter be referred to as a dimple position d1/L), the ratio of the size A of the portion of a dimple 112 parallel to the long diameter of the tube 110 to the long diameter L of the tube 110 (which ratio will hereinafter be referred to as a dimple length A/L), the ratio of the distance d2 between a plurality of the dimples 112 to the long diameter L of the tube 110 (which ratio will hereinafter be referred to as a dimple pitch d2/L), and the ratio of the size h of the protrusion of the dimple 112 to the short diameter H of the tube 110 (which ratio will hereinafter be referred to as the dimple height h/H). The heat release is expressed as the ratio thereof to the heat release of a simple flat, smooth tube free of the dimples 112 and the partitioning wall 111.

[0033] These graphs apparently show that in the case where the thickness t of the tube 110 is not less than 0.1 mm but not more than 0.5 mm and the ratio of the short diameter H to the long diameter L of the tube 110 (which ratio will hereinafter be referred to as the flatness H/L) is not less than 0.35 but not more than 0.1, it is desirable to assure the dimple position d1/L of not less than 0.15 but not more than 0.3, the dimple length A/L of not less than 0.05 but not more than 0.15, the dimple pitch d2/L of not less than 0.15 but not more than 0.25, and the dimple height h/H of not less than 0.15 but not more than 0.25.

[0034] Further, in the case where the flatness H/L is not less than 0.05 but not more than 0.09, it is desirable to assure the dimple position d1/L of not less than 0.2 but not more than 0.25, the dimple length A/L of not less than 0.07 but not more than 0.12, the dimple pitch d2/L of not less than 0.2 but not more than 0.23, and the dimple height h/H of not less than 0.18 but not more than 0.2.

[0035] Incidentally, as shown in Fig. 2, the long diameter L and the short diameter H are measured in terms of the outer diameters of the tube 110, the distance dl is the length from the inner wall surface of the partitioning wall 111 to the center of the dimple 112, and the distance d2 between a plurality of the dimples 112 is measured as a center distance between the dimples 112.

[0036] According to this embodiment, the flow velocity is increased by the partitioning wall 111 and a turbulent flow of the cooling water is obtained in the tube 110 in collaboration with the agitating action of the dimples 112. In the case where the flow velocity of the cooling water flowing into the tube 110 is excessively low, however, it is difficult to obtain a turbulent flow of the cooling water even in this embodiment.

[0037] In the case where the flow velocity of the cooling water flowing into the tube 110 is excessively high, on the other hand, the pressure loss of the tube 110 excessively increases so that the radiation capacity is liable to decrease.

[0038] As the result of studying this problem, the inventors have reached the conclusion that, as shown in Fig. 7, the heat exchanger is desirably used in a state where the flow velocity of the cooling water is approximately not lower than 1.5 m/sec but not higher than 6 m/sec. Incidentally, the heat release is expressed as the ratio thereof to that of a simple flat, smooth tube in the absence of the dimples 112 and the partitioning wall 111.

[0039] According to this embodiment having the partitioning wall 111, the internal long diameter of the tube 110 is substantially reduced, and therefore the tube 110 is prevented from being deformed in the direction along the short diameter thereof. Thus, the reduction in flow velocity and the stress on the fins 120 which otherwise might be caused by the expansive deformation of the tube 110 along the short diameter thereof are alleviated. Therefore, the durability (reliability) of the radiator 100 can be improved while at the same time preventing the radiation capacity from being deteriorated.

[0040] Also, in view of the fact that the partitioning wall 111 is formed integrally of a single plate material, the increase in the production cost of the tube 110 can be suppressed.

(Other embodiments)

[0041] According to the embodiment described above, the interior of the tube 110 is segmented into the two spaces 110b, 110c having the same volume (L1 = L2). As an alternative, the volumes of the two spaces 110b, 110c may be differentiated from each other.

[0042] Further, unlike in the embodiment described above, the interior of the tube 110 may be segmented into three or more spaces instead of the two spaces 110b, 110c.

[0043] Furthermore, the shape of the partitioning wall 111 is not limited to the one shown in Fig. 2 but, as shown in Fig. 8, for example, the partitioning wall 111b may be omitted so that the partitioning wall 111 includes only the partitioning walls 111a, 111c.

[0044] Figs. 9 and 10 show the steps of the process for fabricating the tube 110 according to another embodiment. In this case, the partitioning wall 111 for segmenting the tube 110 into a plurality of spaces 110b is constituted of a groove portion 111d formed on one side of a plate material and an insertion portion 111e formed on the other side thereof. Specifically, as shown in Fig. 9 (b), a rise portion 111f is formed on the plate material by a roller (protrusion forming process). Then, as shown in Figs. 9(c), 9(d) and 9(e) in that order, one side and the other side of the plate material are bent thereby to form the groove portion 111d and the insertion portion 111e (end forming process). In the step of Fig. 9(e), protrusions (dimples) 112 are formed at the same time.

[0045] Then, the plate material is bent (curved) as shown in Figs. 10(a), 10(b), 10(c) and 10(d) in that order, so that the insertion portion 111e is assembled by being inserted in the groove portion 111d (insertion forming process).

[0046] By doing so, the tube 110 having the partitioning wall 111 is formed.

[0047] The present invention is not confined to the aforementioned embodiments in which the dimples 112 are formed in staggered arrangement on the two opposed sides of the inner wall along the direction of the short diameter of the tube 110. As an alternative, the dimples 112 may be formed only on one side of the inner wall of the tube 110.

[0048] While the invention has been described by reference to specific embodiments chosen for purposes of illustration, it should be apparent that numerous modifications could be made thereto by those skilled in the art without departing from the basic concept and scope of the invention.

Claims

1. A heat exchanger for cooling the coolant of a liquid-cooled internal combustion engine for automotive vehicles by exchanging heat between the coolant and the air, comprising:

a plurality of tubes (110) having a flat section in which the coolant flows;
a plurality of cooling fins (120) arranged between said tubes (110) for increasing the radiation area by contacting the outer surface of the tubes (110);
a first header tank (130) arranged at a longitudinal end of the tubes (110) for supplying by distributing the coolant to the tubes (110); and
a second header tank (140) arranged at the other longitudinal end of the tubes (110) for recovering by collecting the coolant flowing out from the tubes (110);
wherein the tubes (110) each have therein a partitioning wall (111) for partitioning the internal space (110a) of the tube (110) into a plurality of spaces (110b, 110c) along the length of the tube (110), and the inner wall of the tubes (110) is formed with a plurality of protrusions (112) inward of the tube (110).

2. A heat exchanger according to claim 1, wherein

the ratio of the shorter diameter (H) to the long diameter (L) of the tube (110) is not less than 0.035 but not more than 0.1,
the ratio of the distance (d1) from the partitioning wall (111) to a given protrusion (112) to the long diameter (L) of the tube (110) is not less than 0.15 but not more than 0.3,
the ratio of the size (A) of the portion of the protrusion (112) parallel to the tubes (110) to the long diameter (L) of the tube (110) to the long diameter (L) of said tube (110) is not less than 0.05 but not more than 0.15,
the ratio of the distance (d2) between the protrusions (112) to the long diameter (L) of the tube (110) is not less than 0.15 but not more than 0.25, and
the ratio of the protruded length (h) of the protrusion (112) to the short diameter (H) of the tube (110) is not less than 0.15 but not more than 0.25.

3. A heat exchanger according to claim 1, wherein

the ratio of the short diameter (H) to the long diameter (L) of each of the tubes (110) is not less than 0.05 but not more than 0.09,
the ratio of the distance (d1) between the partitioning wall (111) and a given one of the protrusions (112) to the long diameter (L) of the tube (110) is not less than 0.2 but not more than 0.25,
the ratio of the size (A) of the portion of the protrusion (112) parallel to the long diameter of the tubes (110) to the long diameter (L) of the tubes (110) is not less than 0.07 but not more than 0.12,
the ratio of the distance (d2) between the protrusions (112) to the long diameter (L) of the tube (110) is not less than 0.2 but not more than 0.23, and
the ratio of the protruded length (h) of the protrusions (112) to the short diameter (H) of the tubes (110) is not less than 0.18 but not more than 0.2.

Fig.1

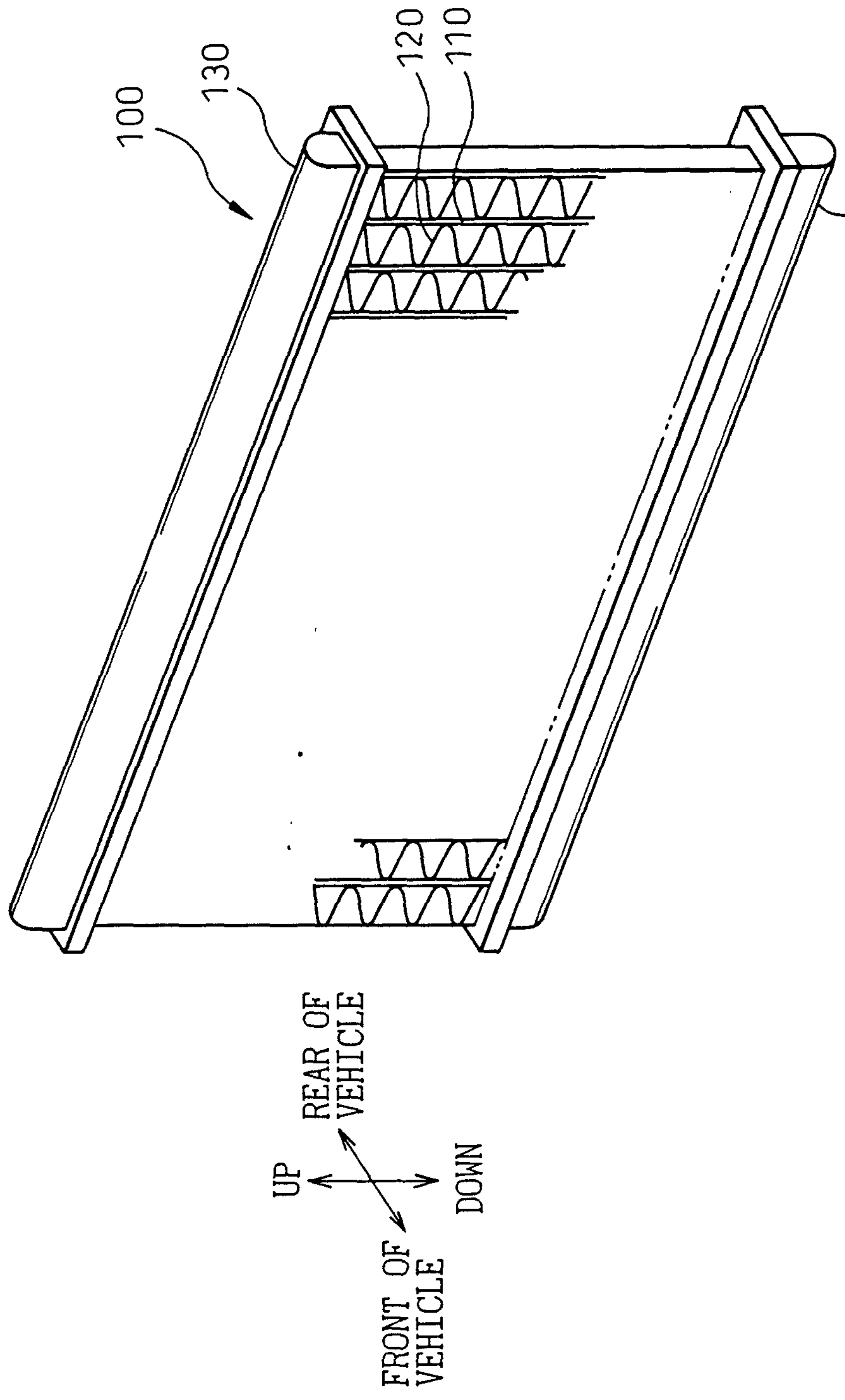


Fig.2

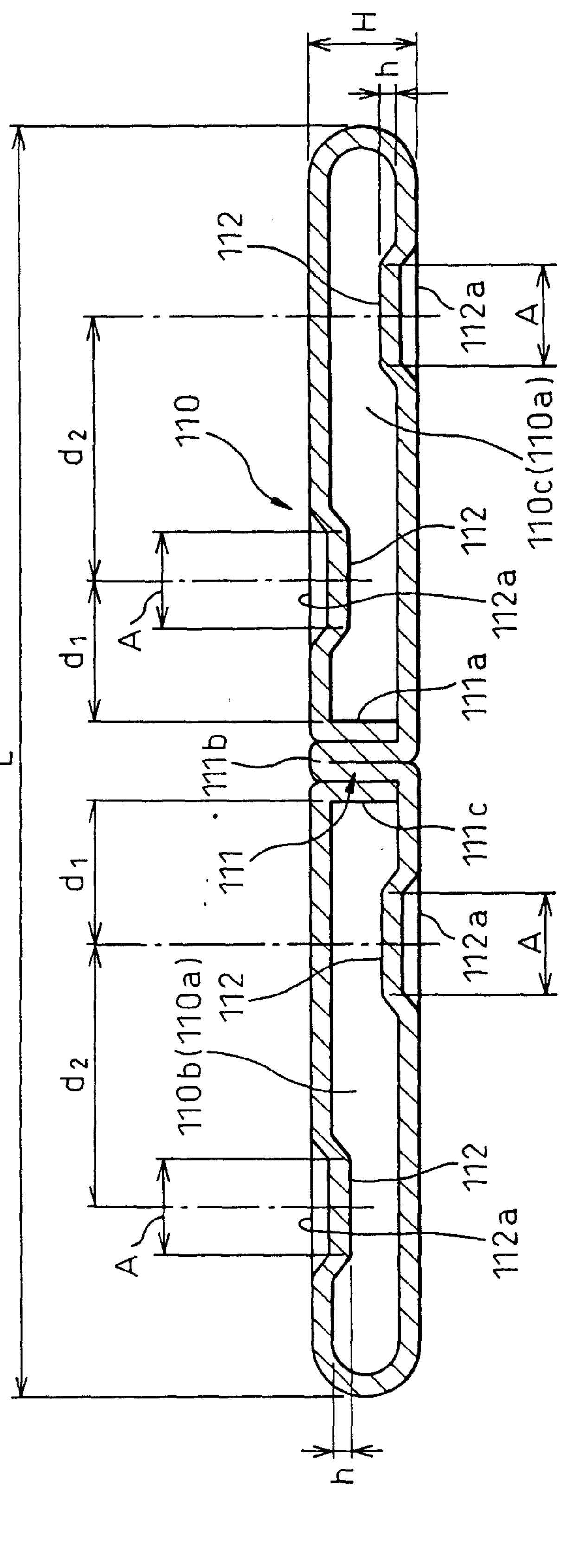


Fig.3

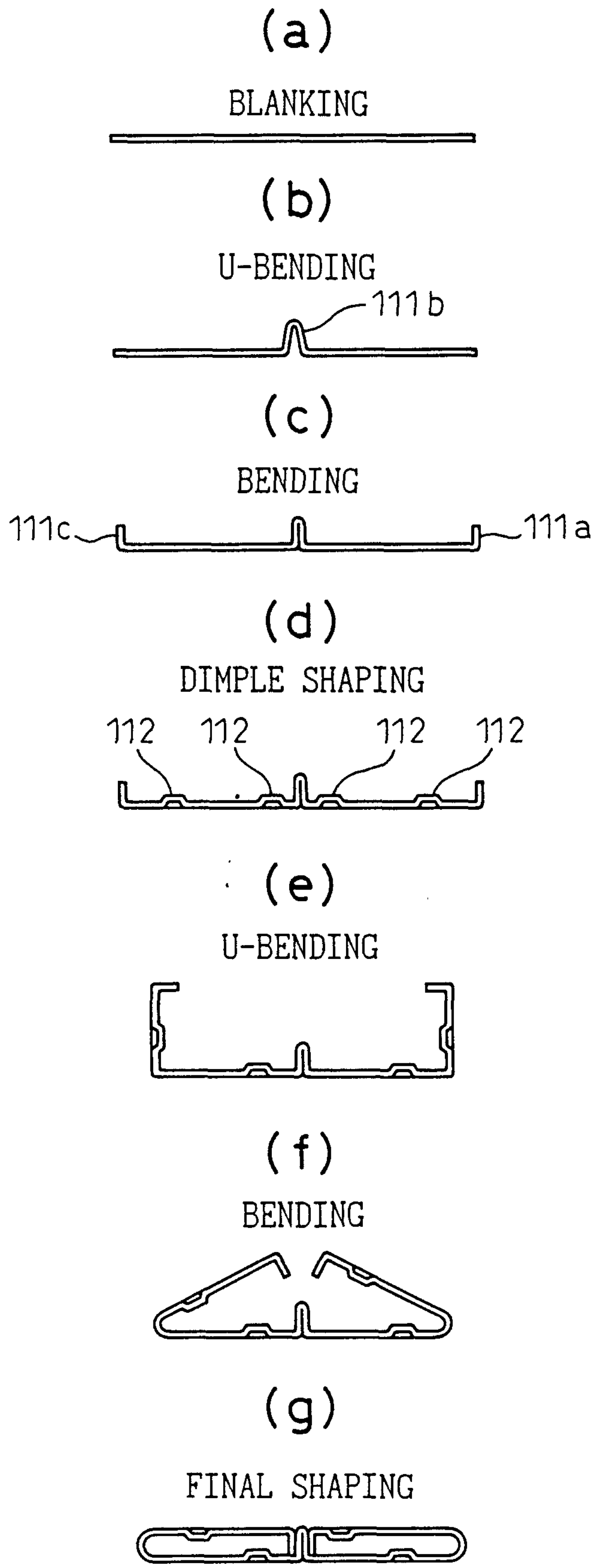

(a)
BLANKING
(b)
U-BENDING
111b
(c)
BENDING
111c
111a
(d)
DIMPLE SHAPING
112
112
112
112
(e)
U-BENDING
(f)
BENDING
(g)
FINAL SHAPING

Fig.4

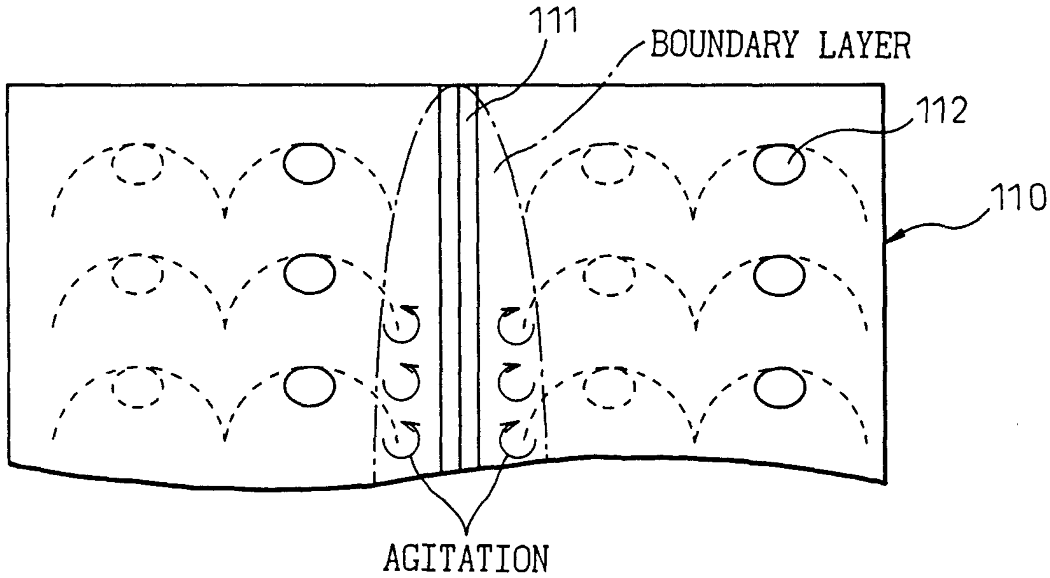


Fig.5

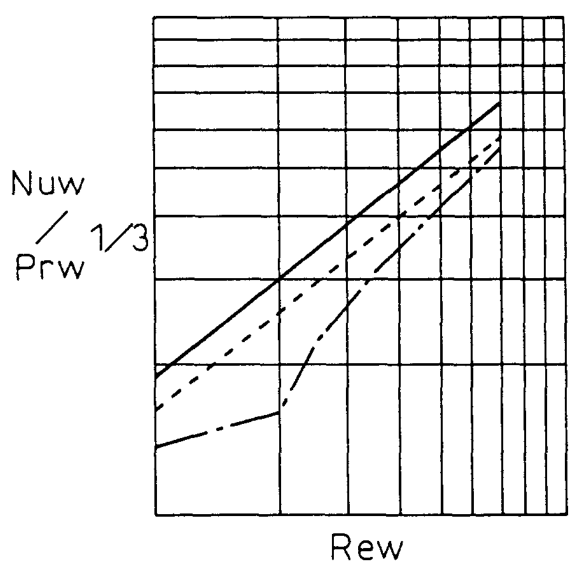


Fig.6A

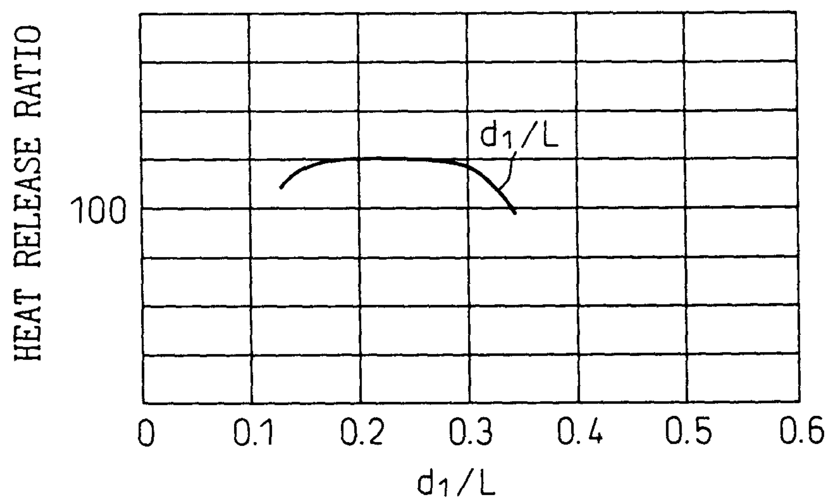

HEAT RELEASE RATIO
100
d1/L
0
0.1
0.2
0.3
0.4
0.5
0.6
d1/L

Fig.6B

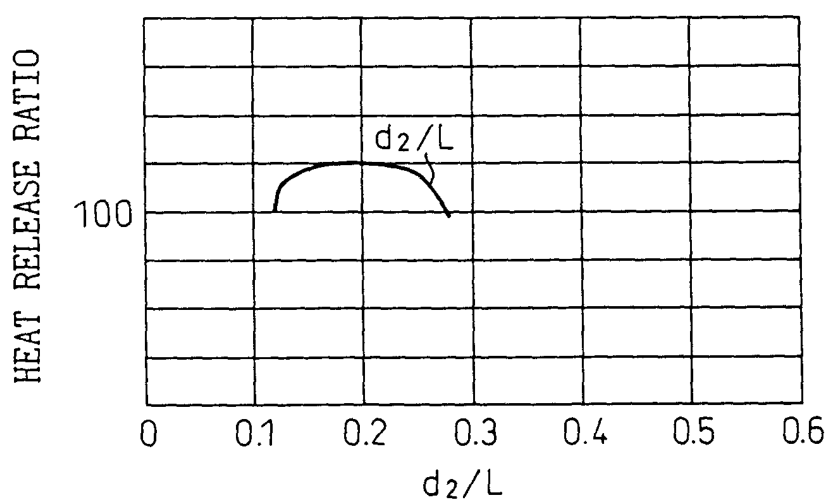
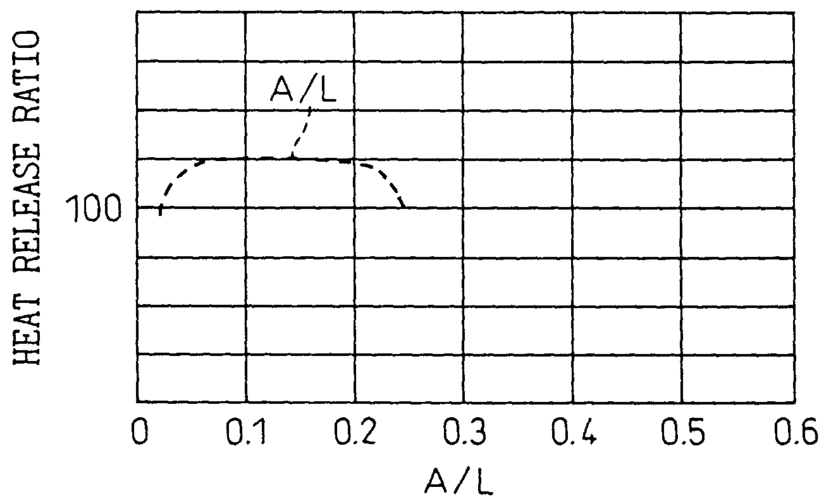

HEAT RELEASE RATIO
100
d2/L
0
0.1
0.2
0.3
0.4
0.5
0.6
d2/L

Fig. 6C


HEAT RELEASE RATIO
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A/L
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0.1
0.2
0.3
0.4
0.5
0.6
A/L

Fig. 6D

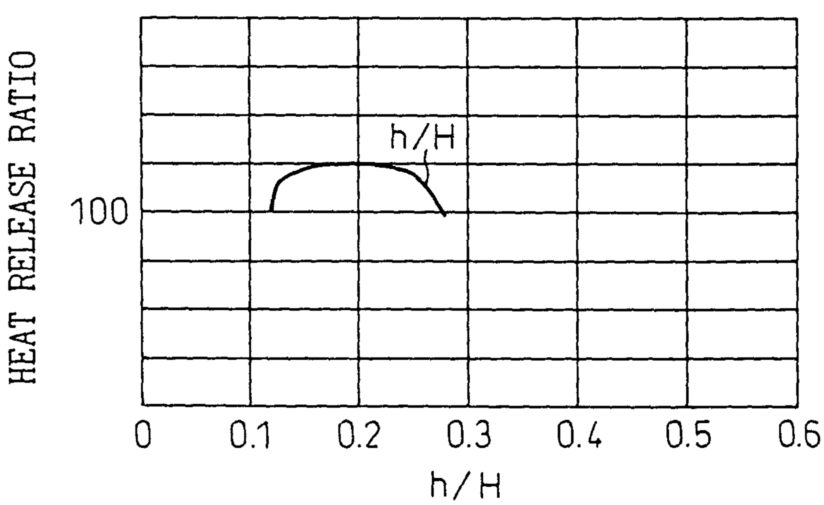

HEAT RELEASE RATIO
100
h/H
0
0.1
0.2
0.3
0.4
0.5
0.6
h/H

Fig.7

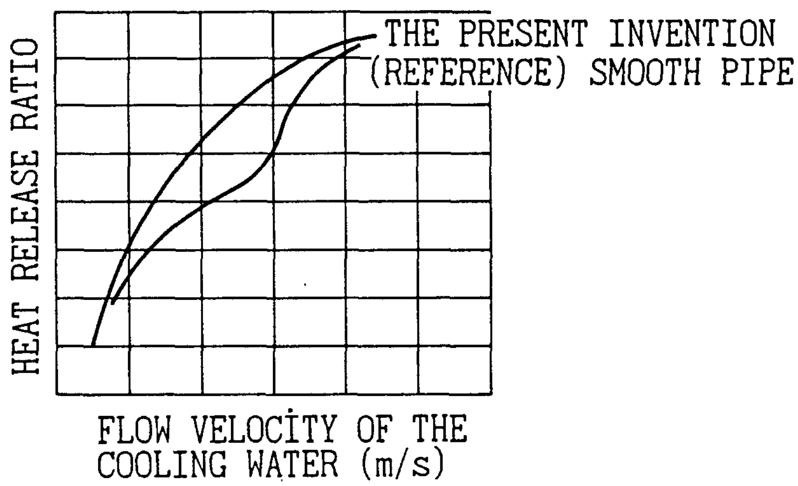

THE PRESENT INVENTION
(REFERENCE) SMOOTH PIPE
HEAT RELEASE RATIO
FLOW VELOCITY OF THE COOLING WATER (m/s)

Fig. 8

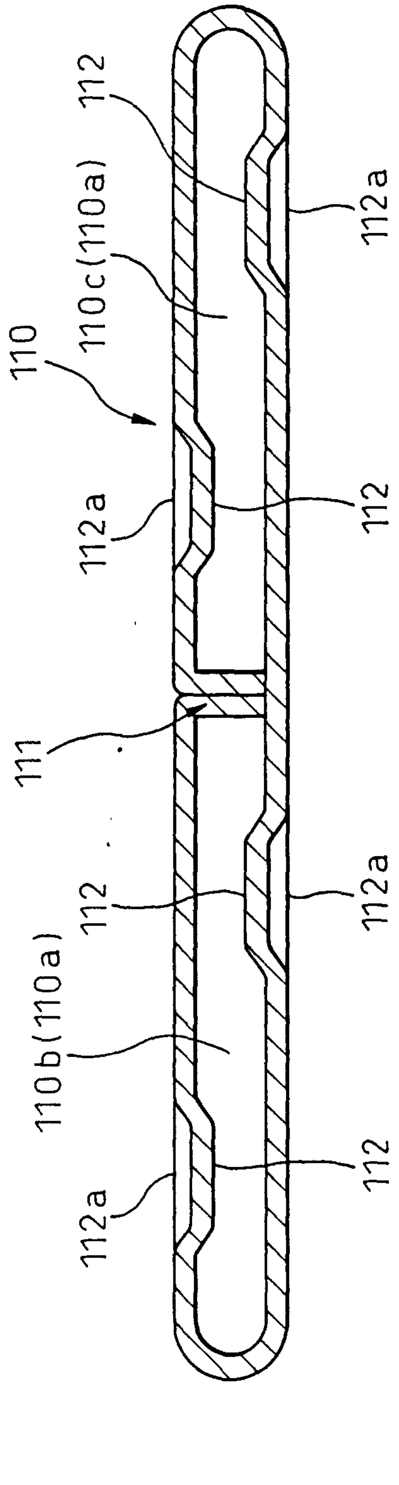

112
112a
110c(110a)
110
112a
112
111
112
112a
110b(110a)
112a
112

Fig.9

(a)

(b)

(c)

(d)

(e)

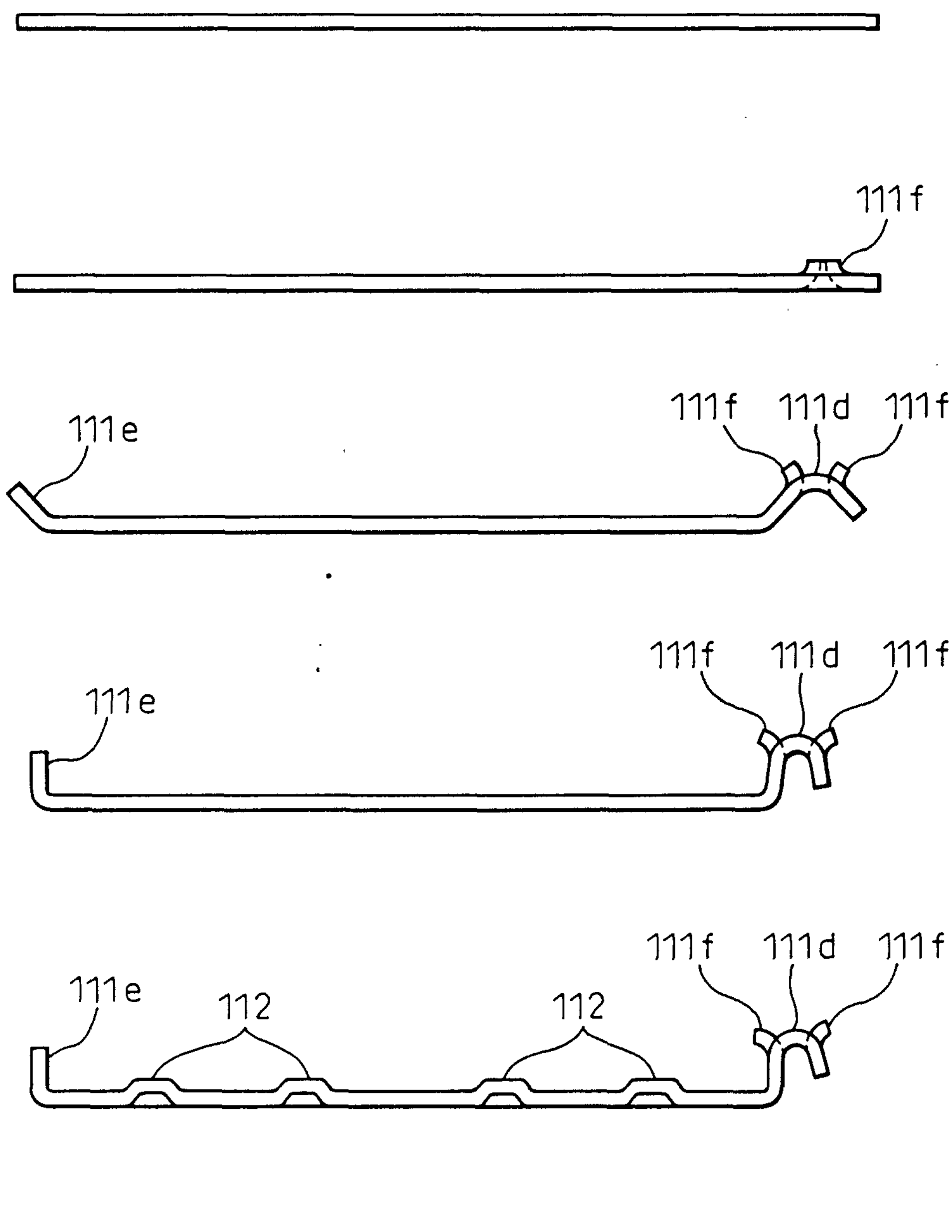
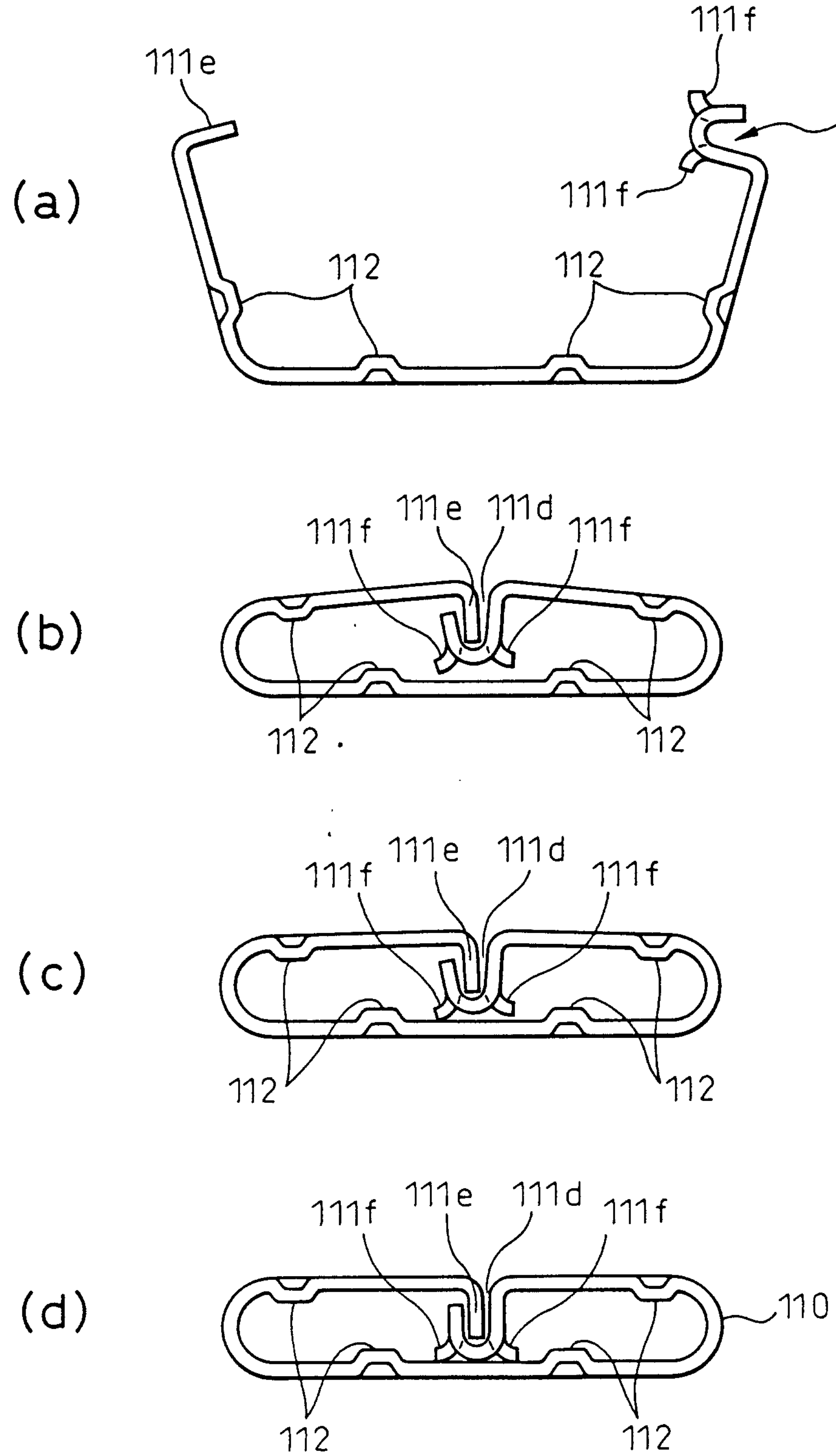

111f
111e
111f
111d
111f
111e
111f
111d
111f
111e
112
112
111f
111d
111f

Fig.10


(a)
111e
111f
111d
111f
112
112
(b)
111f
111e
111d
111f
112
112
(c)
111f
111e
111d
111f
112
112
(d)
111f
111e
111d
111f
110
112
112