June 23, 1936. R. W. KRITZER ET AL 2,045,529

COMBINED COOLING UNIT AND DRIP PAN

Filed April 15, 1935

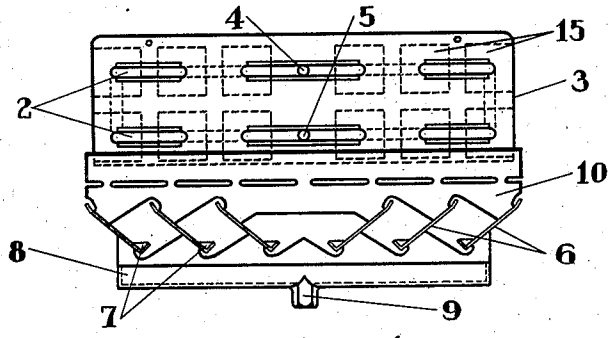


Figure 1

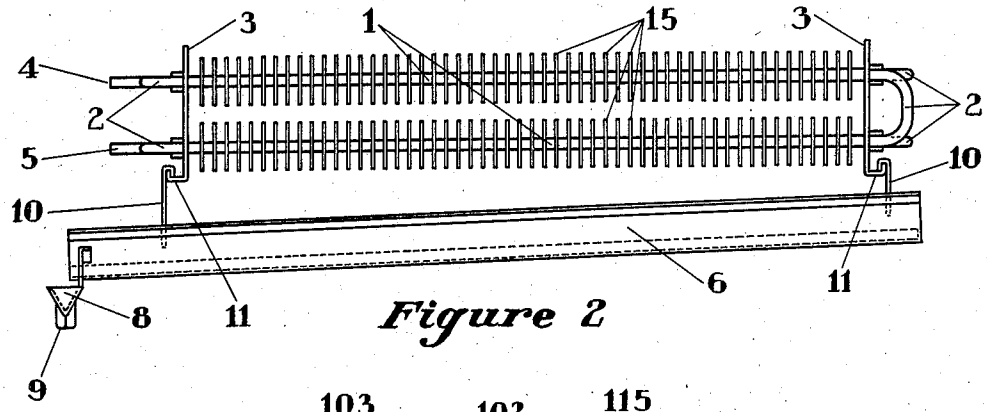


Figure 2

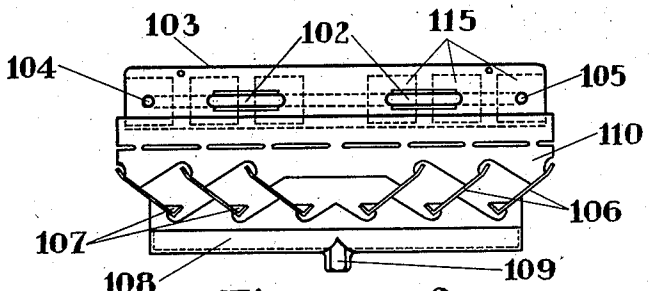


Figure 3

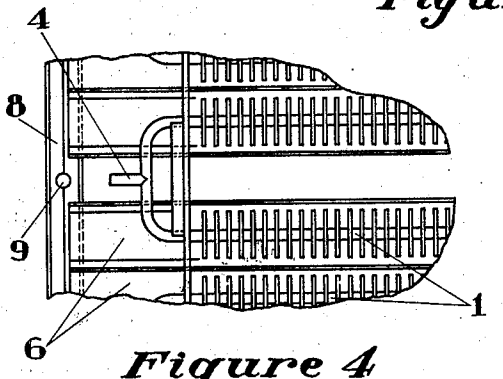


Figure 4

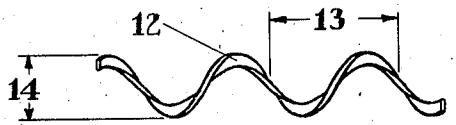


Figure 5

Inventors:

Richard W. Kritzer

Anthony F. Boesel

Patented June 23, 1936

2,045,529

UNITED STATES PATENT OFFICE

2,045,529

COMBINED COOLING UNIT AND DRIP PAN

Richard W. Kritzer and Anthony F. Hoesel, Chicago, Ill., assignors to Peerless Ice Machine Company, Chicago, Ill., a corporation of Illinois

Application April 15, 1935, Serial No. 17,598

9 Claims. (Cl. 62—103)

Cooling units and combination drip pan and air circulating means, for the same, are comparatively old in the art, as disclosed in the Richard W. Kritzer applications No. 711,485 filed February 16, 1934, and No. 712,448 filed February 23, 1934, and No. 715,800 filed March 16, 1934, which are claimed to be improvements upon the prior disclosed art and to which our present joint application relates.

One of the disadvantages of all the combinations used in the art, prior to our joint invention, is the fact that these prior devices would not function correctly whenever compartment temperatures close to 32° F. had to be maintained.

The majority of refrigerating plants, using cooling units and combination drip pan and air circulator means, are automatic in operation. The maximum capacity of the refrigerant circulating compressor and its associated cooling units is in excess of the heat to be absorbed by the cooling units; therefore, the cycle of operation is alternately in the on and off-time cycle.

The major heat load of the refrigerating plants, referred to above, is due to that of the heat in leakage through the floor, ceiling and walls of the compartments. The rate of this heat leakage, per unit area of the boundary surfaces, is entirely dependent upon the type of insulation, the thickness of the insulation and the temperature difference between the outside and inside of the refrigerated compartment.

All of the above mentioned plants are operated in such cycles as to maintain the compartment temperatures between certain definite limits. For instance, in the storage of meats, the individual compartments may be desired to have temperatures maintained between 32° F and 34° F., which gives an average temperature of 33° F. If the compartments were maintained between 29° F. and 37° F., there would also, then, be an average temperature of 33° F.; however, while in the former case the maximum variation in temperature is 2° F., in the latter case the variation in temperature would be 8° F. A minimum temperature variation, as is well known, is the best for minimum dehydration of the stored product and maximum storage time, during which the palatability and fresh appearance of the stored product may be retained. Therefore, there has been a constant endeavor, in the refrigerating art, to decrease compartment temperature variations to the minimum, which would not seriously impair the thermal efficiency of the refrigerating system, increase the cost of operation or add to the complexity of the installation.

In recent years, the efficiency of these refrigerating systems has been greatly improved by the employment of a much larger amount of heat absorption surface in the cooling units, than had been previously used. Since the space used up by the cooling unit detracts from the available space, or at least necessitates a larger gross space than that necessary for storage only in the compartment, it becomes necessary to condense the cooling unit to its lowest possible physical dimensions. Plain refrigerant piping occupies a considerable volume of space per unit surface area; therefore, these plants are equipped with finned refrigerant conduit cooling units, which occupy a much lower volume of space per unit area than the equivalent area of plain piping. These cooling units are termed fin coils.

Whenever the surface area of a cooling unit is condensed, in its boundary dimensions, such as in a fin coil, there is comparatively small space between adjacent surfaces, and since, during the on-cycle time, these cooling units are maintained at a temperature appreciably lower than 32° F., frost forms upon these surfaces.

If this frosting is cumulative, during each on-cycle time, it will not be long before the spaces, between adjacent surfaces and for the purpose of allowing air circulation therethrough, will be completely filled with frost.

During cumulative frosting, and the filling of the spaces thereby, the original surface area of the cooling unit is substantially decreased, which by decreasing the heat absorption capacity of the cooling unit, cuts down the efficiency of the refrigerating system, increases the cost of operation and possibly, in some instances, no longer allows the maintenance of the desired temperatures.

Because of the above, it may be readily realized that a complete defrosting of the cooling unit surfaces, during each off-time cycle, is the most desirable condition for the maximum efficiency.

In order to clearly portray the actual conditions affecting the defrosting of cooling units, we shall take a typical example. Assume a compartment having a heat load rate of 6,000 B.t.u. per hour. Assume a cooling unit in the compartment and, with its associated compressor, having a heat absorption capacity of 8,000 B.t.u. per hour. The on-time cycle will then be 75 percent of the total time.

Supposing that, from the start of the off-time cycle, it takes 10 minutes for the cooling unit to entirely defrost the moisture deposited upon its

surfaces during the on-time cycle, then in order to get a thorough defrosting, at the 75 percent total operating time necessary to maintain the desired compartment temperatures, the on-time cycle will have to be not less than 30 minutes. If, under the above conditions, a temperature variation of 4° F. obtains in the compartment and a temperature variation not to exceed 2° F. must be attained, then it is imperative that some means be employed to cut the defrosing time in half, allowing an on-time cycle time of 15 minutes and an off-time cycle of 5 minutes.

All of the foregoing relates to the drip pan as well as the cooling unit since, during the defrosting of the cooling unit, all of the deposited moisture does not drop from the cooling unit in the form of water. During the operation of the cooling unit, small icicles form at the lowermost edges of the cooling unit, and whenever they attain a certain size, they, during the off-time cycle, drop into the drip pan, where, if the drip pan is not at a sufficiently elevated temperature to thoroughly melt them before the start of the on-time cycle, there will be a gradual build up of ice in the drip pan channels and eventually the channels will overflow and defeat their purpose.

One of the objects of our joint invention is to cut down the time necessary for the defrosing of a finned coil cooling unit.

Another object is to increase the volume of the convection circulation of air through a cooling unit.

Another object is to increase the heat absorption capacity of a finned refrigerant conduit cooling unit.

A further object is to quickly raise the temperature of a drip pan during the off-time cycle of a refrigerating system.

Referring to the drawing:

Figure 1 is an end elevational view of a cooling unit and combination drip pan and air circulator for the same.

Figure 2 is a side elevational view of Figure 1.

Figure 3 is an end elevational view of a modified cooling unit for comparatively low temperature use.

Figure 4 is a fragmentary plan view of Figures 1 and 2 and illustrates the manifolding of the right hand and left hand sections of the cooling unit to a single inlet.

Figure 5 shows a spiral ribbon, which is inserted into the refrigerant conduit to provide a spiral passage adjacent the internal walls of the conduit.

In Figures 1 and 2, the parallel reaches of the refrigerant conduit 1 are joined by return bends 2, which pass through the suspension end plates 3, thereby making a continuous refrigerant circuit in each of the right and left-hand sections, shown in Figure 1, which are manifolded at their inlets 4 and their outlets 5. Upon the parallel conduit reaches 1, are mounted a multiplicity of spaced apart rectangular fins 15, which increase the effective heat absorption surface of the conduits 1. The above comprises the cooling unit.

Directly below the cooling unit, and suspended from the suspension end plates 3, is a combination drip pan and air circulator comprised of a plurality of louvre channels 6, of which the lower edges 7, being bent upwardly, provide a trough for receiving the moisture dripping from the cooling unit during the off-time cycle. All of the channels spill into a common collector 8, having an outlet 9 which can be connected to a sewer.

The hangers 10 serve to maintain the channels 6, which are fastened thereto, in proper spaced relationship to each other. Flanged edges 11, in the lower portion of the suspension end plates 3 and the upper portion of the hangers 10, are the suspension means for the drip pan assembly.

In Figure 3, all of the elements of Figures 1 and 2 obtain and, therefore, 100 has been added to the number of the similar elements of Figures 1 and 2.

Figure 5 shows a spiral ribbon 12 having a helical pitch 13 and an outside diameter 14, which is slightly greater than the internal diameter of the conduit 1, into which it is inserted. The purposes and advantages of this ribbon, or its equivalents, is clearly explained in the Richard W. Kritzer and Anthony F. Hoesel joint application, Serial No. 758,065, filed December 18, 1934, for Conduit for the circulation of refrigerants.

During the operation of the device, the circulation of refrigerant through the conduits lowers the temperature of the cooling unit surfaces, thereby cooling the air immediately in contact therewith. This cooling of the air reduces its volume and consequently increases its weight per unit volume, which tends for the cooled air to drop through the cooling unit, allowing warmer and lighter air to pass into the cooling unit. This convection circulation of the air is continuous as long as the cooling unit temperature is below that of the air temperature.

If the cooling unit surfaces, during the operation, did not drip moisture, there would be no need for any auxiliary device below the cooling unit. However, because of moisture drip, it becomes necessary to provide some means of catching the same and allowing it to pass to a drain or otherwise.

The circulation of the air, during the operation of the cooling unit, has a comparatively low velocity and, therefore, any obstructions placed in the line of natural air circulation tend to restrict the air flow and decrease the heat absorption capacity of the cooling unit.

In the drawing 1 and 3, it will be noticed that the fins 15, of adjacent parallel reaches 1, are substantially separated. This separation performs two functions; first, the separation provides a path for comparatively warm air, the circulation of which is induced by the descending cold air stream adjacent the surfaces; second, the separation provides a greater area for warm air intake to the cooling surfaces. In the first instance, the induced circulation of warm air bathes the louvre channels 6 and keeps them at an increased temperature above that of the temperature of the cold air stream passing between adjacent louvre channels. This allows any small ice particles, which may drip from the cooling unit during the off-time cycle, to thoroughly melt and pass in liquid form through the drain connection 9, before the start of the on-time cycle. This particular feature allows us to operate with much narrower compartment temperature differentials than the structures disclosed in the previously mentioned applications, which tend to cumulatively refreeze these ice particles until the troughs shown at 7 and 107 become filled and the drip then overflows onto the floor of the compartment or the goods stored therein, which condition can only be overcome by increasing the compartment temperature differentials between the on-time cycle and off-time cycle, unless the present invention is employed.

The spiral ribbon 12, inserted within the conduits of the cooling units, increases the internal, refrigerant wetted, surface of the conduits and thereby increases the heat absorption of the cooling unit.

In actual tests of cooling units without this spiral ribbon, we have found out that the heat absorption capacity per degree Fahrenheit dropped rather sharply at comparatively high temperature differentials between the temperature of the air and the temperature of the circulating refrigerant. We definitely determined that this loss in efficiency occurred due to conditions within the conduit, rather than conditions outside the conduit, and the results of our experiments are embodied in the spiral ribbon and its equivalents as disclosed in our previously mentioned joint application.

While the above describes the preferred embodiment of our joint invention, we do not wish to limit it thereto, but rather to the specific combinations set forth within the following claims.

We claim:

1. The combination of a refrigerant conduit cooling unit having spaced apart parallel reaches with a multiplicity of spaced apart fins mounted thereon and individual to each reach, the spacing of the parallel reaches being substantially greater than the fin dimensions parallel to the line of spacing and a multiple louvred drip pan below the assembly, and each bottom parallel reach of the cooling unit being served by a single louvre.

2. The combination of a refrigerant conduit cooling unit having spaced apart parallel reaches with a multiplicity of spaced apart fins mounted thereon and individual to each reach, the spacing of the parallel reaches being substantially greater than the fin dimensions parallel to the line of spacing and a multiple louvred drip pan below the assembly, and each bottom parallel reach of the cooling unit being served by a single louvre, having a projected horizontal width greater than the width of the fin.

3. The combination of a finned refrigerant conduit cooling unit having plural sections manifolded to a single inlet and mounted upon common spacing means, a multiple louvred drip pan below the assembly, and each bottom reach being served by a single louvre.

4. The combination of a refrigerant conduit cooling unit having spaced apart parallel reaches with a multiplicity of spaced apart fins mounted thereon and individual to each reach, the spacing of the parallel reaches being substantially greater than the fin dimensions parallel to the line of spacing and a multiple louvred drip pan below the assembly, and each bottom parallel reach of the cooling unit being served by a single louvre, the said conduit having a spiral passage adjacent its internal walls.

5. The combination of a refrigerant conduit cooling unit having spaced apart parallel reaches with a multiplicity of spaced apart fins mounted thereon and individual to each reach, the spacing of the parallel reaches being substantially greater than the fin dimensions parallel to the line of spacing and a multiple louvred drip pan below the assembly, and each bottom parallel reach of the cooling unit being served by a single louvre, having a projected horizontal width greater than the width of the fin, the said conduit having a spiral passage adjacent its internal walls.

6. The combination of a finned refrigerant conduit cooling unit having plural sections manifolded to a single inlet and mounted upon common spacing means, a multiple louvred drip pan below the assembly, and each bottom reach being served by a single louvre, the said conduit having a spiral passage adjacent its internal walls.

7. The combination of a refrigerant conduit cooling unit having spaced apart parallel reaches with a multiplicity of spaced apart fins mounted thereon and individual to each reach, the spacing of the parallel reaches being substantially greater than the fin dimensions parallel to the line of spacing and providing a space between the fins of adjacent conduit reaches and a multiple louvred drip pan below the assembly, the upper edge of the louvre being positioned within the vertical prolongation of said space.

8. The combination of a refrigerant conduit cooling unit having spaced apart parallel reaches with a multiplicity of spaced apart fins mounted thereon and individual to each reach, the spacing of the parallel reaches being substantially greater than the fin dimensions parallel to the line of spacing and providing a space between the fins of adjacent conduit reaches and a multiple louvred drip pan below the assembly, and each bottom parallel reach of the cooling unit being served by a single louvre, the upper edge of the louvre being positioned within the vertical prolongation of said space.

9. The combination of a refrigerant conduit cooling unit having spaced apart parallel reaches with a multiplicity of spaced apart fins mounted thereon and individual to each reach, the spacing of the parallel reaches being substantially greater than the fin dimensions parallel to the line of spacing and providing a space between the fins of adjacent conduit reaches and a multiple louvred drip pan below the assembly, and each bottom parallel reach of the cooling unit being served by a single louvre, having a projected horizontal width greater than the width of the fin, the upper edge of the louvre being positioned within the vertical prolongation of said space.

RICHARD W. KRITZER.
ANTHONY F. HOESEL.