Oct. 30, 1956 R. H. GUYTON 2,768,508

REFRIGERATOR CONDENSER

Filed March 30, 1953 4 Sheets-Sheet 1

FIG. 1

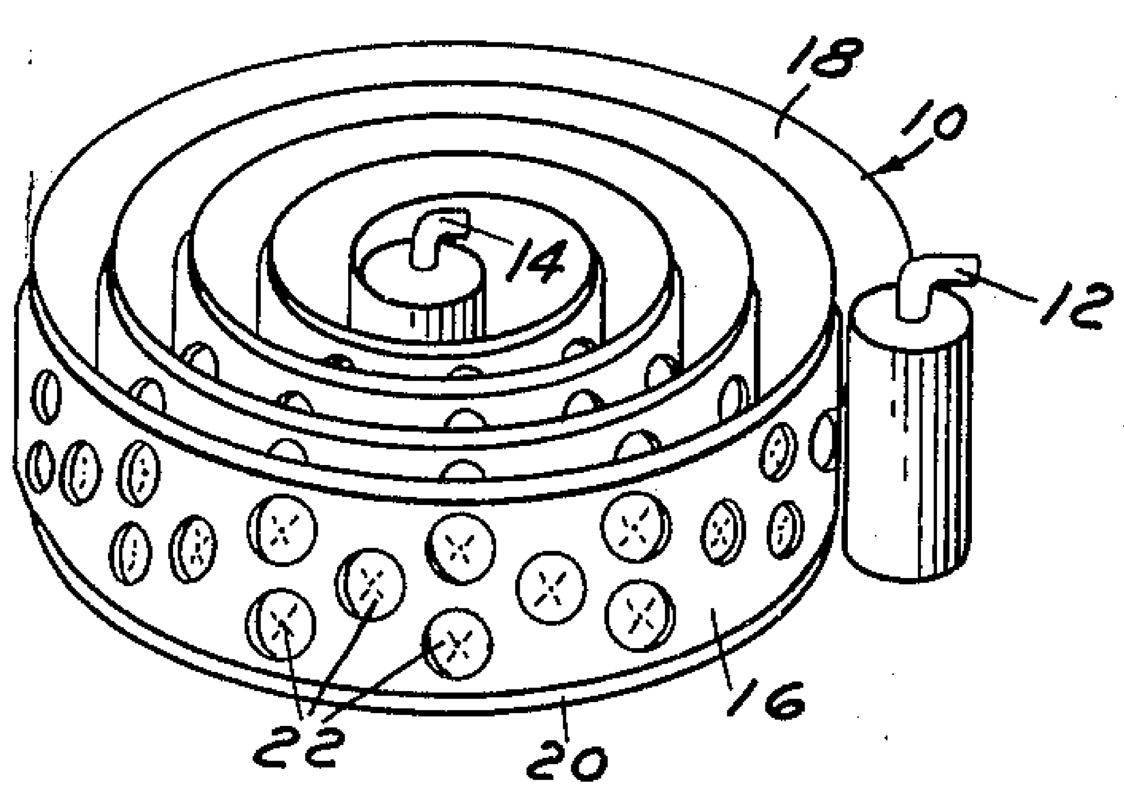


FIG. 2

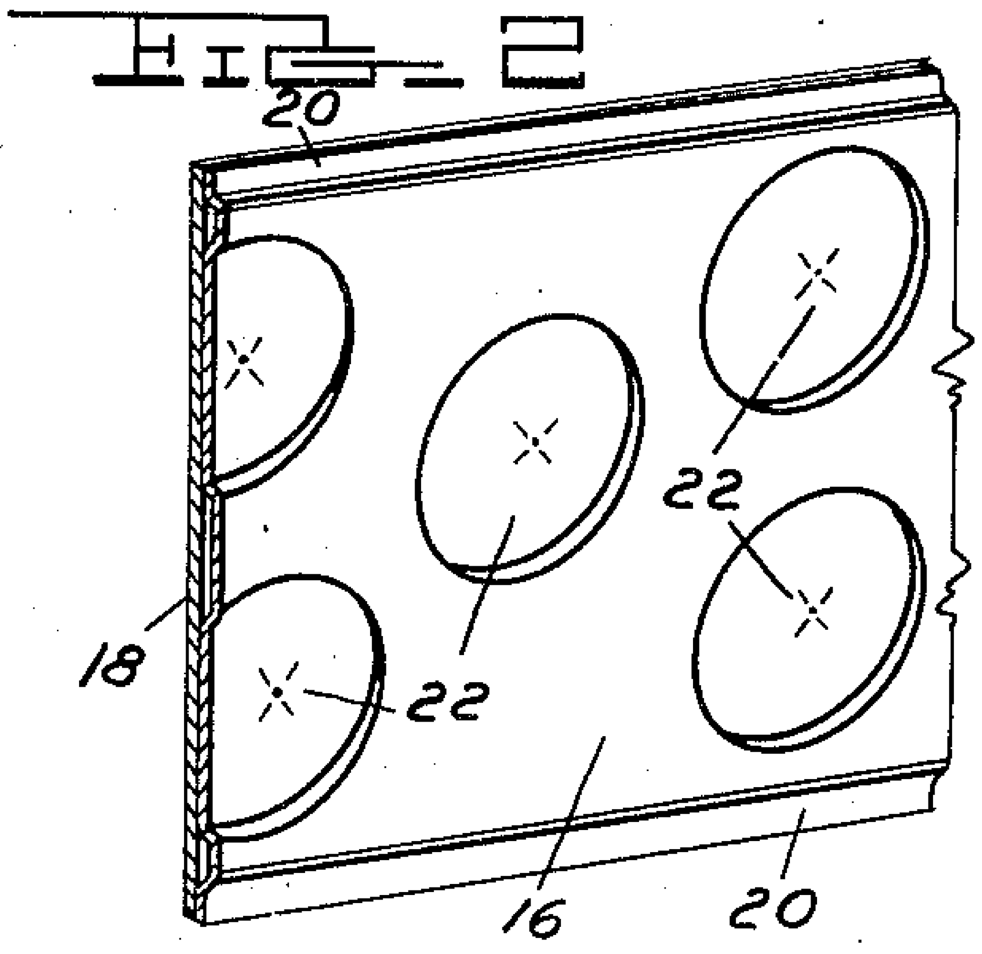


FIG. 4

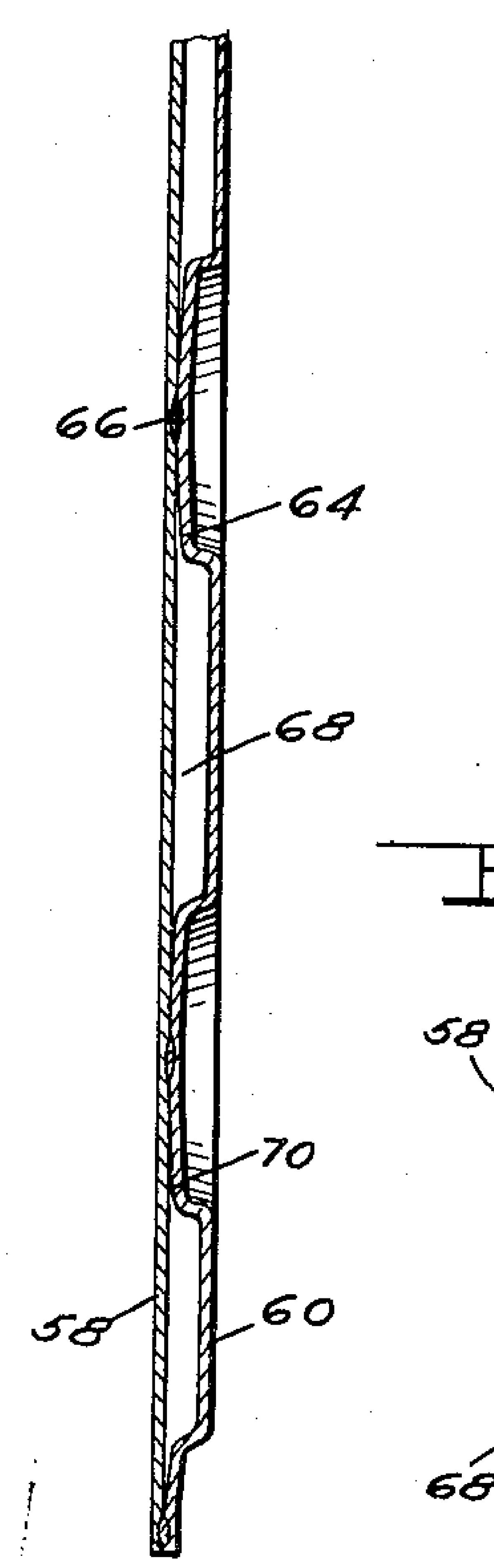


FIG. 5

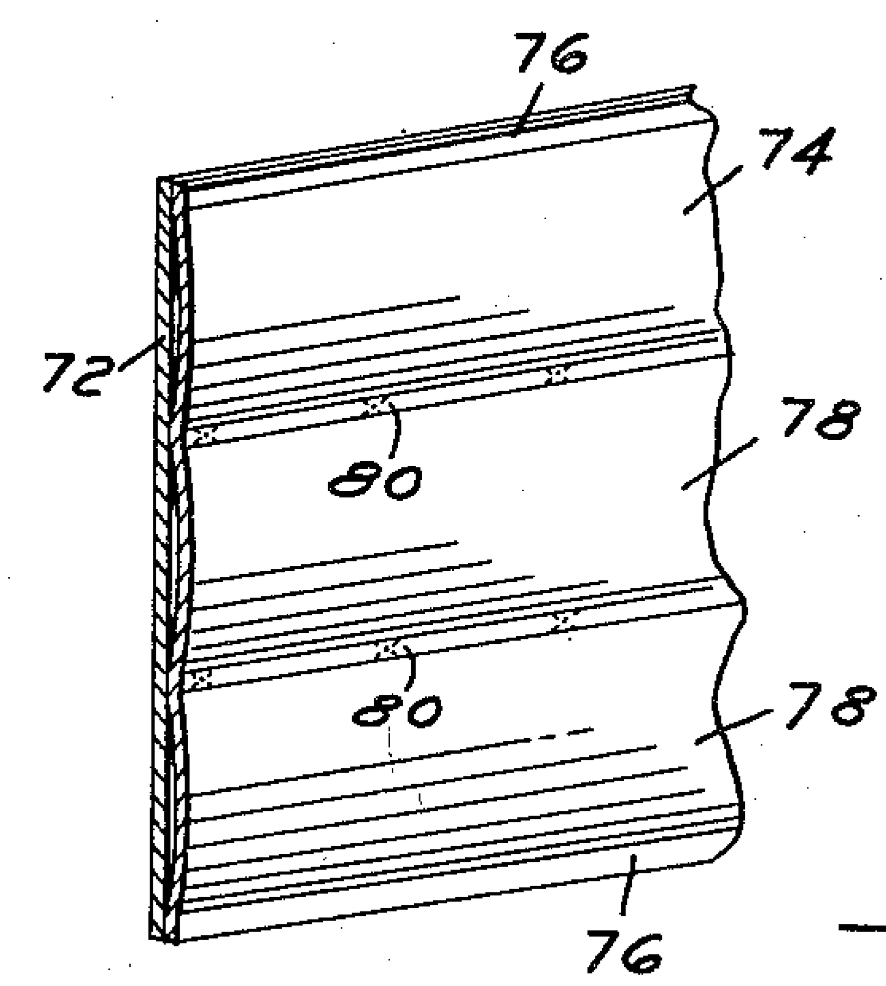


FIG. 3

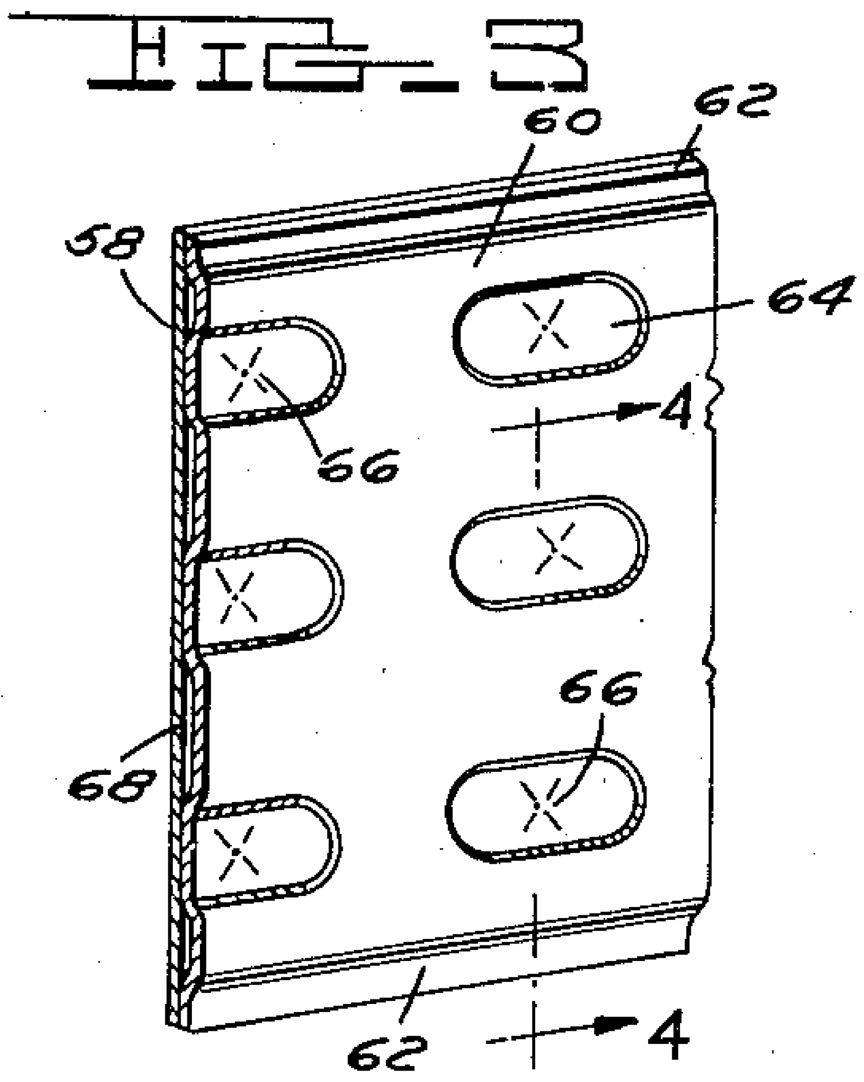
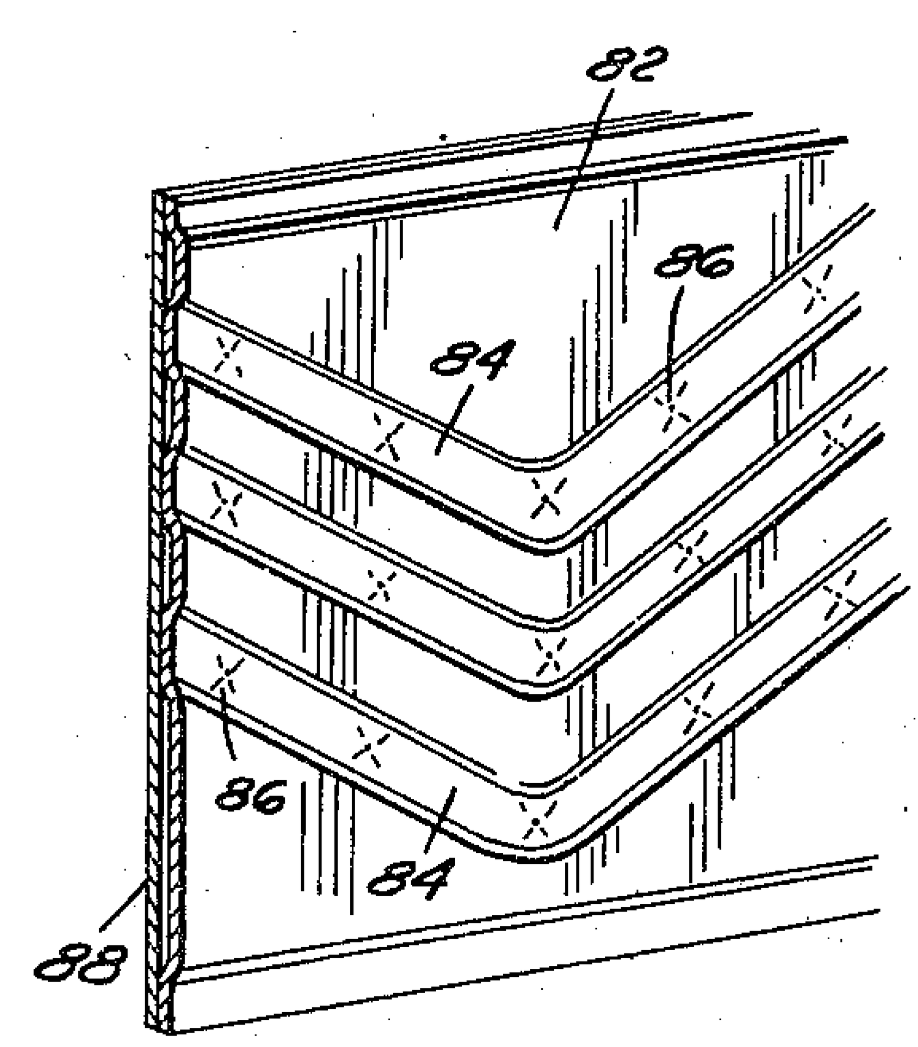


FIG. 6



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REFRIGERATOR CONDENSER

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FIG. 7

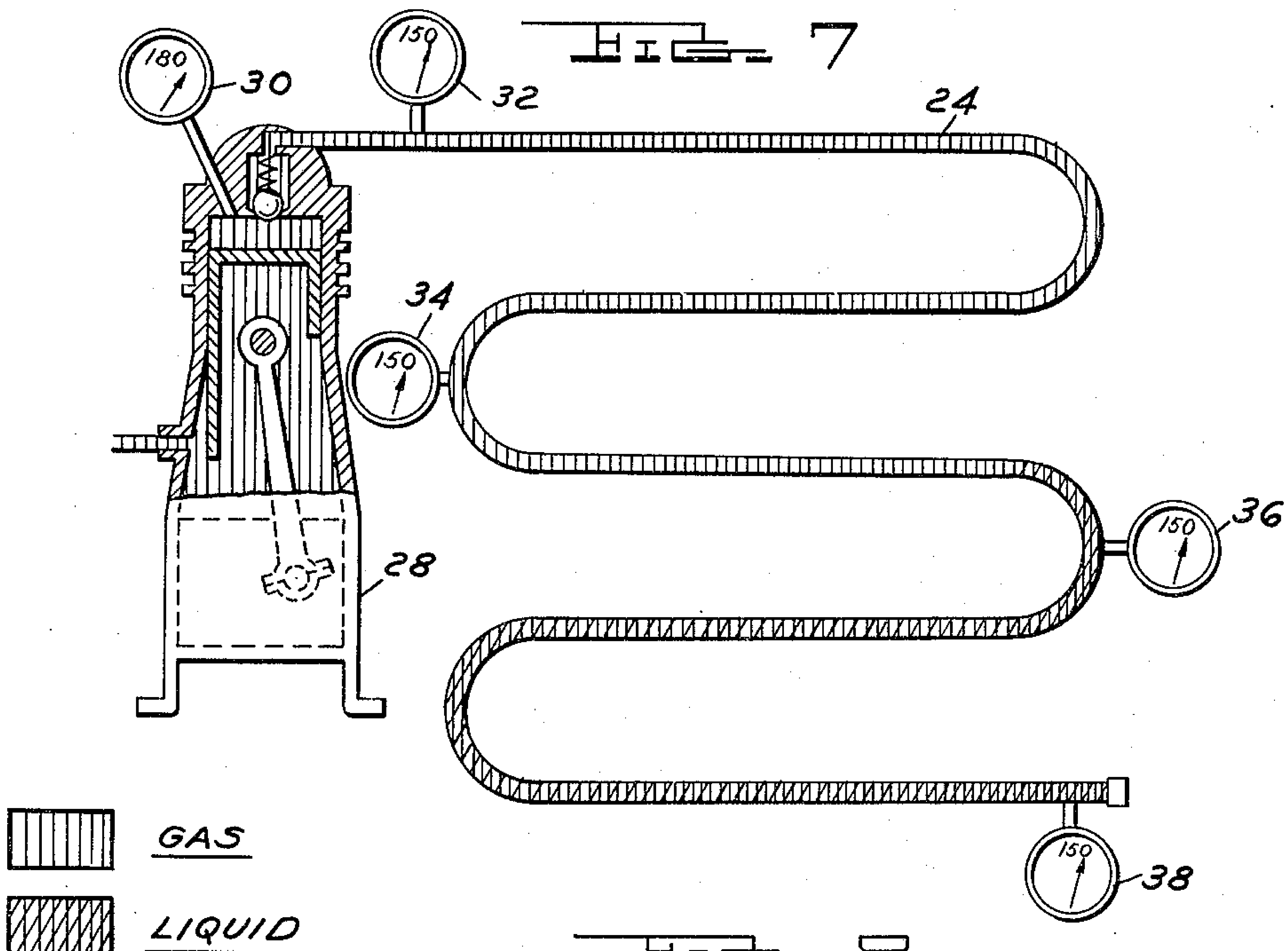
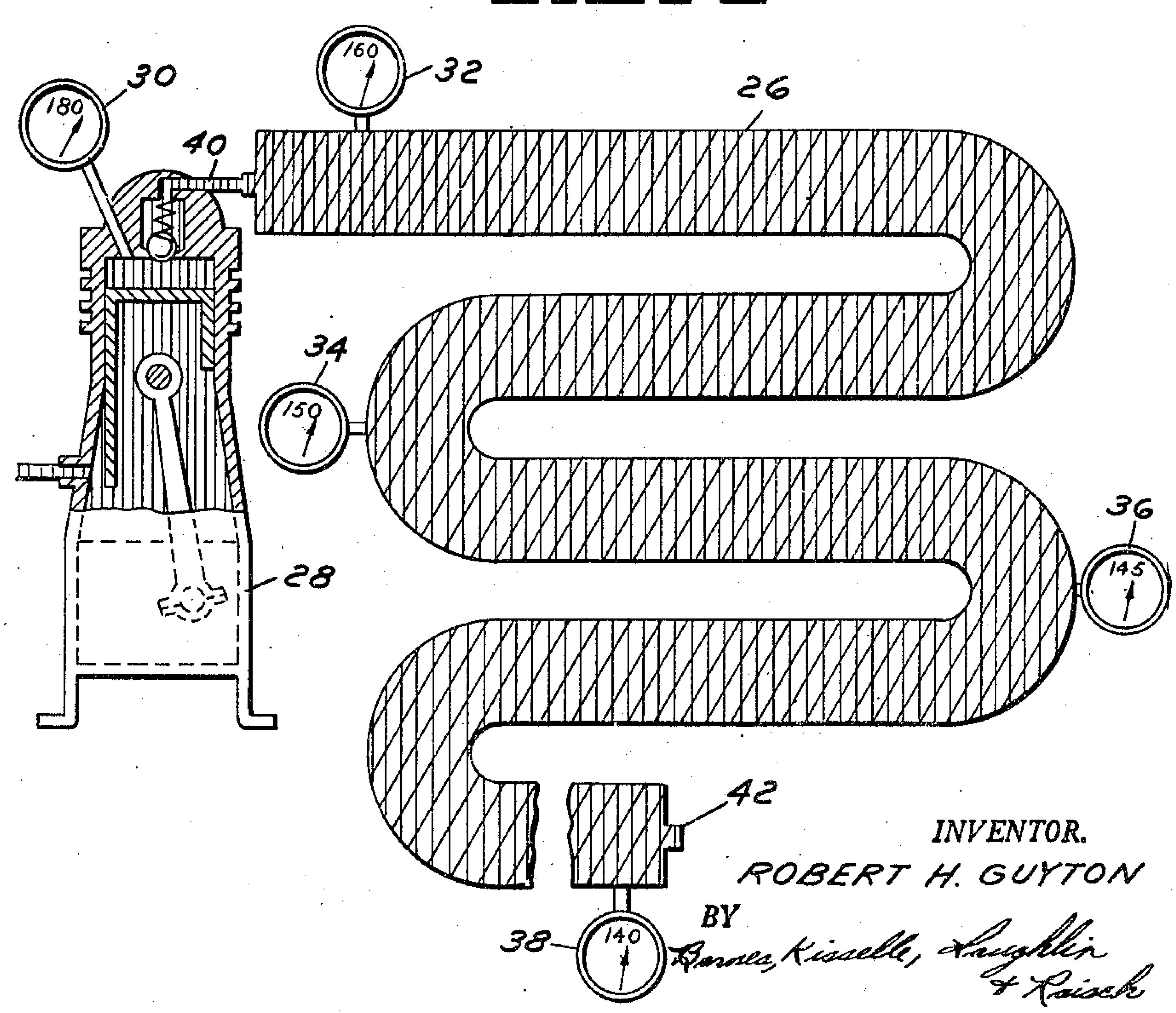


FIG. 8



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REFRIGERATOR CONDENSER

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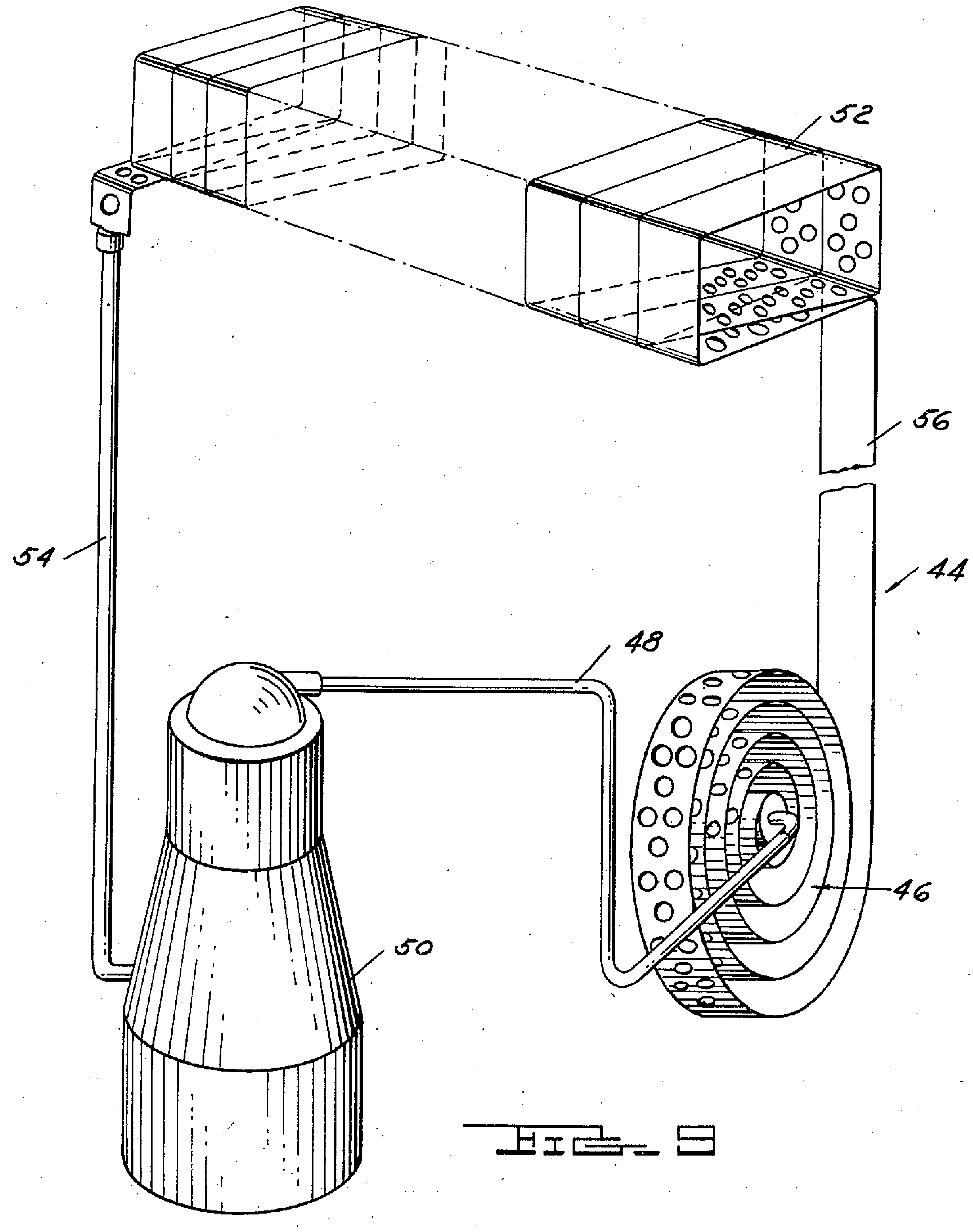

FIG. 9

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FIG. 10

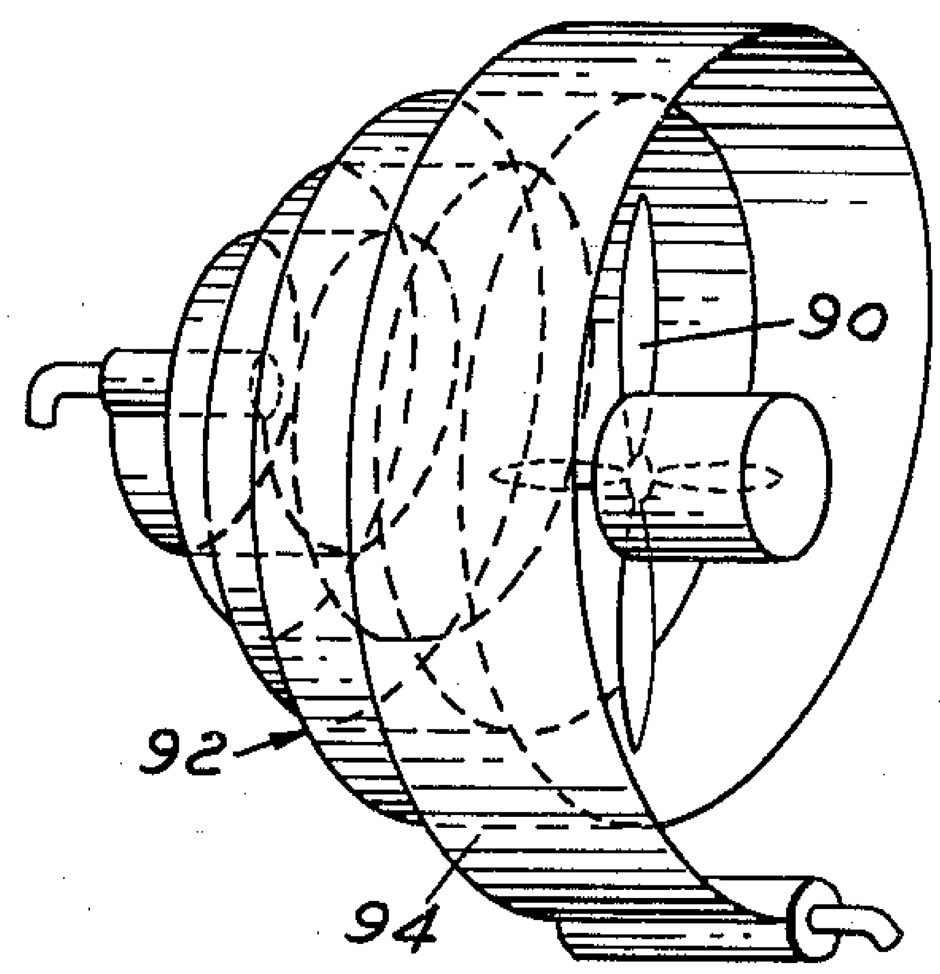


FIG. 11

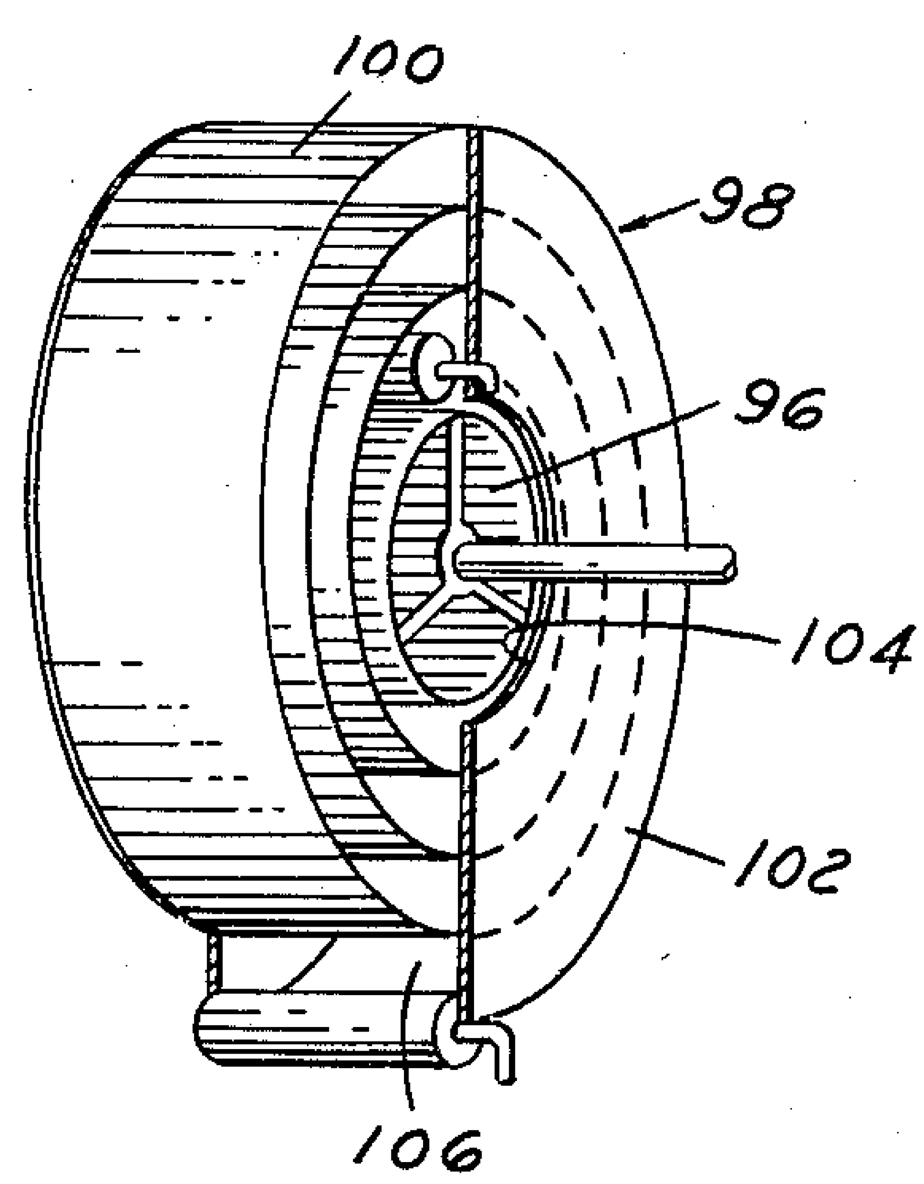
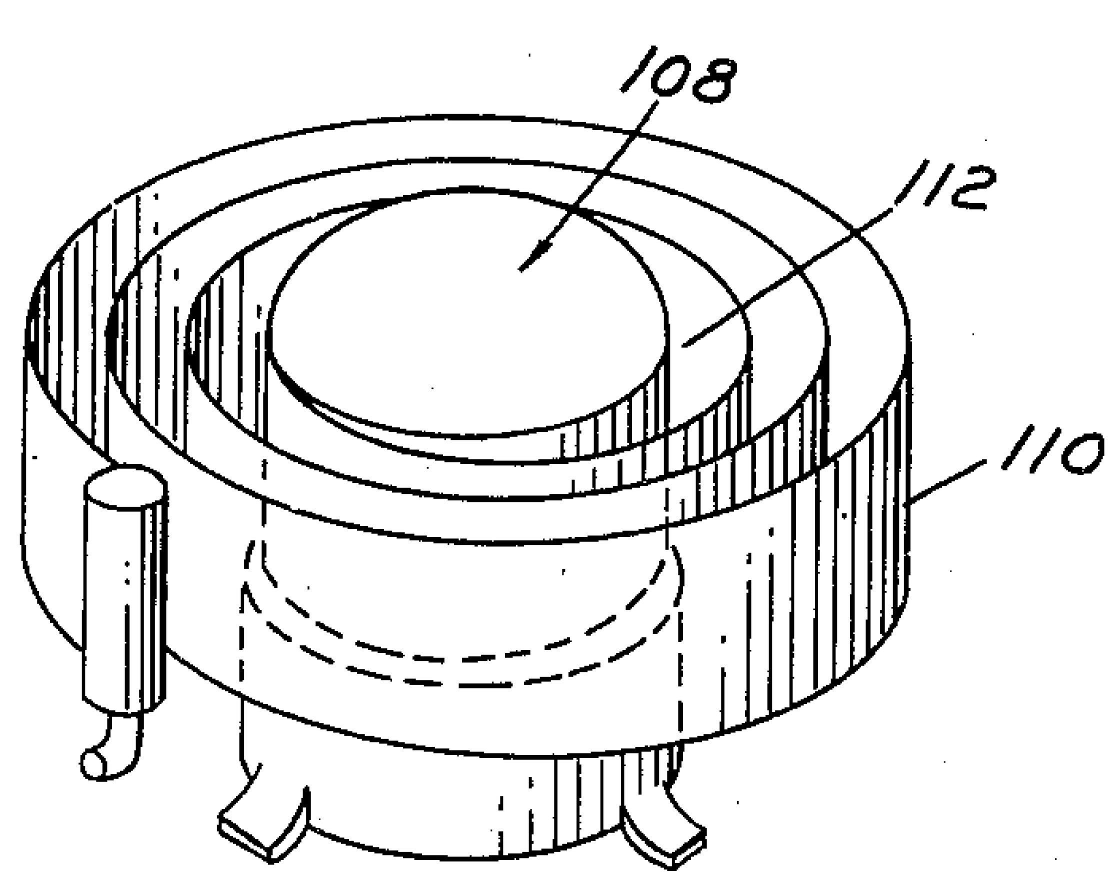


FIG. 12



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United States Patent Office

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REFRIGERATOR CONDENSER

Robert H. Guyton, Lima, Ohio

Application March 30, 1953, Serial No. 345,569

12 Claims. (Cl. 62—115)

This invention relates to a refrigerator condenser and more particularly to a condenser having a higher rate of condensation capacity per unit of surface area and in which is incorporated a novel principle of condensing action.

Refrigerating condensers as presently constructed for use in the conventional compressor-condenser-expander circuit are usually in the form of a round tube of sufficient cross sectional area to accommodate the volume of gas that has to flow through it without any appreciable pressure drop. The tube is usually provided with a series of return bent portions so that the condenser lies generally in a flat plane. The surface area of the tube itself is insufficient to extricate all the heat necessary to condense the gas, and it is the usual custom therefore to provide a plurality of closely spaced fins which serve as secondary cooling surfaces. A condenser of this type not only occupies a relatively large volume of space but even more serious a problem is the fact that after the condenser has been in use for a relatively short period of time the space between the fins becomes clogged with lint and dust and the condensing action is seriously impaired. In order to obtain the necessary cooling with condensers of this type, it has been found necessary to provide as many as twenty or more fins per each lineal inch of condenser tube. The natural tendency for such condensers to become clogged is obvious.

Aside from the mechanical design of conventional condensers, there is associated with such condensers another objection, namely, the fact that condensation of the gaseous refrigerant within the condenser tube occurs through a thermo-siphon action. A thin layer of gas adjacent the inside surface of the tube is first cooled and condensed and before further condensation and cooling take place, this cooled or condensed layer of gas must be displaced by another layer of gas which moves radially from the central portion of the tube. I have determined that this method of condensing gaseous refrigerant into liquid refrigerant is inefficient. In an efficient condensation action, tiny seeds or droplets of refrigerant are formed throughout the condenser, and these seeds in turn cause further condensation either by attracting gaseous refrigerant or by absorbing the heat from the surrounding gas to thereby cool the same.

Another feature of refrigerating condensers as presently constructed which should be mentioned in connection with the present invention is that such condensers are always designed to produce no appreciable pressure drop across the condenser. It has been believed that a drop in pressure across the condenser would produce a corresponding lowering in the efficiency of the condenser. It has also been believed that, if the head pressure on the condensing unit is raised above a predetermined value, the efficiency of the condensing unit would likewise be decreased.

It is an object of this invention to provide a condenser construction which is more efficient than condensers of conventional construction described above.

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A further object of this invention resides in the provision of a condenser which is of economical construction and which can be made very compact so as to occupy a minimum of space.

A further object of this invention resides in the provision of a condenser which is not susceptible of becoming clogged by dust, lint, and the like.

I have discovered that the condensing action within a condenser can be materially increased by constructing the condenser so that the passageway for refrigerant provided by the condenser is relatively long and thin. The walls of the condenser are spaced apart so that the refrigerant passing therethrough is in the form of a very thin layer, these walls being relatively wide as compared with the spacing between the walls so that a maximum amount of cooling surface is provided for the refrigerant flowing through these walls.

Furthermore, I have found that the condensing action can be materially increased by increasing the velocity of flow through the condenser, and this increase in velocity of flow can be produced by constructing the condenser so that there will be a substantial drop in pressure across the condenser. The condenser is constructed so that this drop in pressure will be uniform along the extent of the condenser.

In the drawings:

Fig. 1 is a perspective view of one form of condenser embodying the present invention.

Fig. 2 is an enlarged fragmentary view of a portion of the condenser shown in Fig. 1.

Fig. 3 is a view similar to Fig. 2 and showing another form of construction of the condenser side walls.

Fig. 4 is a fragmentary enlarged sectional view along the line 4—4 in Fig. 3.

Figs. 5 and 6 are views similar to Fig. 2 and showing further modifications in the side wall construction of the condenser of this invention.

Figs. 7 and 8 are diagrammatic views illustrating the difference in the condensing action that occurs in a condenser of conventional construction as compared with a condenser constructed in accordance with this invention.

Fig. 9 is a somewhat diagrammatic view showing a construction wherein a condenser and an evaporator are formed from a single conduit constructed in accordance with this invention.

Fig. 10 is a perspective view of a condenser and fan combination incorporating another form of the present invention.

Fig. 11 is a perspective view, partly cut away, showing a condenser and blower embodying another form of the present invention.

Fig. 12 is a perspective view of a condenser and compressor combination illustrated in another form of the present invention.

The condenser illustrated in Figs. 1 and 2 comprises a spiral **10** formed of flattened tubular shape having an inlet fitting at **12** and an outlet fitting at **14**. Intermediate the fittings the spiral is formed with two flat walls **16** and **18** which are joined along their edges as at **20** by seam welding or the like. The wall **16** may be provided with a plurality of indentations **22** at spaced intervals along its width and length which are in contact with the wall **18**. These indentations **22** may be spot welded to the wall **18** if desired, but I have found that with the spirally shaped condenser such as illustrated at **10** spot welding at the indentations **22** is unnecessary. The curved contour of the two walls **16** and **18** maintain the indentations **22** in contact with the wall **18**. The condenser may be fabricated from sheet metal by taking two strips, forming one of them to the contour illustrated at **16**, and seam welding or otherwise welding the plates together at the edges **20**. The assembly may then be bent into the form of a spiral and the inlet and outlet fittings **12** and **14** brazed or otherwise secured to the open ends of the spiral.

In order to obtain the maximum benefits from the present invention the condenser should be designed with respect to the system in which it is to operate so that a very high velocity of gaseous flow through the condenser is achieved. The spacing between the walls **16** and **18** is only slightly larger than the space which will permit capillary action of the liquid to occur. When a condenser of this construction is connected to the usual refrigerating circuit, I have found that the condensing action is materially increased per unit of condenser area and furthermore that this results in more efficient operation of the refrigerating system. This may be attributed to several features in the design of my condenser construction.

In the first place, by spacing the walls **16** and **18** very close together, the refrigerant is caused to flow through the condenser in a very thin layer and is in contact with a maximum area of cooling surface. Thus, the entire layer of gaseous refrigerant flowing through the condenser can be cooled substantially instantaneously as compared with a round tube condenser of conventional construction where the cooling throughout the cross section of the tube necessarily depends upon a thermo-siphoning action. Furthermore, the spacing of the walls **16** and **18** creates considerable friction and tends to produce a drop in pressure along the condenser. This drop in pressure increases the velocity of flow through the condenser and at the same time increases the rate of cooling of the hot gaseous refrigerant flowing through the condenser.

Another feature of my design which is believed to be important is the spiral shape of the condenser. This shape prevents the refrigerant from flowing in a straight line. The refrigerant continually impinges against the walls **16** and **18** and small droplets of liquid are caused to form which act as "seeds" which absorb heat from the surrounding gaseous refrigerant and encourage the further condensation of additional particles of gas into liquid.

Still another feature of the construction shown in Figs. 1 and 2 which is believed to be important from the standpoint of increasing the efficiency of the condenser resides in the provision of the indentations **22**. In the first place, these indentations form obstructions to the flow of gaseous refrigerant through the condenser and thereby produce a further impingement of the gaseous refrigerant against cooling surfaces. These indentations are also important from another standpoint. Although the walls **16** and **18** contact one another at the indentations **22**, it is impossible from a practical standpoint to prevent gas from entering the minute spaces between walls **16** and **18** at the indentations **22**. The gas which flows into these tiny spaces is immediately condensed and the pressure is locally reduced, thus causing additional gas to flow into these spaces and driving the liquid out into the larger passageways between the walls **16** and **18**. As the liquid is driven out from between the indentations **22** and the wall **18**, this liquid absorbs heat from the surrounding gaseous refrigerant and produces additional condensation. With this arrangement the condenser is maintained in a wet condition substantially throughout its length as compared with a conventional condenser which is wet only through the lower portion thereof.

The difference in the condensing action between a condenser of this invention and a condenser of conventional construction is illustrated diagrammatically in Figs. 7 and 8 wherein a conventional condenser in the form of a round tube provided with return bent portions is shown at **24** in Fig. 7 and a condenser constructed in accordance with this invention as shown at **26** in Fig. 8. In the showing of Fig. 7 a compressor is indicated at **28** and the condenser **24** is shown connected to the head of the compressor. Using Freon **12** as refrigerant under one set of normal operating conditions, the pressure in the head of the compressor, which for the purpose of illustration is indicated by a gage **30**, might momentarily reach 180 pounds and the pressures at successive points along the condenser as indicated by gages **32**, **34**, **36** and **38** will normally be quite uniform, perhaps at 150 pounds gage. Adjacent the inlet end of condenser **24** the refrigerant would be substantially completely gaseous as indicated by the legend and adjacent the outlet end of the condenser the refrigerant would be substantially completely liquid. At a point perhaps one-third the distance from the outlet end of the condenser, at the location of gage **36**, the state of the refrigerant changes to predominantly liquid.

With my condenser operating under the same conditions shown in Fig. 7, the pressure at the head compressor **28** might likewise be in the neighborhood of 180 pounds gage as indicated by gage **30**; but due to the construction of my condenser, the head pressure on the condenser as indicated at gage **32** would probably be in the neighborhood of 160 pounds gage and the pressure would gradually drop along the compressor so that at gage **34** the pressure would be in the neighborhood of 150 pounds; at gage **36**, 145 pounds; and 140 pounds at the outlet end of the condenser as indicated by gage **38**. The condenser **26** would contain a substantial amount of liquid refrigerant from substantially its inlet **40** throughout the length thereof to the outlet **42**. Thus, in operation the head pressure in my condenser is substantially higher than in a conventional condenser, and there is a substantial drop in pressure across my condenser as compared with no appreciable drop in pressure across a conventional condenser.

As an illustration of the results obtainable by this construction, tests have been conducted in comparison with a conventional finned tube condenser. Both condensers were connected with identical ¼ H. P. motor compressor units. At 90° F. room temperature and at 15° F. suction gas temperature using a 9 inch blower fan, the unit using my condenser showed a head pressure of 162 pounds per square inch. Under the same conditions, the unit with a standard finned condenser showed a head pressure of 148 pounds per square inch. However, the wattage of the unit with the spiral condenser was 354 watts while the unit with the standard condenser was 365 watts. The pressure drop through the spiral condenser was 8 pounds per square inch whereas it was negligible in the finned condenser. In this case, the spiral condenser consisted of a three inch wide (two inch effective surface) steel strip approximately ten feet long and wound to a spiral form of about nine inches outside diameter. Thus, the total radiating surface was 5 square feet. The finned condenser was formed of 22½ feet of ⅜ inch tubing having six fins and providing a total radiating area of 20 square feet.

To further illustrate the difference in operation of a refrigerating unit employing a condenser constructed in accordance with this invention as compared with a condenser of conventional construction, there is set forth in the table below various operating conditions of two refrigerating units identical in every respect except the condenser. Both units were operated by means of a ¼ H. P. motor at 90° F. room temperature with a 9 inch fan. The condenser of the present invention was identical with the condenser described in the paragraph above having a total radiating surface of 5 square feet and the other condenser was similar to the one described in the paragraph above but having a total radiating surface of 25 square feet.

Suction Temp., ° F.	New Condenser		Conventional Condenser	
	Head Press., lb./sq. in.	Watts	Head Press., lb./sq. in.	Watts
−8½	134	277		
1½	142	310		
10½	157	345	143	340
18½	172	386	152	387
25½	183	433	162	450

At each of the operating pressures set forth in the above table there was no appreciable pressure drop across the condenser of conventional design. Using the above described condenser on a ⅓ H. P. unit, it was found that in a 90° F. room with a head pressure of 157 pounds the pressure at the outlet end of the condenser was about 109 pounds, at 172 pounds head pressure the outlet pressure was 110 pounds, and at 183 pounds head pressure the pressure at the outlet end was 115 pounds. I have determined that a drop in pressure across the condenser of as little as 2% of the head pressure produces the desirable operation described herein. However, a drop in pressure of at least 10% of the head pressure is preferred.

The above data shows that with my condenser the head pressure was increased while at the same time the refrigerating system required less wattage and thus was more efficient as compared with the conventional condenser. With a higher head pressure as indicated above, it is obvious that the temperature of the gas entering the condenser of my construction was higher than the temperature of the gas entering the condenser of conventional construction. Thus, with my construction there is a greater differential in temperature between the refrigerant in the condenser near the inlet end thereof and the ambient temperature within the room where the unit is located. Condensation is produced by removing the superheat from the gaseous refrigerant, and I have found that this superheat can be removed more quickly and more efficiently when the temperature differential is greater. Thus, with a condenser of my construction condensation of the liquid begins to occur at the inlet end of the condenser and progressively increases in the direction of the outlet end. At the same time the pressure in the condenser gradually decreases in a direction toward the outlet end.

I have found by experiment that with a condenser of my construction I am actually able to obtain super-cooling of the refrigerant in the condenser, that is, by reason of the rapid condensation which takes place in a condenser constructed in accordance with this invention, the refrigerant can be cooled to below the ambient temperature of the room; and a re-expansion of the liquid into gaseous refrigerant may be obtained. Accordingly, the construction shown herein may be designed to form a continuous path, a portion of which serves as a condenser and another portion of which serves as an evaporator.

This arrangement is shown in Fig. 9 wherein the conduit 44, which is fashioned in the manner illustrated in Figs. 1 and 2, has one portion 46 thereof wound spirally into the shape of a condenser and connected as by a conduit 48 with the high side of a compressor 50. Another portion of conduit 44 is fashioned into an evaporator 52 which is connected as by a conduit 54 with the low side of compressor 50. An intermediate portion 56 serves to connect the condenser portion 46 with the evaporator portion 52 and serves a function similar to the capillary tube used in conventional refrigerating systems. By proportioning the size and length of the several portions of conduit 44, the system may be operated so that the refrigerant is condensed in condenser 46, the pressure of the liquid refrigerant substantially reduced in the portion 56, and caused to boil in the evaporator portion 52. With this arrangement it will be observed that a single conduit serves as the means for condensing the gas into a liquid, for reducing the pressure thereof, and for re-expanding the liquid.

It will be appreciated that the invention here may take various forms, the gist of the invention primarily residing in the closely spaced walls of the condenser which are preferably fashioned so that throughout the length of the condenser portions the two side walls are in contact or substantially in contact so that condensation is promoted. For example, there is shown in Fig. 3 another form of condensing unit wherein the passageway is formed by two side walls 58 and 60 which are seam welded along their edges as at 62. Side wall 60 is held in closely spaced relation from side wall 58 by a plurality of indentations 64 which are preferably spot welded at the center thereof as indicated at 66 to the side wall 58. As an example of the dimensions of the condensing passageway, a condenser constructed with side walls ten feet long, three inches wide and with a spacing between the side walls of approximately $\frac{5}{32}$ of an inch performs very satisfactorily in a refrigerating unit employing a ¼ H. P. motor. The spot welds 66 are preferred in order to hold the two side walls in a generally spaced apart relation.

In Fig. 4 the cross section of the unit shown in Fig. 3 is illustrated. The passageways between indentations 64 are designated 68 and it will be appreciated that fine crevices are bound to be present between the indentations 64 and the adjacent portion of side wall 58 in which gaseous refrigerant will flow. These crevices indicated in an exaggerated manner at 70, provide a relatively large cooling surface for a very small volume of gas. Thus, as soon as the gaseous refrigerant flows into these small crevices, the gas will be immediately condensed thereby reducing the pressure in these crevices and causing additional refrigerant gas to flow therein and drive out the condensed refrigerant. The refrigerant droplets driven from the crevices 70 into the passageway 68 will absorb heat from the surrounding gaseous refrigerant and promote further condensation.

Alternative constructions for the condenser unit are shown in Figs. 5 and 6. In Fig. 5 side walls 72 and 74 are welded together along their longitudinal edges at 76, and the side wall 74 is provided with a series of corrugations 78 between which the side walls are spot welded together as at 80. In the showing in Fig. 6 one side wall 82 is provided with a series of zig-zag indentations 84 which are spot welded at successive points indicated at 86 to the other side wall 88. In each of these constructions small crevices are formed in the area surrounding the spot welds which promote the condensing action referred to above with reference to Figs. 3 and 4.

Where the condenser is to be subjected to forced or induced air circulation, improved action may be obtained by so forming the spiral as to shroud the member which induces air circulation. Thus, in Fig. 10 a motor-driven screw blade fan 90 is mounted adjacent the face of the spiral and the outer turns 94 of the spiral are displaced laterally of the plane of the inner turns so as to shroud the tips of the fan blades and take advantage of the air circulation which spills off the ends of the fan tips. In Fig. 11 a blower 96 of the Sirocco type is provided with a housing 98 in which a spiral condenser 100 forms a multiple convolution exit path for the output of blower 96. The edges of the spiral condenser are enclosed by flat discs 102 having an air intake opening 104 at the axial center of housing 98 and the air is discharged at the end of the last turn of the spiral through an opening 106. In the form illustrated in Fig. 12 the convection currents induced around a motor compressor unit 108 are utilized to assert circulation through a spiral condenser 110. In the form of construction shown in Fig. 12 the spiral is formed with a large opening 112 at the axial center thereof to receive the compressor unit **108**. The spiral **60**, being mounted in a horizontal plane surrounding the compressor unit **108**, is in the path of the up-draft induced by the heat of the compressor and also by the heat given off by the condenser.

Thus, it will be seen that I have provided a condenser construction which possesses distinct advantages over the condenser construction of conventional design. My condenser unit operates more efficiently. It requires less radiating surface as compared with a conventional condenser of the same capacity and occupies a smaller space. Furthermore, the condenser of my construction is designed so that it will not become clogged by dirt and lint which flow past the spirals or coils of the condenser. No closely spaced radiating fins are required to provide the necessary cooling surfaces.

This application is a continuation-in-part of my prior application Serial No. 733,524, filed March 10, 1947, covering Refrigerator Condenser, now abandoned.

I claim:

1. A refrigerating system of the condenser-evaporator-compressor type characterized in that said condenser comprises a relatively long enclosed passage having an inlet and an outlet, said passage being formed with substantially flat parallel sheet metal walls which are generally spaced apart in close proximity and substantially uniformly throughout the axial extent of said passage a distance greater than a distance such as to feed refrigerant by capillary action but being spaced sufficiently close together such that the predominantly greater portion of cross section of the refrigerant vapor flowing therebetween is scrubbed by said walls and the friction resulting from said scrubbing action produces decreasing pressure along the condenser so that the drop across the condenser is at least substantially 5% of the head pressure of the refrigerant at the inlet of the condenser at 98° F.

2. The combination set forth in claim 1 wherein the passage formed by said sheet metal walls is arranged as a spiral.

3. The combination set forth in claim 1 wherein at least one of said sheet metal walls is provided with indentations intermediate the edges of said sheet metal wall which are in contact with the other sheet metal wall to provide obstructions to the flow of refrigerant through said passage.

4. The combination set forth in claim 3 wherein said indentations are generally in coplanar engagement with said other wall and provide extremely small crevices therebetween into which gaseous refrigerant is permitted to flow, the walls of said crevices being so closely spaced that the gas flowing into said crevices is substantially instantaneously condensed.

5. The combination set forth in claim 4 wherein said side walls are spot welded together at said indentations.

6. The combination set forth in claim 5 wherein the area of said indentations is substantially greater than the area of the spot welds.

7. A refrigerating system of the condenser-evaporator-compressor type wherein said condenser and said evaporator comprise a relatively long enclosed continuously extending passage having one end communicating with the high side of the compressor and the opposite end communicating with the low side of the compressor, said passage being formed of sheet metal walls and being generally divided into three serially connected portions, the portion of said passage connected with the high side of the compressor forming said condenser, the portion of said passage connected to the low side of the compressor being shaped to provide said evaporator and the intermediate portion of said passage forming a connecting conduit between said condenser and said evaporator, said walls being sufficiently closely spaced throughout their length to produce a pressure drop across the extent of said passage which is substantially equal to the difference in pressures between the high and low sides of said compressor, said intermediate and evaporator portions of themselves serving as a means for reducing the pressure of the refrigerant from the high to the low side of the system and the walls of said portion of said passage forming said condenser being sufficiently closely spaced to produce a drop in pressure across the condenser of at least 2% of the head pressure of the refrigerant at the inlet of the condenser at 90° F.

8. The combination set forth in claim 7 wherein at least one side wall of that portion of the passage which forms said condenser is provided with indentations which contact the other side wall of said passage.

9. The combination set forth in claim 7 wherein at least one side wall of the portions of said passage which forms said condenser and said intermediate connecting portion is provided with indentations which contact the other side wall of said passage.

10. A refrigerating system of the condenser-evaporator-compressor type characterized in that said condenser comprises a passageway having an inlet and an outlet, the passageway being defined by sheet metal walls which are spaced apart in close proximity throughout a major portion of the axial extent of said passageway a distance greater than such as to feed refrigerant by capillary action but sufficiently close together to produce a vigorous scrubbing action of a predominantly greater portion of the gas flowing therethrough and produce a substantial pressure drop across the condenser, said walls having a plurality of indentations therein, said walls at said indentations being spaced apart in extremely close proximity such that the gas flowing therebetween from said areas of close proximity is immediately condensed thereby producing localized low pressure areas at said indentations into which surrounding gas rapidly flows to drive the condensed refrigerant out into the surrounding gas where it re-expands, thus producing further condensation and tending to maintain the walls of said condenser wet throughout the axial extent of said passageway.

11. The combination set forth in claim 10 wherein the portion of said walls in close proximity and the portion of said walls in extremely close proximity are sufficiently closely spaced to produce a pressure drop across the condenser of substantially at least 5% of the head pressure of the refrigerant at the inlet of the condenser at 90° F.

12. The combination set forth in claim 11 wherein the spacing of the portions of said wall in close proximity is approximately 5/32".

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