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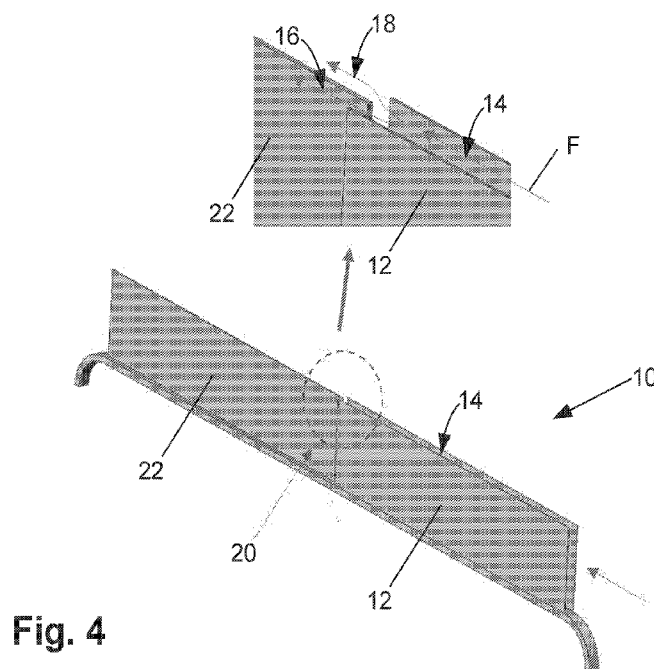


Fig. 4

(57) Abstract: An air cooled condenser fin comprises flow channel walls defining an air flow channel. Flow channel walls include planar sections separated by intermittent flow interruptions which are spaced apart along the air flow channel. Intermittent flow interruptions are defined by the flow channel walls. Intermittent flow interruptions may for example comprise splits formed by a staggered arrangement in which the planar sections of the flow channel walls before and after each split are staggered; or intermittent sinusoidal waves formed into the flow channel walls; or louvers formed into the flow channel walls to create openings passing through the flow channel walls at the louvers. Intermittent flow interruptions may be spaced apart along the air flow channel by at least 5 hydraulic diameters, and in some embodiments by 5-10 hydraulic diameters. A plurality of such air cooled condenser fins are suitably employed with the air flow channels arranged in parallel.



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## PERTURBING AIR COOLED CONDENSER FIN

**[0001]** This application claims the benefit of U.S. Provisional Application No. 62/835,706 filed April 18, 2019. U.S. Provisional Application No. 62/835,706 filed April 18, 2019 is incorporated herein by reference in its entirety.

### BACKGROUND

**[0002]** The following relates to air cooled condensers, heat exchanger fins and arrays thereof for air cooled condensers, and so forth.

**[0003]** Traditional heat exchange fins are of the straight or plain variety. Heat transfer occurs through a fin channel wherein air enters in the channel and establishes a fluid boundary layer. Heat transfer normalizes once the fluid boundary layer is established. Some straight fins may have blemishes or other features on their surface as a result of manufacturing processes. As such traditional fin designs may not be entirely straight nor uniform from fin to fin within a fin array.

**[0004]** A known modification of the straight fin design is to include perforations in the fins to disturb air flow with the flow channel, i.e. pressed or cut perforations. They offer cross channel flow paths and are commonly used in automotive applications. The perforations, while effective at disturbing the fluid boundary layer increasing heat transfer, can be impractical in certain applications as they increase the pressure drop of the air flow.

**[0005]** Another known fin design employs offset fins. In this design the straight path is cyclically offset within a channel to disturb the boundary layer. This design, as with the perforations, increase heat transfer capability but in doing so also increases pressure drop.

**[0006]** Another known fin design employs wavy or ruffled fins. In this design the straight fins are curved to form sinusoidal waves. The periodic waves enables disruption of the fluid boundary layer. Haushalter, U.S. Patent No. 5,209,289 issued May 11, 1993, discloses a modified fin array incorporating wavy offsets in a unique combination.

**[0007]** Bugler et al., U.S. Publication No. 2018/0023901 A1 published January 25, 2018, discloses a heat exchange tube fin design in which a plurality of arrowhead shapes are pressed into or embossed onto each fin. The pressed arrowhead shapes

are grouped into nested pairs, and one of the arrowheads in a pair is pressed as a positive relative to the fin plane and the other of the pair is pressed as a negative relative to the fin plane. The arrowhead pairs are placed in rows parallel to the air flow direction and arrowhead pairs in one row are preferably staggered relative to the arrowhead pairs in the adjacent row along the fin in the air flow direction.

## BRIEF SUMMARY

**[0008]** In some aspects disclosed herein, an air cooled condenser fin comprises flow channel walls defining an air flow channel. The flow channel walls include planar sections separated by intermittent flow interruptions which are spaced apart along the air flow channel. The intermittent flow interruptions are defined by the flow channel walls.

**[0009]** In some illustrative embodiments, the intermittent flow interruptions comprise splits formed by a staggered arrangement in which the planar sections of the flow channel walls before and after each split are staggered. In some embodiments, the staggering of the flow channel walls after each split is about one-half of a width of the air flow channel.

**[0010]** In some illustrative embodiments, the intermittent flow interruptions comprise intermittent sinusoidal waves formed into the flow channel walls.

**[0011]** In some illustrative embodiments, the intermittent flow interruptions comprise louvers formed into the flow channel walls to create openings passing through the flow channel walls at the louvers. In some embodiments, the louvers are angled between 1 degree and 30 degrees to an air flow direction of the air flow channel. In some embodiments, the flow channel walls are secured to a tube of an air cooled condenser.

**[0012]** In any of the foregoing embodiments, the intermittent flow interruptions are in some more specific embodiments spaced between 5 hydraulic diameters and 10 hydraulic diameters apart along the air flow channel.

**[0013]** In any of the foregoing embodiments, the intermittent flow interruptions are in some more specific embodiments spaced apart along the air flow channel by at least 5 hydraulic diameters.

**[0014]** In some aspects disclosed herein, a plurality of air cooled condenser fins as set forth in any of the preceding paragraphs is provided, in which the air flow channels of the air cooled condenser fins are arranged in parallel.

**[0015]** In some aspects disclosed herein, an air cooled condenser comprises steam/condensate tubes and fins attached to the steam/condensate tubes. The fins comprise flow channel walls defining parallel air flow channels. The flow channel walls include planar sections separated by intermittent flow interruptions which are spaced apart along the air flow channels. The intermittent flow interruptions are defined by the flow channel walls. The intermittent flow interruptions in some embodiments comprise splits formed by a staggered arrangement in which the planar sections of the flow channel walls before and after each split are staggered. The staggering of the flow channel walls after each split is, in some more specific embodiments, about one-half of a width of the air flow channel. The intermittent flow interruptions in some embodiments comprise intermittent sinusoidal waves formed into the flow channel walls. The intermittent flow interruptions in some embodiments comprise louvers formed into the flow channel walls to create openings passing through the flow channel walls at the louvers. The louvers are, in some more specific embodiments, angled between 1 degree and 30 degrees to an air flow direction of the air flow channel. In any of the foregoing embodiments of this paragraph, the intermittent flow interruptions may in some more specific embodiments be spaced between 5 hydraulic diameters and 10 hydraulic diameters apart along the air flow channel. In any of the foregoing embodiments of this paragraph, the intermittent flow interruptions may in some more specific embodiments be spaced apart along the air flow channel by at least 5 hydraulic diameters.

**[0016]** Some embodiments of an air cooled condenser as set forth in the immediately preceding paragraph further comprise distribution headers connected to feed steam into the steam/condensate tubes, and an air moving system comprising a fan arranged to drive an airflow across the fins attached to the steam/condensate tubes. Some more specific embodiments further include risers connected to feed the steam into the distribution headers, wherein the steam/condensate tubes, the distribution headers, the risers, and the air moving system are arranged to form the air cooled condenser as an A-frame type air cooled condenser or other types.

**[0017]** In some aspects disclosed herein, a method of cooling using an air cooled condenser fin is disclosed. The method comprises flowing air through an air

flow channel defined by flow channel walls, and interrupting the flowing of air at intermittent flow interruptions defined by the flow channel walls which are spaced apart along the air flow channel. In some more specific embodiments, the intermittent flow interruptions are placed at locations where a boundary layer of the flowing air has normalized. In some more specific embodiments, the intermittent flow interruptions are spaced apart along the air flow channel by at least 5 hydraulic diameters.

**[0018]** These and other non-limiting aspects and/or objects of the disclosure are more particularly described below.

## BRIEF DESCRIPTION OF THE DRAWINGS

**[0019]** The invention may take form in various components and arrangements of components, and in various process operations and arrangements of process operations. The drawings are only for purposes of illustrating preferred embodiments and are not to be construed as limiting the invention. This disclosure includes the following drawings.

**[0020]** FIGURE 1 diagrammatically shows differential temperature versus length for a planar fin.

**[0021]** FIGURE 2 diagrammatically shows heat transfer coefficient versus length for a planar fin.

**[0022]** FIGURE 3 diagrammatically shows incremental air pressure drop versus length for a planar fin.

**[0023]** FIGURE 4 diagrammatically shows a perspective view of a portion of an air cooled condenser fin having a single split-fin.

**[0024]** FIGURE 5 diagrammatically shows a top view of a portion of an air cooled condenser fin array including a plurality of splits of the type shown in FIGURE 4.

**[0025]** FIGURE 6 diagrammatically shows a perspective view air flow velocity map for the split fin air flow.

**[0026]** FIGURE 7 diagrammatically shows a perspective view of a portion of an air cooled condenser fin having intermittent sinusoidal waves.

**[0027]** FIGURE 8 diagrammatically shows a perspective view of a portion of an air cooled condenser fin having louvers.

**[0028]** FIGURE 9 diagrammatically shows a top view of a portion of an air cooled condenser fin array including a plurality of louvers of the type shown in FIGURE 8.

**[0029]** FIGURE 10 plots heat transfer coefficient of an air cooled condenser fin with splits, versus position.

**[0030]** FIGURE 11 plots heat transfer coefficient of an air cooled condenser fin with intermittent sinusoidal waves, versus position.

**[0031]** FIGURE 12 diagrammatically shows a typical air cooled condenser application of the disclosed improved fins, where the illustrative air cooled condenser is of the forced draft A-frame type.

## DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

**[0032]** For certain applications, such as air-cooled condensers, excessive pressure drop creates design constraints when applying fins to a tube geometry. A need exists for new and improved fin and fin array designs that minimize heat exchanger footprint opposite need for additional power input needed to move air/overcome excessive pressure drops of existing designs.

**[0033]** Air-cooled condenser applications have some requirements regarding the steam flow area and resultant pressure drop that places constraints on the minimum sizing of the heat exchanger fin tube base. Use of existing non-planar fin designs (Louvered, Offset, Wavy, etc.) requires additional power input (larger air mover sizing) to apply the fin to the required tube geometry. Further, redesigning tube geometry to incorporate a non-planar fin design in a lower power input design is often uneconomical.

**[0034]** Flow on the air-side of an air-cooled condenser generally operates in the laminar regime, which is defined by Reynolds numbers less than 2000. In this regime, momentum and energy transport occur via the mechanism of molecular diffusion, which is driven by gradients in the velocity profile. The velocity gradients near the fin wall are especially important in determining the momentum and energy transport rates in the local region as the air flows through the fin channel. As the air in the free-stream approach region enters the fin channels, extremely high velocity gradients result, based on the large velocity differential between the entering air velocity and the zero-velocity condition at the fin wall. This results in large friction factors and heat transfer

coefficients at the lead edge of the fin. As the flow progresses down the fin channel, the velocity profile approaches the fully developed profile (generally parabolic). As this transition occurs, the local velocity gradients at the fin wall are reduced, and the local friction factor and heat transfer coefficient values gradually approach the fully developed values. This transition often occurs within ten (10) hydraulic diameters from the entrance to the fin section. The hydraulic diameter,  $D_H$ , is a commonly used term when handling flow in non-circular tubes and channels, and is defined as

$$D_H = \frac{4A}{P} \quad (1)$$

where  $A$  is the cross-sectional area of the flow and  $P$  is the wetted perimeter of the cross-section (where the wetted perimeter includes all surfaces acted upon by shear stress from the fluid). For a closed rectangular channel of dimensions  $a \times b$ , the hydraulic diameter  $D_H$  is given by:

$$D_{H,rect} = \frac{4A}{P} = \frac{4(a \times b)}{a + b + a + b} = \frac{2ab}{a + b} \quad (2)$$

The entrance region of the fin is therefore more effective in terms of heat transfer than the remainder of the fin, although the increase in heat transfer comes at the cost of added pressure drop.

**[0035]** With reference to FIGURES 1-3, numerical simulations of differential temperature versus length (FIGURE 1), heat transfer coefficient versus length (FIGURE 2), and incremental air pressure drop versus length (FIGURE 3) are shown for a planar fin. In FIGURES 1, 2, and 3, the entrance region is to the left, with the air flow proceeding along the channel of length  $L$  to the exit side of the fin on the right of the curve. The most effective region of heat transfer in a planar fin of an air cooled condenser is at the air entrance into the fin channel. This is evident in FIGURE 2 where the heat transfer coefficient decreases rapidly as air flows along the channel length.

**[0036]** FIGURE 4 is a graphic representation showing a single split-fin **10** as disclosed herein. The split-fin takes advantage of the enhanced heat transfer which

occurs at the lead edge of a fin channel entrance to reduce the total surface area of the overall fin assembly. The split-fin arrangement of FIGURE 4 comprises a straight fin section **12** having a fin channel **14** that is subsequently split into two fin channels **16**, **18** at a split point **20** along the length of the fin channel **14**. The split **20** preferably occurs at or around the point where air flow has fully developed in the first channel section **14**. As air flow (indicated by arrows **F** in FIGURE 4) enters the staggered fin arrangement **20**, it is split into subsequent sections, i.e. split air flows **16**, **18**. The fin wall of the downstream fin section **22** is located at or near the center of the upstream flow path or channel **14** (in other words, the staggering of the fin walls after the split is about one-half of a width of the air flow channel), thereby being exposed to high velocity gradients near the wall, similar to what occurs at the entrance of the first fin section **12**. This process is repeated at each flow split, resulting in increased heat transfer at the lead edge region of every fin section **12**, **22**. A more compact fin assembly configuration results; requiring less fin surface area and comparatively less material needed to construct the fin array than traditional designs.

**[0037]** With reference to FIGURE 5, when the split-fin is formed into a fin array, each flow split results in increased heat transfer coefficient and friction factor in the local region near the respective fin section entrance. FIGURE 5 illustrates a fin array including successive illustrated fin sections **12a**, **12b**, **12c** with respective channels **14a**, **14b**, **14c**, with fin split point **20ab** at the junction of fin sections **12a**, **12b** and fin split point **20bc** at the junction of fin sections **12b**, **12c**. This is repeated for as many fin split sections exist in the assembly. The local increases in friction factor result in an increase in pressure drop, which is generally undesirable. Heat transfer from the fin to the air depends on two factors, the local heat transfer coefficient, and the local bulk air-to-fin temperature difference. The flow splits **20** increase the local heat transfer coefficient, which is beneficial. As with friction factor, the local increase in heat transfer coefficient resulting from each flow split **20** is consistent throughout the fin assembly. However, as the air flow **F** proceeds through the assembly, the temperature difference between the air and the fin is continuously reduced. Therefore, the effectiveness of the flow splits **20** with respect to increased local heat transfer decreases the farther the particular flow split is from the fin assembly entrance. For this reason, it is beneficial to cluster the flow splits **20** near the entrance to the assembly and use a more continuous section at the trail end of a fin assembly comprising multiple split-fins.

**[0038]** With reference to FIGURE 6, in another embodiment, an air cooled condenser utilizes single row finned tubes and includes a split-feature within the air flow channel which disturbs the boundary layer along the flow channel wall. FIGURE 6 shows a perspective view air flow velocity map for the split fin air flow.

**[0039]** The concepts of split-fins is not intended to be limited by the preceding discussion. Split features may be repeating or intermediate. Flow channel walls may be discontinuous or continuous. Flow along the wall of a planar fin may be perturbed by the channel being cut, and a new channel formed with the opening offset from the outlet of the original channel. Fin channels may consist of single or multiple splits.

**[0040]** Channel length of the fin sections **12** is preferably determined by finding the point along the wall in which the air flow boundary layer approaches fully developed profile. In one embodiment having multiple splits the splits are spaced between about 5 hydraulic diameters and about 10 hydraulic diameters apart.

**[0041]** With reference to FIGURE 7, in an alternative embodiment one or more fin channels in the fin array may include intermittent sinusoidal waves. A graphic representation of a fin **30** having a channel **34** with an intermittent sinusoidal wave **32** is shown in FIGURE 7. Flow **F** along the wall of the channel **34** is redirected by the sinusoidal wave **32** in the transverse direction of air flow **F**. Wave geometry of the sinusoidal wave **32** is designed to optimize full channel recirculation downstream from the disturbance. Planar fin wall is placed between the multiple waves **32** until boundary layer normalizes to reduce pressure drop. The sinusoidal waves **32** are preferably spaced between about 5 hydraulic diameters and about 10 hydraulic diameters apart.

**[0042]** With reference to FIGURES 8 and 9, in yet another embodiment, louvered fins **40** are disclosed. In this embodiment, openings **42** between adjacent fin channels **44** are used. Openings may take the form of louvers **46** angled between about 1 and about 30 degrees to the direction of flow. Figures 8 and 9 provide graphic representations of the openings **42** between adjacent fin channels **44**. As seen in FIGURE 8, the openings **42** do not comprise the entirety of the flow channel wall. While FIGURES 8 and 9 show alignment between openings in adjacent channels, such alignment is not present in some embodiments; rather, in these embodiments the opening may alternatively be offset. FIGURE 8 also illustrates the louvered fins **40** soldered (or otherwise attached) to a tube **48** as is common in the case of an air cooled condenser (where the steam or other fluid being condensed flows through the tube **48**). It is noted that FIGURES 8 and 9 show the louvers in the “pointing downstream”

configuration. In this configuration the tips of the louvers are pointing roughly in the direction of the flow from upstream to downstream. In a variant embodiment, the louvers may be reversed, so as to point “into” the flow in the upstream direction. Either configuration can be employed effectively.

**[0043]** The innovations disclosed herein may be used on a single channel, a combination of channels, and/or combined with one another to form new and unique fin arrays that improve heat transfer over a variety of tube geometries that may be subject to space constraints and otherwise have limitations on ability to overcome pressure drop concerns. Further advantageous is the reduction in materials requirements for fin arrays enabled by the approaches disclosed herein.

**[0044]** FIGURES 10 and 11 plots show simulated data relating to heat transfer coefficient of the split-fin and intermittent wave, respectively, against position. As shown in these figures, the heat transfer peaks at the location of the flow interruption and decreases along the length of the channel as the boundary layer is reestablished and the difference in temperature between the two fluids, air and steam, is gradually reduced.

**[0045]** With reference now to FIGURE 12, a typical air cooled condenser application of the disclosed improved fins is shown. An illustrative air cooled condenser shown in FIGURE 12 is of the forced draft A-frame type. An electric power generator **52** is driven by a steam turbine **54** using steam **56**. Exhaust steam **58** discharged from the steam turbine **54** flows into a main steam duct **60** and a distribution manifold **62**, that distributes the steam to a set of air cooled condensers, an illustrative one of which is shown in FIGURE 12. The steam flows up through risers **64** which are connected to feed the steam into distribution headers **66** which in turn are connected to feed the steam into bundles **70** that include steam/condensate tubes (e.g., the illustrative steam/condensate tube **48** shown in FIGURE 8) with fins (such as split fins **10** as shown in FIGURES 4-6; or fins **30** with intermittent sinusoidal waves **32** as shown in FIGURE 7; or fins **40** with louvers **46** as shown in FIGURES 8 and 9) soldered or otherwise attached to the steam/condensate tubes. An air moving system **72**, such as a fan, drives an airflow across the fins of the bundles **70** in order to cool and condense the steam in the tubes to form condensate. FIGURE 12 further shows the condenser superstructure including a fan deck **74** supported by support structure **76** and bracing **78**, and a windwall structure **80** atop the fan deck **74**. While the illustrative air cooled condenser is of the forced draft A-frame type, the disclosed

improved fins are suitably employed in conjunction with air cooled condensers of other types, such as an induced draft V-frame condenser type, a flat condenser type, or so forth.

**[0046]** The inventors have performed computer simulations of the performance of various designs of split fins **10** (FIGURES 4-6), fins **30** with intermittent sinusoidal waves **32** (FIGURE 7), and fins **40** with louvers **46** (FIGURES 8 and 9). In these simulations, the fins were modeled as rectangular channels with rectangular dimension  $H_{fin}$  being the fin height (that is, the distance the fin extends away from the steam/condensate tube to which it is soldered) and dimension  $S_{fin}$  being the separation between the fin walls defining the air flow channel. Using the hydraulic diameter for a rectangular channel given in Equation (2), this yields the following hydraulic diameter  $D_{H,fin}$  for the fins:

$$D_{H,fin} = \frac{4A}{P} = \frac{2ab}{a+b} = \frac{2(S_{fin} \times H_{fin})}{S_{fin} + H_{fin}} \quad (3)$$

More generally, for an air flow channel of arbitrary cross sectional dimensions the first expression of Equation (3) holds, i.e.  $D_{H,fin} = \frac{4A}{P}$  where  $A$  is the cross-sectional area of the air flow channel and  $P$  is the perimeter of the cross-section of the air flow channel, and with a tube bundle length "L" (also indicated as bundle length **82** if FIGURE 12). The simulations were for a design of the bundles **70** that included 11 fins per inch. The simulations also modeled the energy (in horsepower which is related to pressure loss across a bundle) of the air moving system (e.g. fan) **72** and the bundle tube length **82**. In addition to modeling performance of the designs of the split fins **10**, fins **30** with intermittent sinusoidal waves **32**, and louvered fins **40**, simulations were performed to model the performance of conventional planar fin and continuous wavy fin designs. The simulations concluded that utilizing the planar fin design with the intermittent perforations, i.e., split fins **10**, fins **30** with intermittent sinusoidal waves **32**, and louvered fins **40** would enable reduction of the tube bundle length **82** by approximately 10% with only a moderate increase of 30% in the fan energy consumption (horsepower). The cost savings in reducing the tube bundle length by 10% is order magnitude greater than the cost of higher energy fan. On the other hand,

the bundle with the continuous wavy fin design would reduce the tube bundle length by 16% but at the price of 700% increase in energy consumption which is prohibitive.

**[0047]** These simulations confirm the mechanism for improved performance disclosed herein, namely that employing mostly planar fins but with intermittent flow interruptions positioned at points where the boundary layer normalizes can achieve the desired heat transfer efficiency improvement while only imposing a modest increase in pressure drop. It was found that the intermittent flow interruptions are in some embodiments preferably spaced between about 5 hydraulic diameters and about 10 hydraulic diameters apart to optimally balance heat transfer efficiency (improved by the intermittent flow interruptions) against pressure drop introduced by the interruptions. The intermittent flow interruptions can be fin splits **20** (as in the embodiments of FIGURES 4-6), intermittent sinusoidal waves **32** (as in the embodiment of FIGURE 7), louvers **46** (as in the embodiments of FIGURES 8 and 9), or more generally any other type of intermittent interruption. The simulations also found that placing the intermittent flow interruptions nearer the entrance side of the fin maximizes the heat transfer benefit while imposing the least additional pressure drop. For example, in some nonlimiting embodiments at least 70% of the intermittent flow interruptions are positioned within the first one-half of the fin length  $L$  (that is, within the half-fin length closest to the entrance side of the fin). In some other nonlimiting embodiments, at least 80% of the intermittent flow interruptions are positioned within the first one-third of the fin length  $L$  (that is, within the first third of the fin length that is closest to the entrance side of the fin).

**[0048]** It should be noted that the term “planar fin” is used herein in its usual and ordinary meaning in the art, as a fin that channels air flow principally along a single planar channel. In a planar section of a fin, the flow channel walls defining the air flow channel may have some deviations from geometrically perfect planar form, for example due to unintended manufacturing-induced variations, dimples, wall curvature, or so forth. Such a imperfections typically do not have a meaningful impact on air flow and hence are considered “planar” fin sections as used herein. Likewise, the term “intermittent flow interruption” as used herein is an intentional (i.e. design-basis) modification to a fin wall or walls, or a fin split, that is sufficient to induce air flow interruption as described herein. Hence, unintended manufacturing-induced variations, dimples, wall curvature, or so forth are not considered “intermittent flow interruptions” as used herein.

**[0049]** Illustrative embodiments including the preferred embodiments have been described. While specific embodiments have been shown and described in detail to illustrate the application and principles of the invention and methods, it will be understood that it is not intended that the present invention be limited thereto and that the invention may be embodied otherwise without departing from such principles. In some embodiments of the invention, certain features of the invention may sometimes be used to advantage without a corresponding use of the other features. Accordingly, all such changes and embodiments properly fall within the scope of the following claims. Obviously, modifications and alterations will occur to others upon reading and understanding the preceding detailed description. It is intended that the present disclosure be construed as including all such modifications and alterations insofar as they come within the scope of the appended claims or the equivalents thereof.

## CLAIMS

We claim:

1. An air cooled condenser fin comprising:  
flow channel walls defining an air flow channel;  
wherein the flow channel walls include planar sections separated by intermittent flow interruptions which are spaced apart along the air flow channel;  
wherein the intermittent flow interruptions are defined by the flow channel walls.
2. The air cooled condenser fin of claim 1 wherein the intermittent flow interruptions comprise splits formed by a staggered arrangement in which the planar sections of the flow channel walls before and after each split are staggered.
3. The air cooled condenser fin of claim 2 wherein the staggering of the flow channel walls after each split is about one-half of a width of the air flow channel.
4. The air cooled condenser fin of claim 2 wherein the splits are spaced between 5 hydraulic diameters and 10 hydraulic diameters apart along the air flow channel.
5. The air cooled condenser fin of claim 2 wherein the splits are spaced apart along the air flow channel by at least 5 hydraulic diameters.
6. The air cooled condenser fin of claim 1 wherein the intermittent flow interruptions comprise intermittent sinusoidal waves formed into the flow channel walls.
7. The air cooled condenser fin of claim 6 wherein the intermittent sinusoidal waves are spaced between 5 hydraulic diameters and 10 hydraulic diameters apart along the air flow channel.
8. The air cooled condenser fin of claim 6 wherein the intermittent sinusoidal waves are spaced apart along the air flow channel by at least 5 hydraulic diameters.

9. The air cooled condenser fin of claim 1 wherein the intermittent flow interruptions comprise louvers formed into the flow channel walls to create openings passing through the flow channel walls at the louvers.

10. The air cooled condenser fin of claim 9 wherein the louvers are angled between 1 degree and 30 degrees to an air flow direction of the air flow channel.

11. The air cooled condenser fin of claim 9 wherein the flow channel walls are secured to a tube of an air cooled condenser.

12. The air cooled condenser fin of claim 9 wherein the louvers are spaced between 5 hydraulic diameters and 10 hydraulic diameters apart along the air flow channel.

13. The air cooled condenser fin of claim 9 wherein the louvers are spaced apart along the air flow channel by at least 5 hydraulic diameters.

14. The air cooled condenser fin of claim 1 wherein the intermittent flow interruptions are spaced between 5 hydraulic diameters and 10 hydraulic diameters apart along the air flow channel.

15. The air cooled condenser fin of claim 1 wherein the intermittent flow interruptions are spaced apart along the air flow channel by at least 5 hydraulic diameters.

16. An air cooled condenser comprising a plurality of air cooled condenser fins as set forth in claim 1 wherein the air flow channels of the air cooled condenser fins are arranged in parallel.

17. An air cooled condenser comprising:  
steam/condensate tubes;  
fins attached to the steam/condensate tubes;

wherein the fins comprise flow channel walls defining parallel air flow channels, the flow channel walls including planar sections separated by intermittent flow interruptions which are spaced apart along the air flow channels;

wherein the intermittent flow interruptions are defined by the flow channel walls.

18. The air cooled condenser of claim 17 wherein the intermittent flow interruptions comprise splits formed by a staggered arrangement in which the planar sections of the flow channel walls before and after each split are staggered.

19. The air cooled condenser of claim 18 wherein the staggering of the flow channel walls after each split is about one-half of a width of the air flow channel.

20. The air cooled condenser of claim 17 wherein the intermittent flow interruptions comprise intermittent sinusoidal waves formed into the flow channel walls.

21. The air cooled condenser of claim 17 wherein the intermittent flow interruptions comprise louvers formed into the flow channel walls to create openings passing through the flow channel walls at the louvers.

22. The air cooled condenser of claim 21 wherein the louvers are angled between 1 degree and 30 degrees to an air flow direction of the air flow channel.

23. The air cooled condenser of claim 17 wherein the intermittent flow interruptions are spaced between 5 hydraulic diameters and 10 hydraulic diameters apart along the air flow channel.

24. The air cooled condenser of claim 23 wherein the air flow channels are rectangular with the hydraulic diameter  $D_{H,fin}$  of each air flow channel being:

$$D_{H,fin} = \frac{2(S_{fin} \times H_{fin})}{S_{fin} + H_{fin}}$$

where  $H_{fin}$  is a fin height and  $S_{fin}$  is a separation between the fin walls defining each air flow channel.

25. The air cooled condenser of claim 23 wherein the hydraulic diameter  $D_{H,fin}$  of each air flow channel is:

$$D_{H,fin} = \frac{4A}{P}$$

where  $A$  is the cross-sectional area of each air flow channel and  $P$  is the perimeter of the cross-section of each air flow channel.

26. The air cooled condenser of claim 17 wherein the intermittent flow interruptions are spaced apart along the air flow channel by at least 5 hydraulic diameters.

27. The air cooled condenser of claim 26 wherein the air flow channels are rectangular with the hydraulic diameter  $D_{H,fin}$  of each air flow channel being:

$$D_{H,fin} = \frac{2(S_{fin} \times H_{fin})}{S_{fin} + H_{fin}}$$

where  $H_{fin}$  is a fin height and  $S_{fin}$  is a separation between the fin walls defining each air flow channel.

28. The air cooled condenser of claim 26 wherein the hydraulic diameter  $D_{H,fin}$  of each air flow channel is:

$$D_{H,fin} = \frac{4A}{P}$$

where  $A$  is the cross-sectional area of each air flow channel and  $P$  is the perimeter of the cross-section of each air flow channel.

29. The air cooled condenser of claim 17 further comprising:  
distribution headers connected to feed steam into the steam/condensate tubes;  
and

an air moving system comprising a fan arranged to drive an airflow across the fins attached to the steam/condensate tubes.

30. The air cooled condenser of claim 29 further comprising:  
risers connected to feed the steam into the distribution headers;  
wherein the steam/condensate tubes, the distribution headers, the risers, and the air moving system are arranged to form the air cooled condenser as an A-frame type air cooled condenser.

31. The air cooled condenser of claim 17 wherein at least 70% of the intermittent flow interruptions are positioned within a first one-half of a length of the fins closest to an air flow entrance of fins.

32. A method of cooling using an air cooled condenser fin, the method comprising:  
flowing air through an air flow channel defined by flow channel walls; and  
interrupting the flowing of air at intermittent flow interruptions defined by the flow channel walls which are spaced apart along the air flow channel.

33. The method of claim 32 wherein the intermittent flow interruptions are placed at locations where a boundary layer of the flowing air has normalized.

34. The method of claim 32 wherein the intermittent flow interruptions are spaced apart along the air flow channel by at least 5 hydraulic diameters.

Differential Temperature versus Length

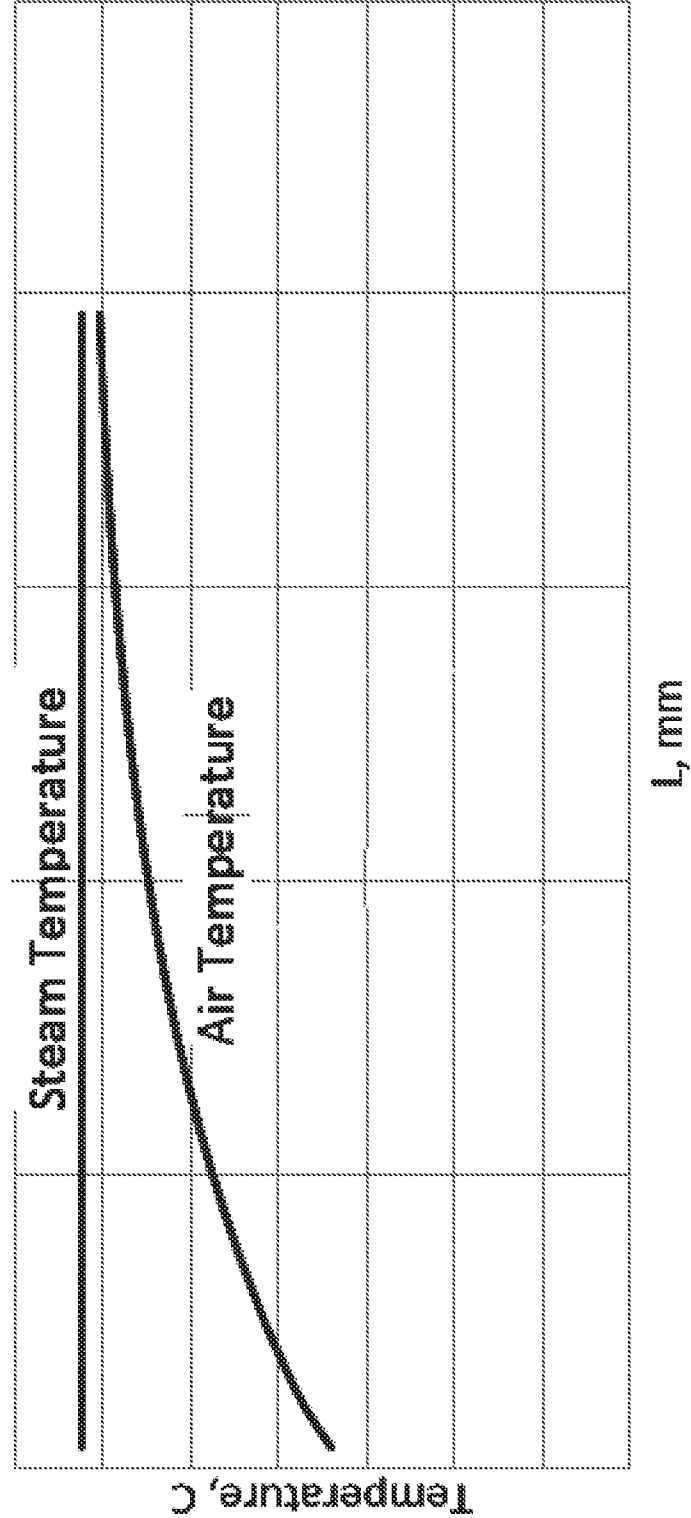


Fig. 1

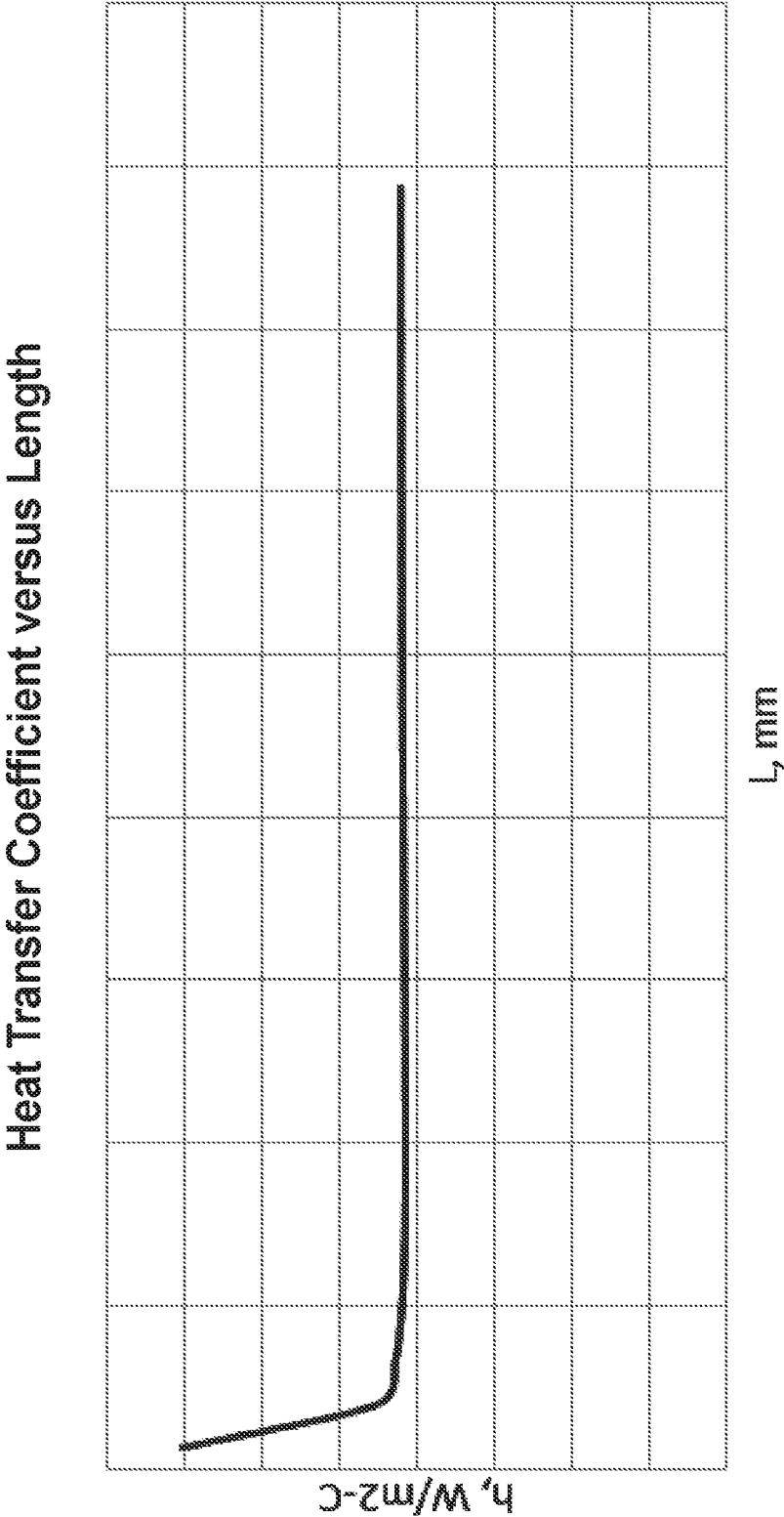


Fig. 2

Air Pressure Drop versus Length

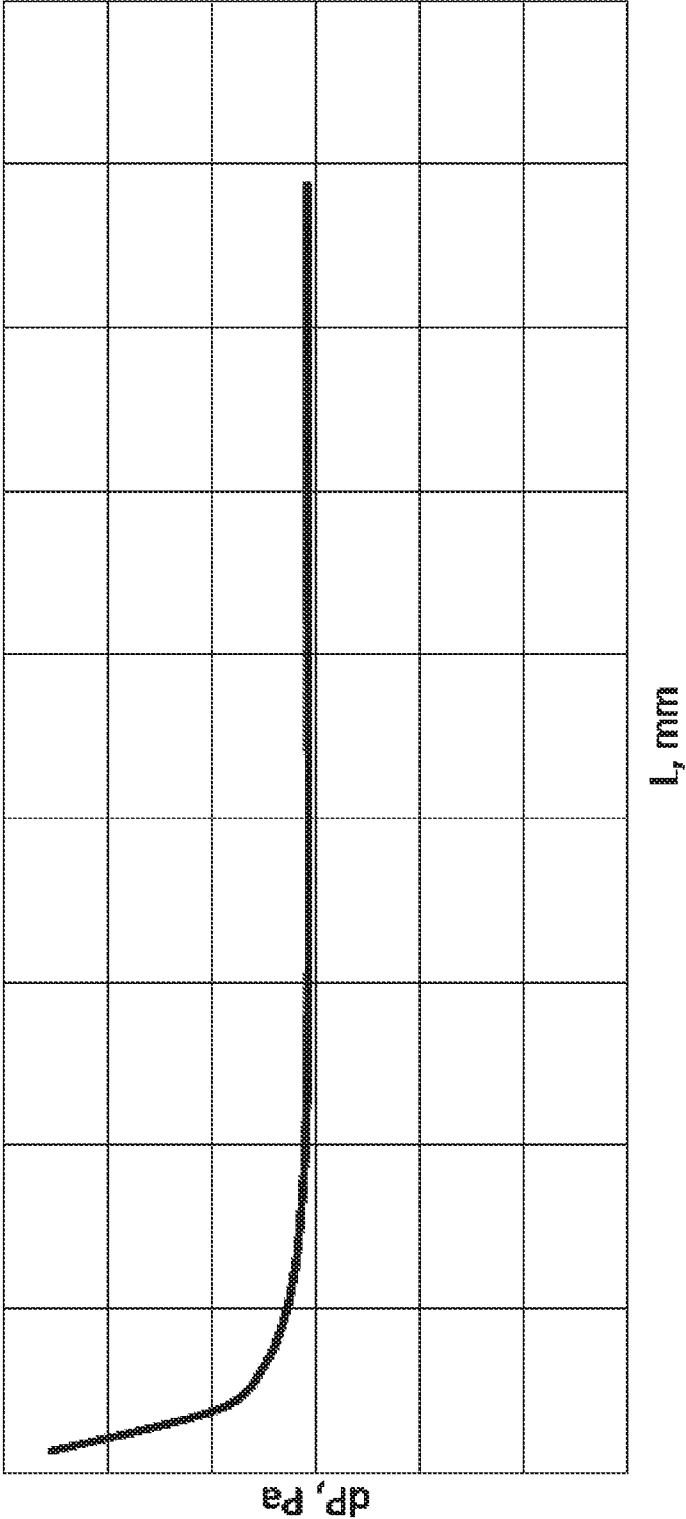


Fig. 3

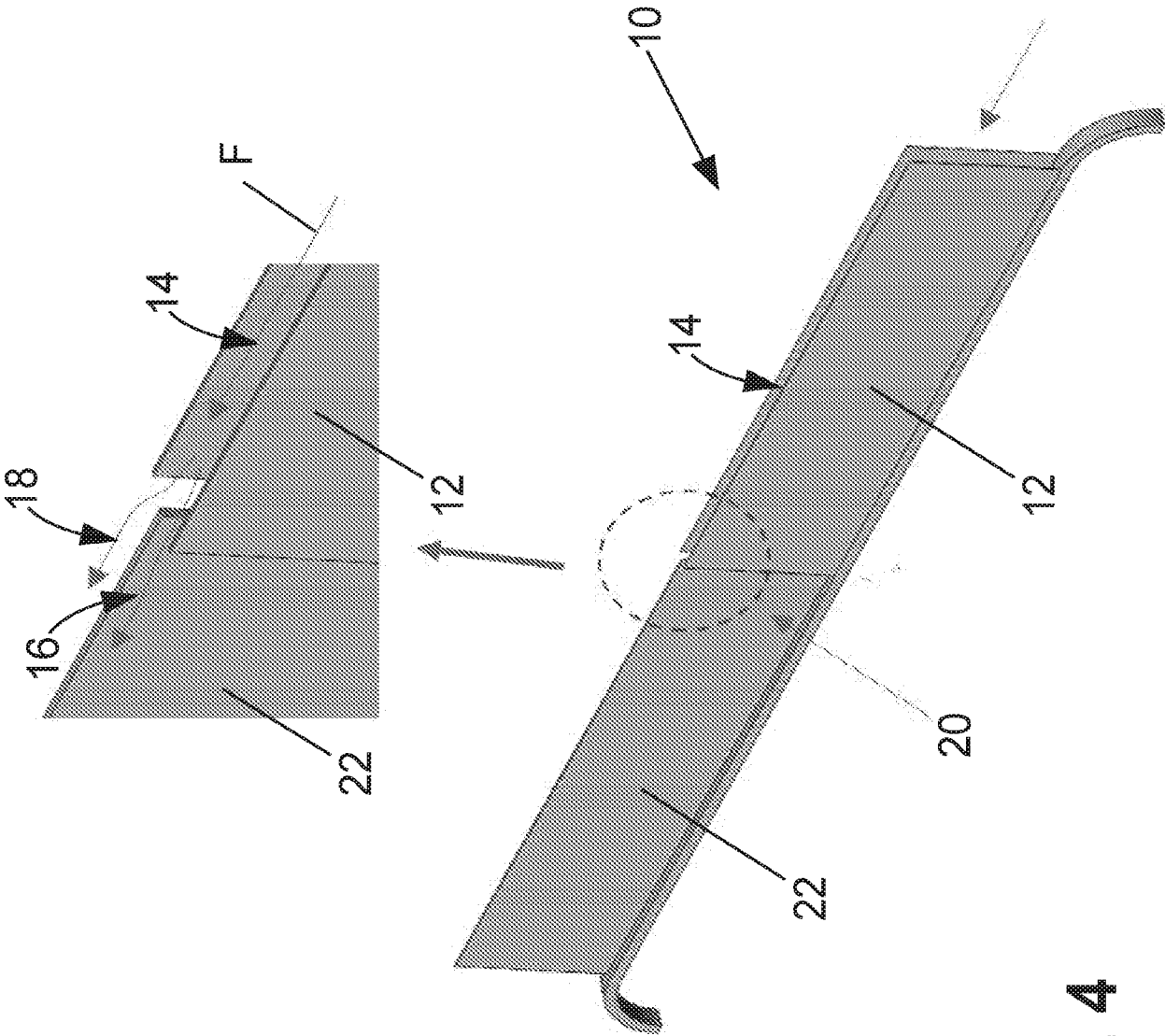


Fig. 4

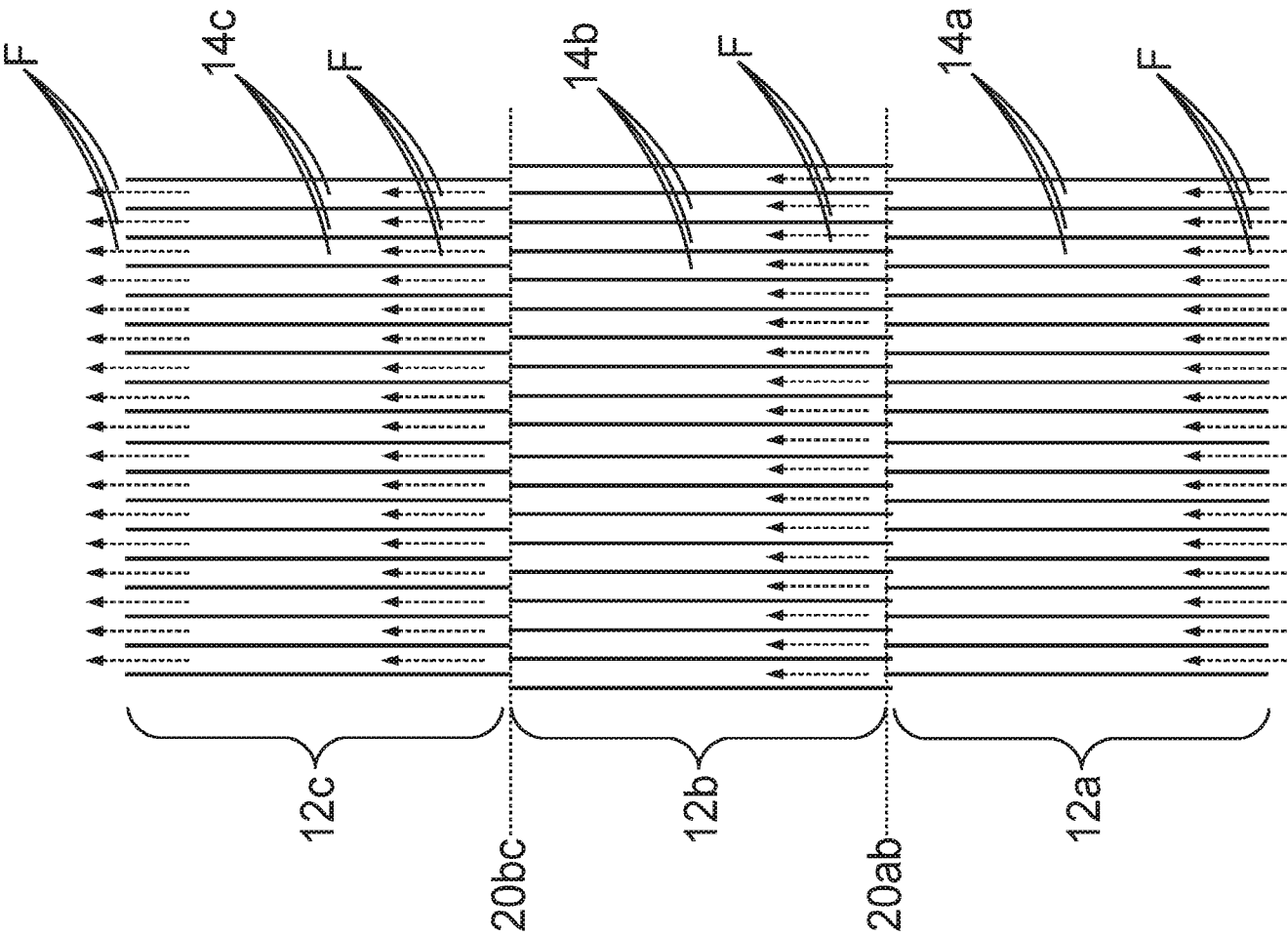


Fig. 5

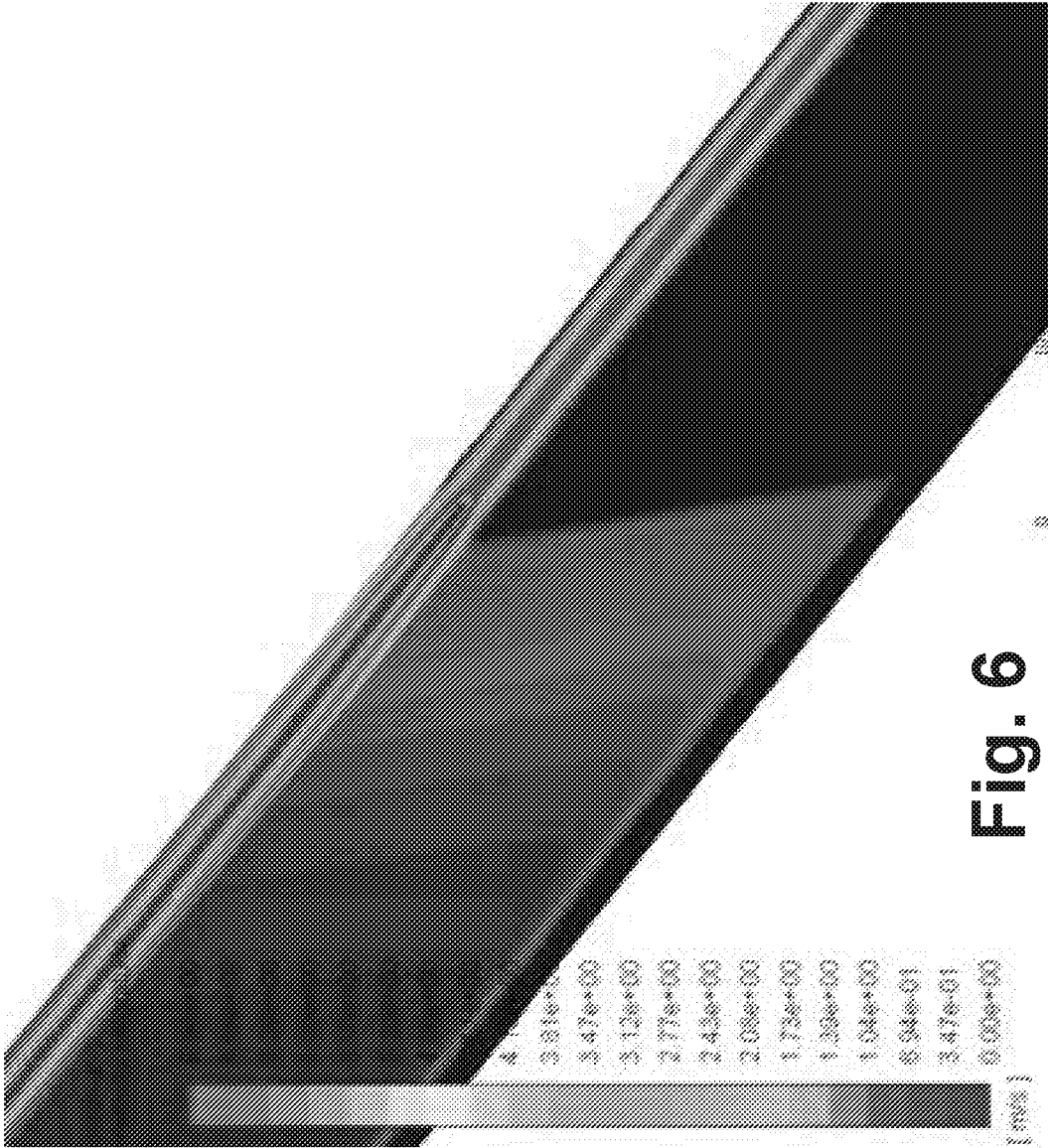


Fig. 6

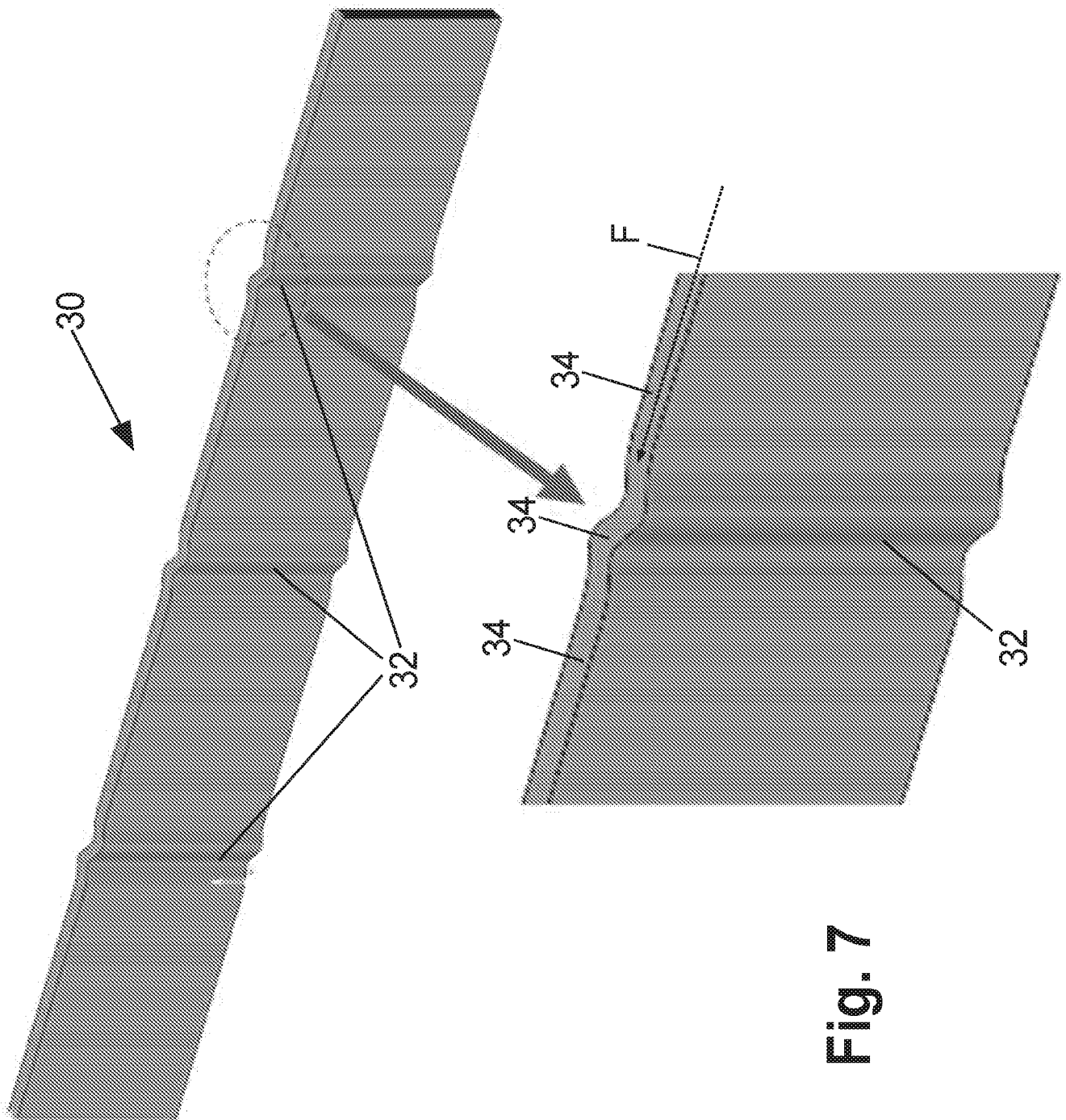
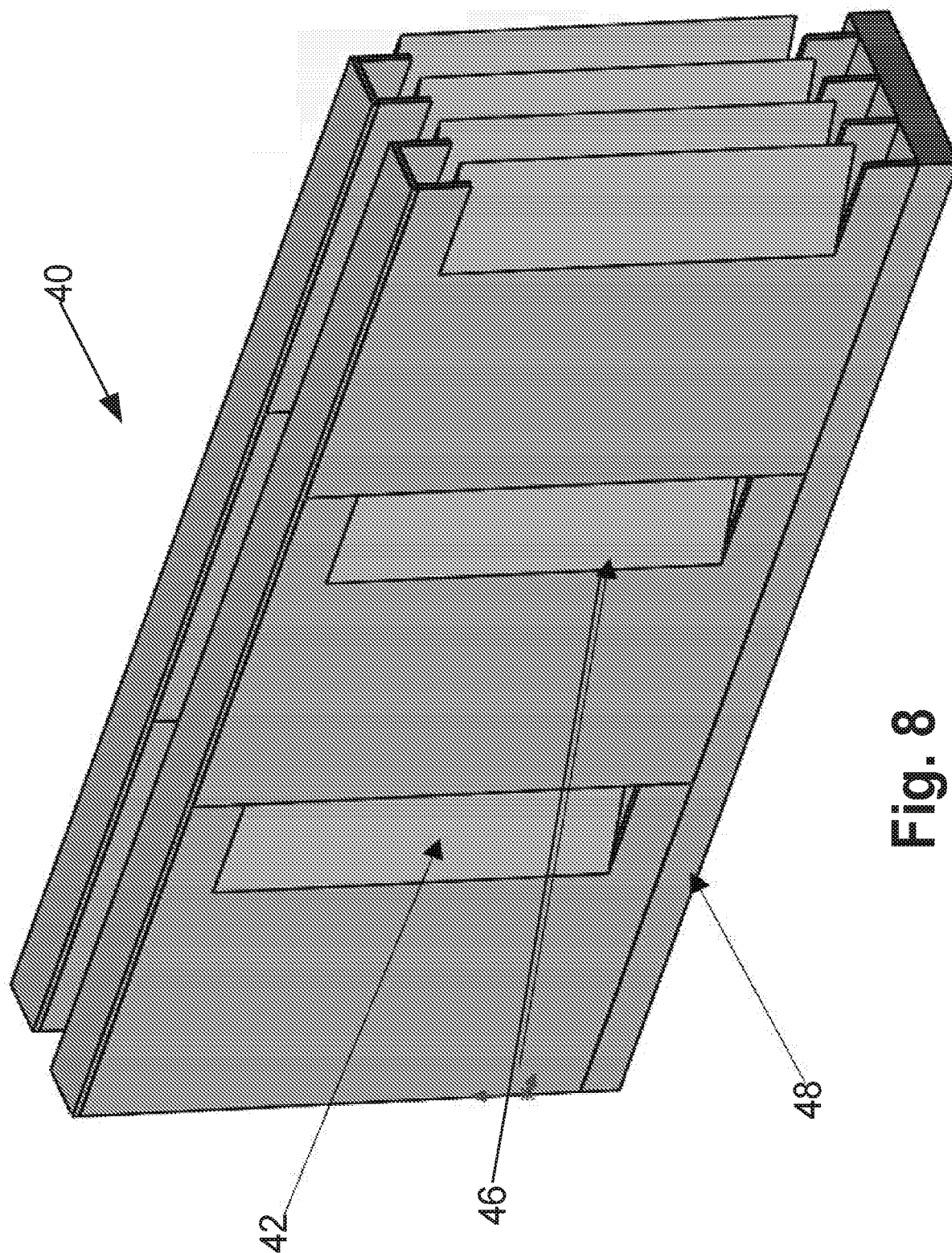


Fig. 7



**Fig. 8**

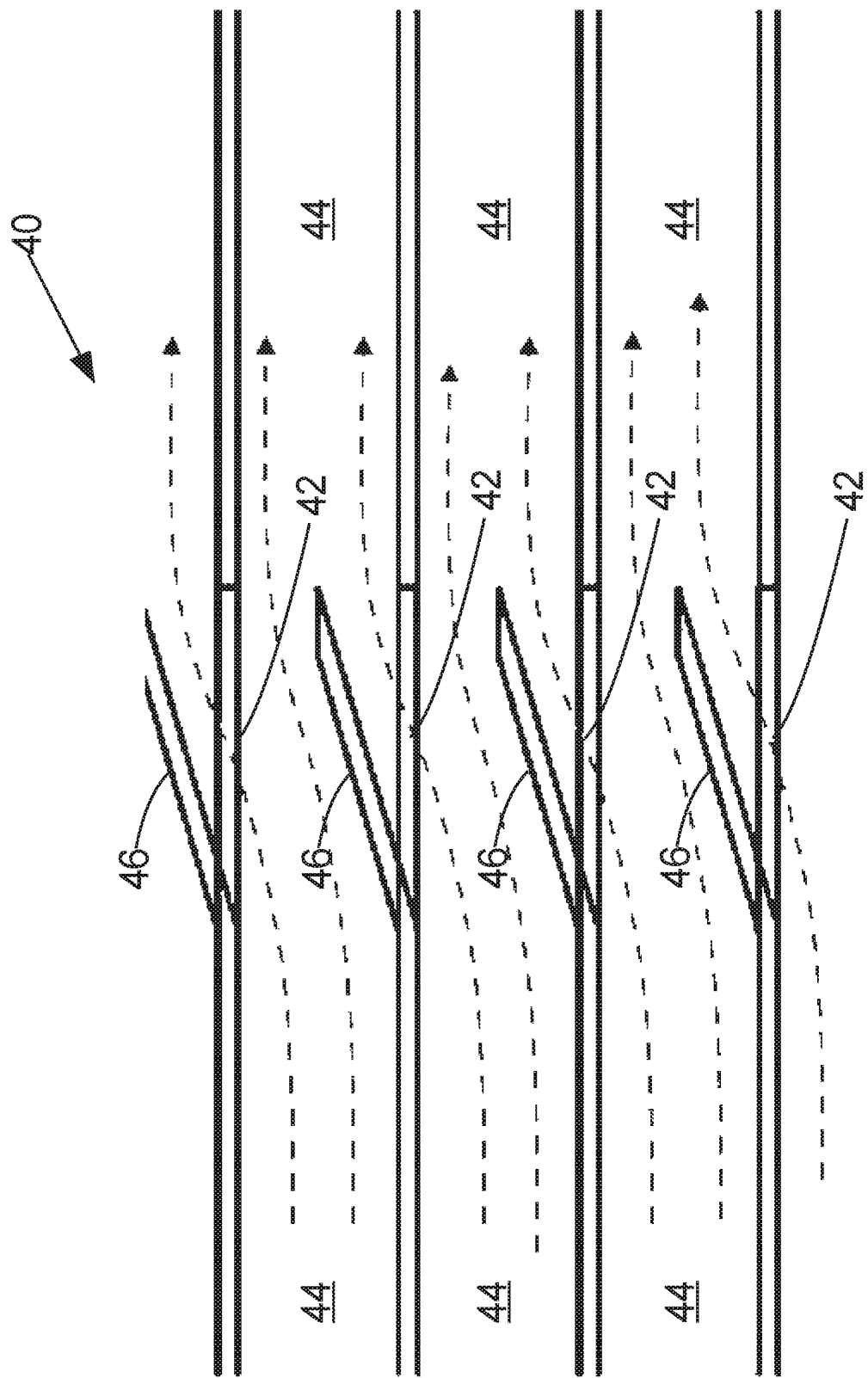


Fig. 9

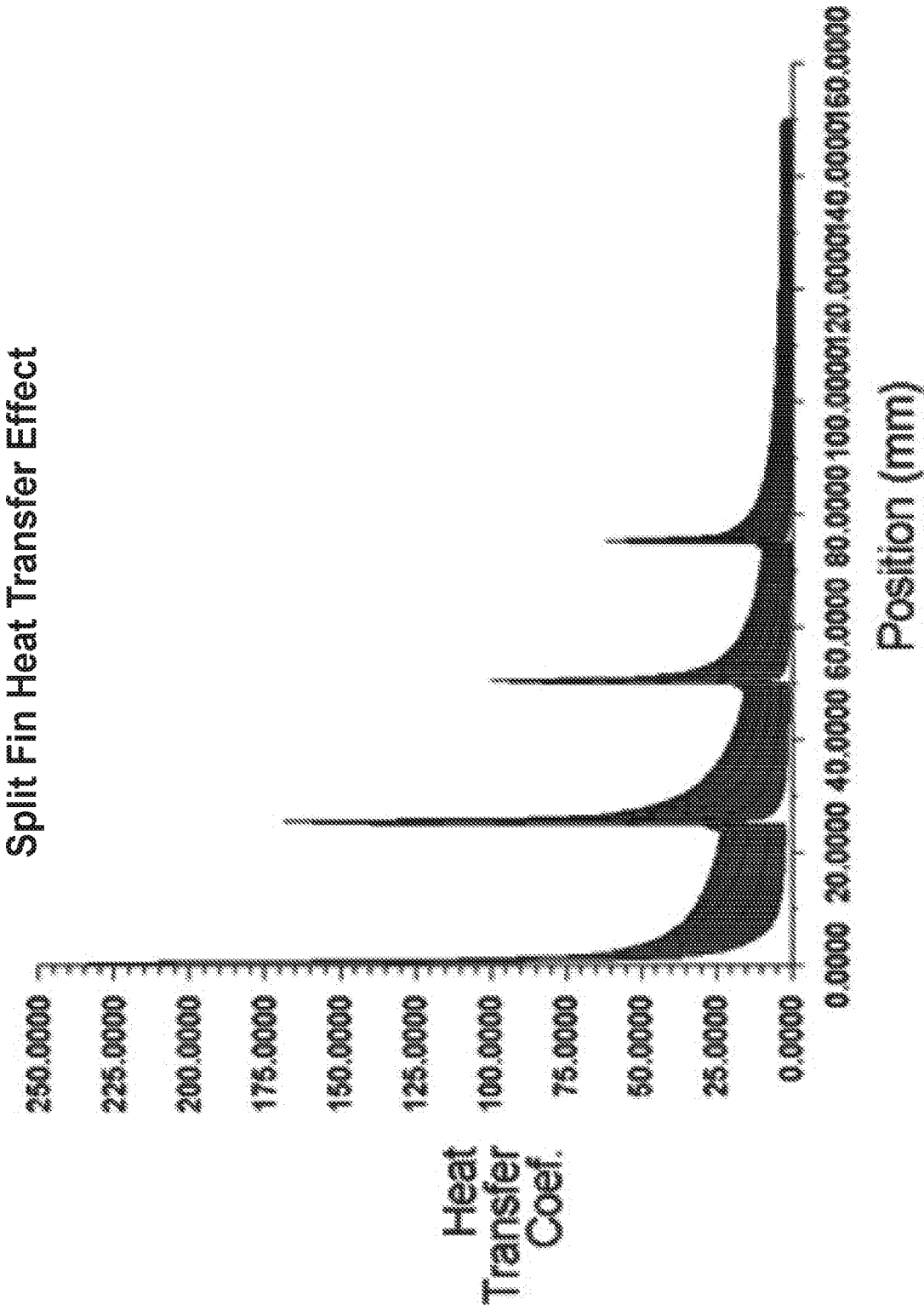


Fig. 10

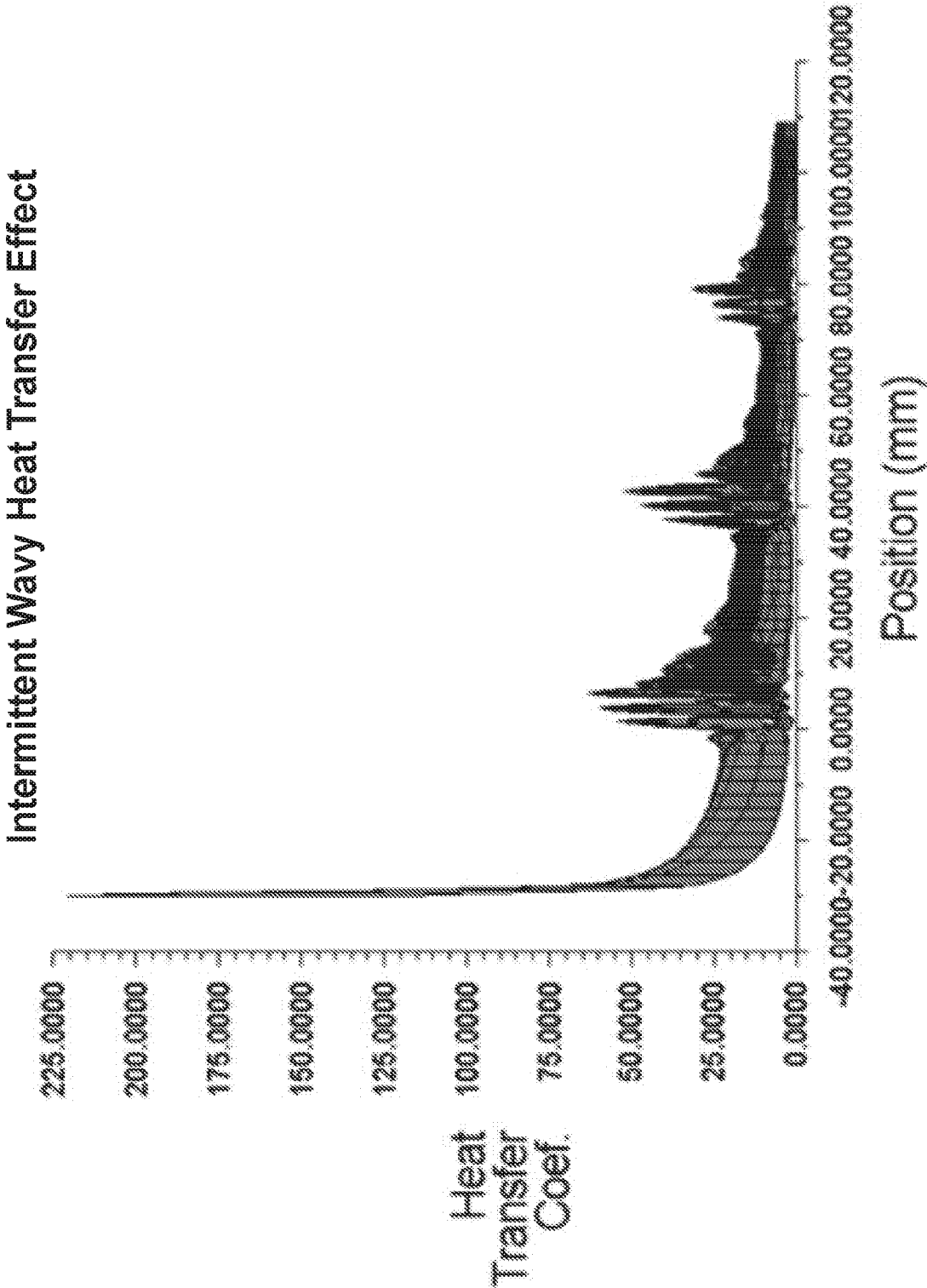


Fig. 11

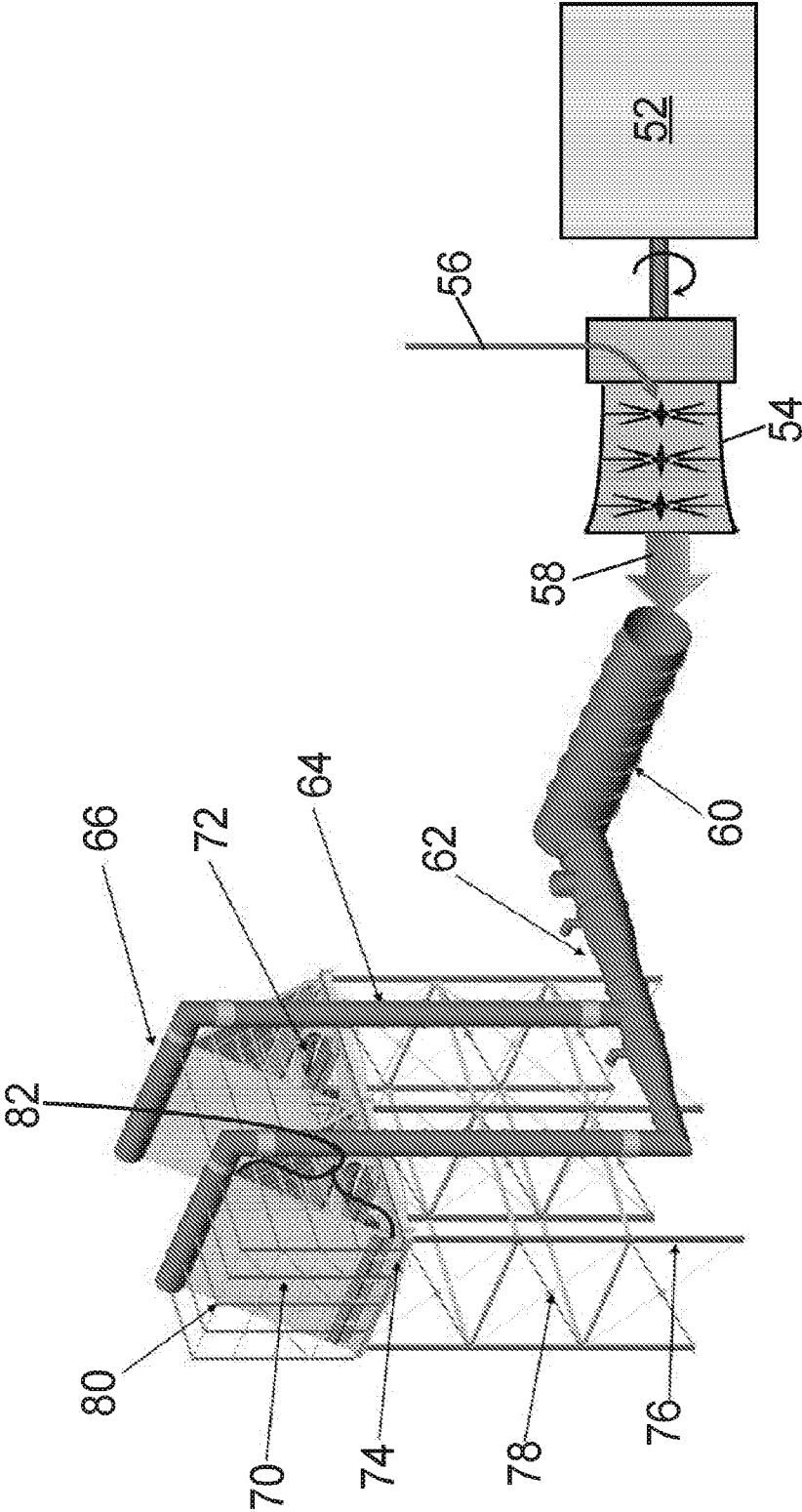


Fig. 12

## INTERNATIONAL SEARCH REPORT

International application No.

PCT/US2020/028659

## A. CLASSIFICATION OF SUBJECT MATTER

IPC(8) - F28F 3/04; F24F 13/08; F28B 1/06; F28F 3/02 (2020.01)

CPC - F28F 3/04; F24F 13/08; F28B 1/06; F28D 1/0233; F28F 3/02; F28F 3/027; F28F 3/046; F28F 3/048; F28F 2215/00; F28F 2215/08 (2020.05)

According to International Patent Classification (IPC) or to both national classification and IPC

## B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)

see Search History document

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

see Search History document

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)

see Search History document

## C. DOCUMENTS CONSIDERED TO BE RELEVANT

| Category* | Citation of document, with indication, where appropriate, of the relevant passages | Relevant to claim No.                     |
|-----------|------------------------------------------------------------------------------------|-------------------------------------------|
| X         | US 3,810,509 A (KUN) 14 May 1974 (14.05.1974) entire document                      | 1, 2, 4-18, 20-23, 25, 26, 28, 29, 32, 34 |
| ---       |                                                                                    | ---                                       |
| Y         |                                                                                    | 30                                        |
| Y         | US 2017/0363357 A1 (EVAPCO, INC.) 21 December 2017 (21.12.2017) entire document    | 30                                        |
| A         | US 2006/0120213 A1 (TONKOVICH et al) 08 June 2006 (08.06.2006) entire document     | 1-34                                      |
| A         | US 2004/0134226 A1 (KRAAY et al) 15 July 2004 (15.07.2004) entire document         | 1-34                                      |

☐ Further documents are listed in the continuation of Box C.☐ See patent family annex.

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"O" document referring to an oral disclosure, use, exhibition or other means

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"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention

"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone

"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art

"&amp;" document member of the same patent family

Date of the actual completion of the international search

08 June 2020

Date of mailing of the international search report

02 JUL 2020

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