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(54) HEATING AND REFRIGERATION SYSTEMS USING REFRIGERANT MASS FLOW

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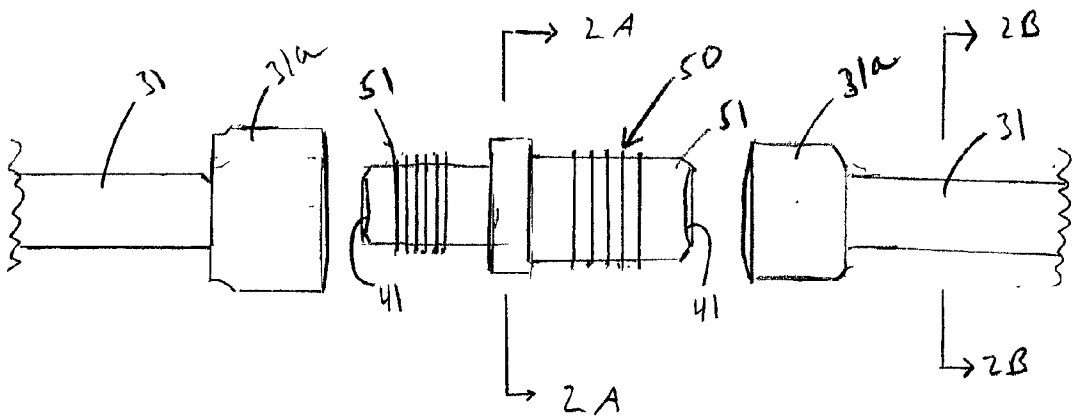
(57) ABSTRACT

Vapor compression heat exchange systems are disclosed that are designed to allow for optimal mass flow of refrigerant there through. The systems of the present invention do not employ conventional refrigerant metering devices, such as capillary tubes and expansion valves, which restrict mass flow, but rather incorporate an openly fixed orifice in-line with the conduits connecting the condenser to the evaporator, thereby maintaining the pressure differential between the high pressure condenser side and low pressure evaporator side of the system during operation. Provision of the fixed orifice allows for optimal refrigerant mass flow as measured by cooler compressor temperatures, cooler compressor discharge temperatures, increased heat of rejection, increased heat of absorption, and improved heating and cooling efficiency. The present invention may employ any conventional refrigerant, including the newer HFC refrigerants, such as R-410A.



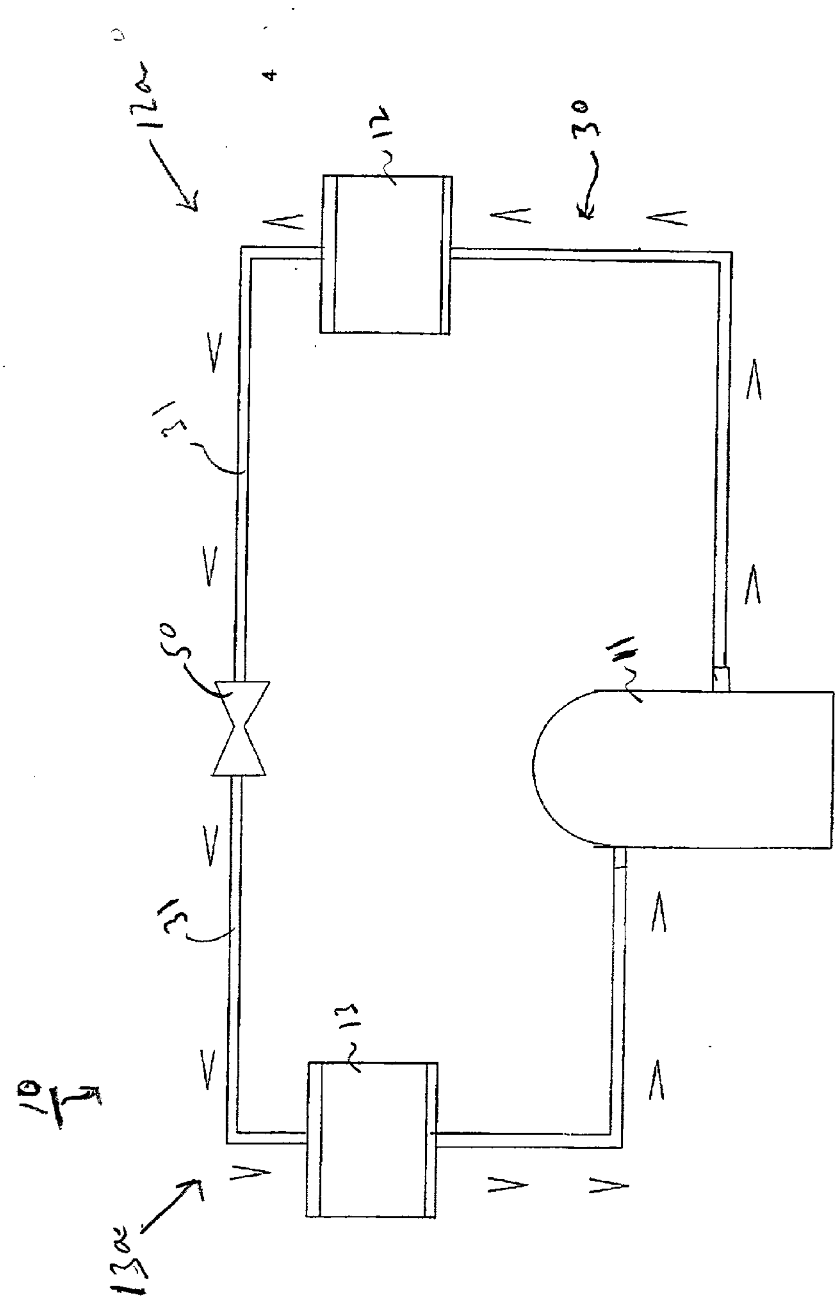


FIG. 1

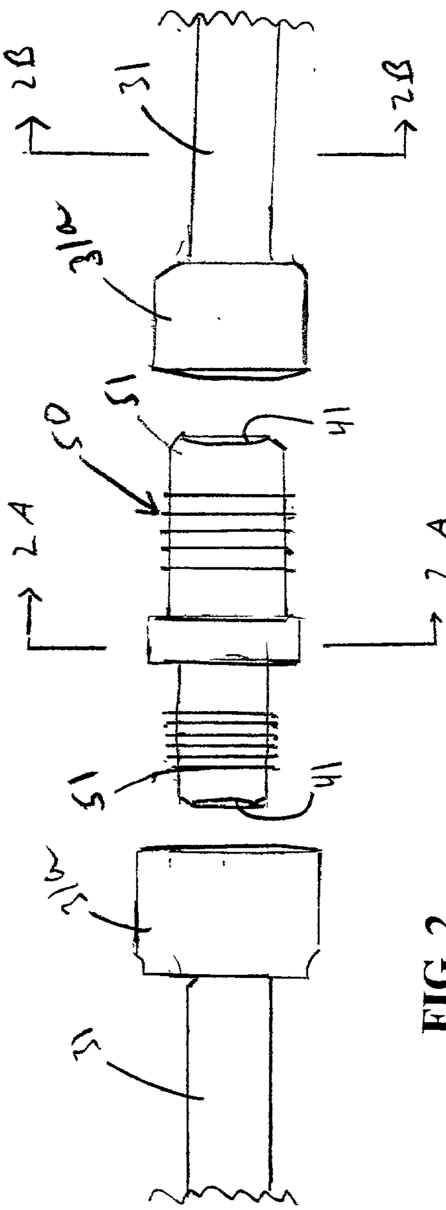


FIG. 2

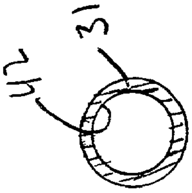


FIG. 2B

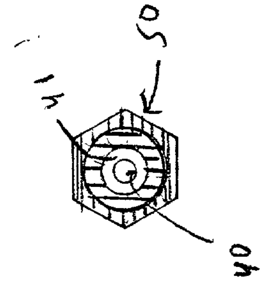


FIG. 2A

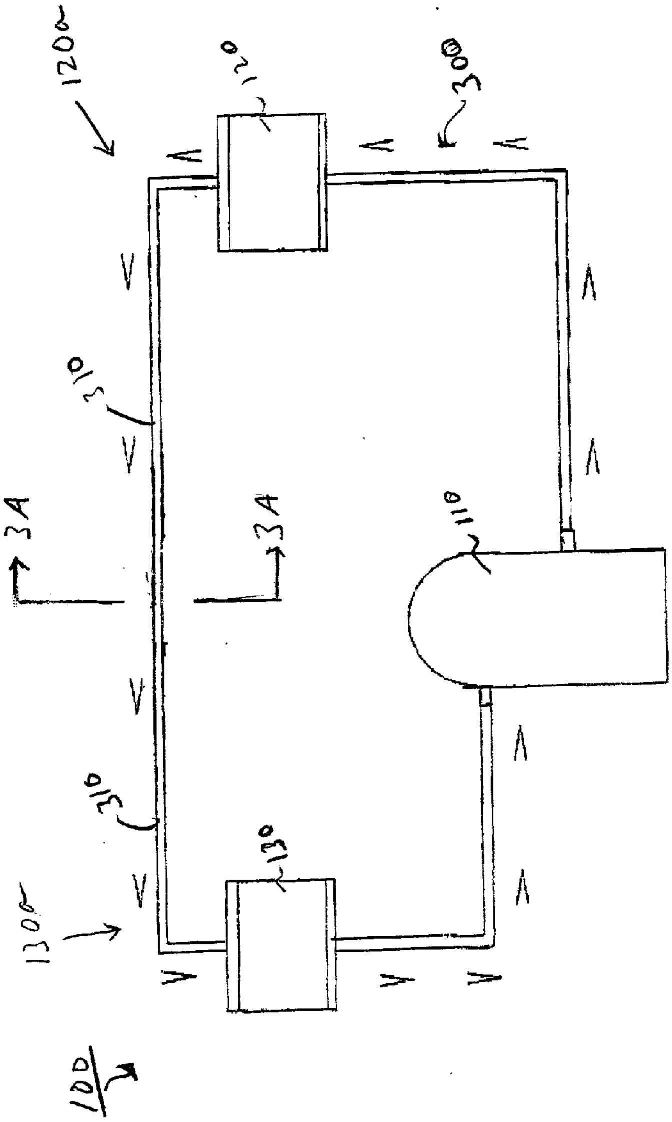


FIG. 3

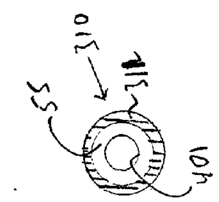


FIG. 4

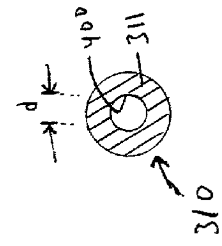


FIG. 3A

HEATING AND REFRIGERATION SYSTEMS USING REFRIGERANT MASS FLOW

BACKGROUND OF THE INVENTION

[0001] A. Basic Components of Conventional Heating and Cooling Systems:

[0002] Most conventional heating and cooling systems comprise a motorized compressor, an evaporator, a condenser, and a series of conduits in communication with the compressor, evaporator, and condenser. The conduits carry high pressure refrigerant gas from the compressor to the condenser, where the gas is condensed to a liquid upon dissipation of latent heat from the condenser. The refrigerant liquid is then carried through a refrigerant metering device to the evaporator, wherein latent heat is absorbed to vaporize the refrigerant. The resulting low pressure refrigerant gas is then suctioned into the compressor where it is compressed into a high pressure gas again, and the cycle is repeated.

[0003] The refrigerant metering device allows the refrigerant to flow from the high pressure condenser side to the low pressure evaporator side and functions, in part, to maintain this pressure differential while the compressor is operating. In the evaporator, the liquid refrigerant, under a much reduced pressure, absorbs heat and is vaporized very rapidly, thereby cooling the evaporator. The compressor creates a low pressure by suctioning the refrigerant gas from the evaporator and then compresses the gas back to a high pressure gas.

[0004] Typical refrigerant metering devices include expansion valves (e.g. automatic expansion, thermostatic expansion, and thermal electric expansion) and capillary tubes. To regulate refrigerant flow, the expansion valve uses feedback about conditions in the evaporator and/or suction line. The expansion valves are the most popular type of metering devices and provide for operation over a wide range of temperature conditions. Expansion valves are also used in these systems to throttle back or restrict refrigerant flow in order to prevent flooding of the compressor, the belief being that such flooding will compromise the operation of most compressors and ultimately cause damage to the entire system.

[0005] The capillary tube, on the other hand, has a fixed diameter restriction which serves as a constant throttle or restriction on the flow of refrigerant. The length, inside diameter, and configuration of the capillary tube as well as the operating temperature and pressure differential across the capillary tube affect the rate of refrigerant flow. Thus, unlike the expansion valve, the capillary tube cannot be adjusted and is only effective over a much narrower temperature range for a given compressor.

[0006] B. Refrigerants

[0007] For years the most common refrigerant of choice in heat pump and air-conditioning systems has been R-22, a hydrochlorofluorocarbon (HCFC). HCFC's are not as harmful to the ozone layer as chlorofluorcarbons (CFC), which have been completely phased out since 1996; however, HCFC's still contain ozone-destroying chlorine. Thus, pursuant to 1992 amendments to the Montreal Protocol [an international environmental agreement requiring worldwide phase out of ozone-depleting CFC's], there has been an established schedule for the phase-out of HCFC's.

[0008] As a result of the gradual phase out of chlorine-containing refrigerants, non-ozone-depleting alternative refrigerants have been introduced. One substitute deemed acceptable by the EPA is R-410A, a blend of hydrofluorocarbons (HFC's) that do not contribute to the depletion of the ozone layer. R-410A, which will soon be required in all newly installed heating and cooling systems, is also a very efficient refrigerant, having up to 30% greater heat capacity. One drawback to the use of R-410A and similar refrigerants, however, is that current R-22 heat pump and air-conditioning systems using conventional metering devices must be re-designed to accommodate the high pressure characteristics of these refrigerants. In fact, the use of R-410A in conventional heat pump and air conditioning systems often results in the burn out of the compressors due to a number of factors, such as increased temperature and pressure of the system and the absorption of water into the compressor due to the presence of synthetic lubricating oils which must be used with these high pressure refrigerants. Such redesigns of current heating and cooling systems will result in huge financial expenditures in terms of factory re-tooling as well as retraining of service personnel.

[0009] In view of the foregoing difficulties with the newer HFC refrigerants, it is therefore desirable to incorporate minor, less expensive modifications to existing systems that will allow for the accommodation of these refrigerants without sacrificing cooling and heating efficiency and without damaging the system itself. The present invention will accommodate these new refrigerants as well as the current HCFC refrigerants such as R-22 with minimal modification to the existing heating and cooling designs.

SUMMARY OF THE INVENTION

[0010] The present invention is directed, in certain aspects, to vapor compression heat exchange systems comprising, in part, the components common to many current heating and cooling systems, namely, a compressor, an evaporator, a condenser, refrigerant, and a series of conduits for carrying refrigerant throughout the system. Unlike conventional heating and cooling systems, however, the present invention does not include conventional metering devices, such as capillary tubes, expansion valves, and the like, which restrict mass flow of refrigerant. Instead, the present invention is designed to provide for increased mass flow of refrigerant throughout the heating or cooling system, in part, by replacing conventional refrigerant mass flow restricting metering devices with a fixedly open orifice disposed in-line with the conduits connecting the evaporator and condenser. The orifice functions to maintain the necessary pressure differential between the high pressure condenser side and the low pressure evaporator side. The combination of an optimal orifice diameter size, refrigerant charge, and larger diameter conduits provides increased mass flow of refrigerant throughout the system that is about four to five times faster than refrigerant flow through prior art systems employing conventional expansion valves or capillary tubes. Contrary to what is believed by those of skill in the art, namely that refrigerant mass flow should be restricted at times to minimize condenser flooding, the inventor has discovered that increasing refrigerant mass flow provides a number of benefits without compromising the life of the compressor. Some of these advantages are as follows:

[0011] (1) By increasing mass flow (i.e. moving a greater volume of refrigerant per unit time through the system), a greater volume of refrigerant per unit time is presented to the respective heat exchangers (i.e. evaporator and condenser), thereby optimizing the refrigerant's ability to absorb and dissipate heat through the evaporator and condenser, respectively. Stated another way, greater refrigerant mass flow increases the heat of rejection at the condenser and increases heat absorption at the evaporator, thereby making the present invention useful for both cooling and heating applications. In swimming pool heat pump applications, for example, the inventor has been able to achieve close to 100% exchange of heat from the refrigerant in the condenser to the water coil (typical efficiencies in prior art systems range from 50% to 85%).

[0012] (2) Increased mass flow reduces the operating pressure of R-410A and other HFC refrigerants, thereby allowing the substitution of non-synthetic lubricating oils, such as mineral oil, for more expensive synthetic oils, such as polyolester (POE) (POE is recommended for use with R-410A since it does not break down under high temperatures like mineral oil). A problem with POE, in addition to its relative high monetary cost, is that it has a high tendency to absorb water, which is detrimental to the compressor. The present invention allows for the use of the less expensive and less hygroscopic oils, such as mineral oil, when R-410A, for example, is employed without the fear that it will break down during operation of the system.

[0013] (3) The refrigerant entrains the lubricating oil to give ample lubrication of the compressor, even for oils that are not miscible with the refrigerant.

[0014] (4) The compressor is adequately cooled by high flow suction cooling, thereby extending the life of the compressor. While the compressors in conventional systems are warm to the touch during operation (up to 200° F.), the compressor in the present invention is quite cool (approximately 65° F.).

[0015] (5) The result of high mass flow allows compressor discharge temperatures to be kept far below temperatures achieved using conventional industry methods.

[0016] It should be pointed out that while a capillary tube also provides for a narrow fixed diameter through which refrigerant flows between the high and low pressure sides of the system, the mass flow of refrigerant is much more restricted through the capillary tube due to friction within the length of the capillary tubing. A fixedly open orifice, on the other hand, maintains a pressure differential between the high and low pressure sides, but allows refrigerant to flow through with less restriction—that is, there is no additional length of narrow tubing to slow the refrigerant down. Stated another way, it takes longer for the refrigerant to flow through the long capillary tube compared to the orifice, which essentially has no length (less than one inch), and thus no frictional impedance.

[0017] A key feature of the present invention is that it can easily be incorporated into existing vapor compression heat exchange designs for use with conventional refrigerants, such as HCFC's, as well as HFC's, such as R-410A. This is significant since current teaching in the industry is to extensively redesign current systems, resulting in increased costs due to factory re-tooling and the re-training of service personnel. By simply replacing existing refrigerant metering devices with a fixedly open orifice having a diameter that provides for optimal refrigerant mass flow, as measured by the compressor discharge temperature, compressor temperature, and heating or cooling efficiency achieved, for example, current heat exchange systems may be able to handle effectively all refrigerants on the market, including HFC's.

BRIEF DESCRIPTION OF THE FIGURES

[0018] **FIG. 1** is a schematic representation of the major components of one embodiment of the inventive heat exchange system.

[0019] **FIG. 2** is an enlarged exploded view of an orifice coupler disposed between two conduits.

[0020] **FIG. 2A** is a transverse section take along lines **2A-2A** of **FIG. 2** showing the open orifice.

[0021] **FIG. 2B** is a transverse section of a conduit taken along lines 2B-2B of **FIG. 2**.

[0022] **FIG. 3** is a schematic representation of the major components of an alternative embodiment of the inventive heat exchange system.

[0023] **FIG. 3A** is an enlarged view of a transverse section take along lines **3A-3A** of **FIG. 3** showing the open orifice within the conduit.

[0024] **FIG. 4** is a transverse section view taken of a third embodiment of the orifice provided by a cap disposed within the conduit.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

[0025] Referring now to the figures, the present invention, in certain aspects, is directed to vapor compression heat exchange systems (**10, 100**) designed to allow for an increased optimal mass flow of refrigerant there through for optimal efficiency at lower operating pressures and temperatures. The inventive vapor compression heat exchange systems include, but are not limited to, air-to-air and liquid-to-air heating or cooling systems for heating or cooling, respectively, air in confined spaces (e.g. building interiors) and air-to-liquid and liquid-to-liquid heating or cooling systems for heating or cooling, respectively, liquids in confined spaces (e.g. industrial liquids, water in swimming pools/spas, water in fish tanks, etc.). The vapor compression heat exchange systems of the present invention may be one application systems (i.e. only heating systems or only cooling systems) or reverse-cycle systems. The examples and orifice dimensions described herein are preferred for 4-ton to 6-ton heat exchange systems; however, the present invention may be applied to smaller systems (e.g. home or commercial refrigerators) as well as much larger (e.g. 12-ton to 500-ton or larger), as discussed further below.

[0026] Like conventional heating and cooling systems, the present invention comprises a condenser (**12, 120**), an evaporator (**13, 130**), and a compressor (**11, 110**). Depending upon the size of the system, more than one evaporator and/or condenser may be employed. A series of conduits (**30, 300**) is also employed throughout the system for carrying refrigerant from the compressor (**11, 110**) to the respective heat exchangers (i.e. evaporator and condenser). Unlike conventional vapor compression heating and cooling systems, however, the systems of the present invention do not incorporate any conventional refrigerant metering devices, such as expansion valves and capillary tubes, which restrict refrigerant mass flow. Contrary to popular thinking in the industry, it has been discovered by the inventor that optimal mass flow of refrigerant is not detrimental to the compressor and, in fact, increases the overall life of the compressor while at the same time provides for increased heat of rejection (from the condenser), increased heat of absorption (from evaporator), and improved heating (or cooling) efficiency.

[**0027**] To achieve an optimal mass flow of refrigerant, thereby resulting in improved heating (for heating systems) or cooling (for refrigeration systems) as well as cooler compressor and compressor discharge temperatures, the present invention incorporates a fixedly open orifice (**40, 400, 401**) disposed in-line with the one or more conduits (**31, 310**) connecting the evaporator (**13, 130**) and condenser (**12, 120**). The function of the fixedly open orifice (**40, 400, 401**) is to allow for optimal increased mass flow while maintaining the required pressure differential between the high pressure condenser side (**12***a,* **120***a*) and low pressure evaporator side (**13***a,* **130***a*) of the system.

[**0028**] As used herein, the term "fixedly open" refers to the state of the orifice's diameter size (d) (see **FIG. 3**, for e.g.) during operation of the heat exchange system, namely that the orifice (**40, 400**) remains fully open during the system's operation. In other words, the orifice diameter (d) does not adjust during system operation like an expansion valve but is fixed in one open position. In addition, there are no valves, pistons, or the like that cover or block the orifice during system operation. Thus, the orifice is in a fixed, open position while the system is operating. Moreover, the term "orifice" refers to a single opening through which refrigerant can flow and which has at most a negligible length relative to the size of the heat exchange system (less than one inch, for example, for 4- to 6-ton systems). The "orifice" is distinguishable from a "tube," such as a capillary tube, which is not a single opening but rather comprises an inner channel in communication with the two openings at opposing ends.

[**0029**] As shown in **FIG. 2**, one device for providing the orifice of the present invention is an orifice coupler (**50**). The coupler (**50**) has two open ends (**51**), typically cylindrical and preferably threaded, for engagement with the respective ends (**31***a*) of the conduits (**31**) connecting the evaporator (**13**) and condenser (**12**). The coupler (**50**) may be designed such that it is removable from the system or permanently welded to the conduits. The coupler (**50**) has an inner channel (**41**) slightly narrower in diameter than the channel (**42**) of the connecting conduits (**31**) and a centrally disposed orifice (**40**) having a negligible length (less than one inch) and an internal diameter ranging from about 0.120 inch to about 0.250 inch (for 4-ton to 6-ton heat exchange systems). Larger orifice diameter sizes (and/or an orifice per conduit for multiple conduits running in parallel between the condenser and evaporator) are required for larger systems. Similarly, smaller orifice diameters are employed for smaller heat exchange systems (e.g. refrigerators). While a coupler is preferred, it will be appreciated by those of skill in the art that other means for providing the orifice may be provided, including, but not limited to, the provision of one or more O-rings or a cap (**55**) engaged within the conduit (**310**) and held therein against the inner walls (**311**) of the conduit as shown in **FIG. 4**. In the case of the cap (**55**), a suitably sized orifice (**401**) is drilled there through. The conduit may also be provided with a crimp or thickened inner wall (**311**) to form a suitable orifice (**400**) therein (see **FIG. 3A**). A removable orifice coupler (**50**) as shown and described herein, however, is preferred when the orifice diameter is being adjusted for a given system. While the orifice is fixedly open as discussed above, it will be appreciated by the skilled artisan that an orifice coupler, for example, may be designed such that the orifice diameter size can be adjusted for a particular system prior to operation (not shown). However, even the orifice of an adjustable orifice coupler will remain fixedly open at the selected diameter size during the operation of the system—that is, it will not widen or narrow per a set of feedback conditions while the system is operating.

[**0030**] For optimal refrigerant mass flow, an orifice diameter size is selected that provides for a cool compressor head, a suitable compressor discharge temperature (typically 180° F. or less), increased heat of rejection in BTU's (from condenser) and/or increased heat of absorption in BTU's (from evaporator), and optimal heating or cooling efficiency (approximately 25% increased efficiency over prior art systems). Various conditions will affect the orifice size ultimately selected, such as the refrigerant used, the refrigerant charge, and the type of compressor.

[**0031**] Table 1 provides optimal orifice diameter sizes for a given compressor, refrigerant, and refrigerant charge. It will be recognized by those of ordinary skill in the art, however, that orifice sizes (generally within a 0.120 to 0.25 inch range for systems up to 6 tons) may be adjusted for other compressors, refrigerants, and temperatures, to obtain optimal refrigerant mass flow, thereby resulting in increased heat of rejection (condenser), increased heat of absorption (evaporator), and maximum efficiency while maintaining cooler compressor and compressor discharge temperatures. As a rule of thumb, heating and cooling systems having longer conduit systems will require larger orifice sizes to obtain optimal mass flow. Larger compressors and overall systems also require larger orifice sizes. Table 1 thus provides a good guide of starting point orifice sizes for systems comprising different compressors and refrigerants. As discussed above, the optimal orifice diameter size selected is based upon a balance between optimal cooling (or heating) efficiency, increased heat of rejection in BTU's and/or increased heat of absorption in BTU's, cool compressor head (cold to touch), and cooler compressor discharge temperatures (preferably less than 180° F.).

[**0032**] Generally, only one orifice is required to be disposed within the evaporator/condenser connecting conduit (**31, 310**). However, where two or more conduits are employed, in parallel, to connect multiple evaporators to multiple condensers (not shown), one orifice coupler, for example, per conduit may be installed. This is usually the case for larger systems using very large compressors and multiple evaporators.

[0033] Conventional heating and cooling systems are charged with refrigerant according to weight. For example, a 6 ton system would be charged with about 4 pounds of R-22 refrigerant. For the present invention, however, the inventor has discovered that for optimal results, charging the system with a selected refrigerant to obtain a certain condenser pressure for a given liquid or air temperature to be heated or cooled is preferred. Table 2 lists preferred charging pressures for a given water or air temperature to be heated.

TABLE 1

Nominal Compressor Size	Refrigerant	Orifice size	Refrigerant Charge	Total Heat of Rejection
51,000 BTU	R-410A	0.120 inch (31 drill size)	See Table 2	92,000 BTU
55,000 BTU	R-410A	0.120 inch	See Table 2	99,000 BTU
63,000 BTU	R-410A	0.120 inch (31 drill size)	See Table 2	123,000 BTU
67,000 BTU	R-410A	0.136 inch (29 drill size)	See Table 2	136,000 BTU

[0034]

TABLE 2

Pressure chart for charging systems in relation to water or air temperature R-410A

Water or air input temperature (° F.) to condenser	Charge to high side pressure[1]
50	242.9 PSI
60	270.3 PSI
70	301.2 PSI
80	335.9 PSI
90	374.5 PSI
100	417.5 PSI

[1]For a given water or air temperature, a sufficient amount of refrigerant is added to achieve the pressure shown in the table.

[0035]

TABLE 3

Pressure chart for charging systems in relation to water or air temperature R-22

Water or air input temperature (° F.) to condenser	Charge to high side pressure[1]
50	165 PSI
60	185 PSI
70	210 PSI
80	235 PSI
90	275 PSI
100	320 PSI

[1]For a given water or air temperature, a sufficient amount of refrigerant is added to achieve the pressure shown in the table.

[0036] Conventional heating and cooling systems comprise conduits of different internal diameters, depending upon their respective locations within the system. For example, for up to 6-ton size systems, the conduit between the evaporator and compressor has an internal diameter ranging from ¾ to ⅞ inch. For the same systems, the conduit between the condenser and the refrigerant metering device has an internal diameter ranging from ½ to ⅝ inch. And the conduit between the evaporator and the refrigerant metering device has an internal diameter ranging from ⅜ to ½ inch. For optimal mass flow in the present invention, the conduits used should be on the larger side of the diameter range for a particular conduit location. In addition, while any conventional evaporator and condenser may be used in the present invention, it is important for optimal refrigerant mass flow and efficiency that the selected evaporator and condenser be suitably matched for the system based upon a given compressor size and conditions.

[0037] While the goal in the present invention is to achieve optimal mass flow of refrigerant, which compared to conventional heating and cooling systems in the prior art is a substantially increased mass flow speed, it is important to note that "optimal" mass flow does not necessarily equate to the "fastest" possible mass flow. That is, if there is too much mass flow, too much sub cooling of refrigerant will result, further resulting in the compressor shutting down since there is no heat to pump out of the compressor for the recycling to begin. Too much mass flow is evidenced by decreased cooling or heating efficiency.

[0038] The present invention will work with any refrigerant known to those of skill in the art used in vapor compression heat exchange systems, including, but not limited to, hydrochlorofluorocarbons (HCFC), hydrofluorocarbons (HFC), chlorofluorocarbons, and natural working fluids such as carbon dioxide, hydrocarbons (e.g. propane), and ammonia, and blends thereof (e.g. HFC/hydrocarbon blends, such as R-407C). Exemplary HFC's include R-410A, R-410B, R-134a, R-152a, R-32, R-125. However, as discussed above, the present invention is particularly advantageous over conventional systems in that it works well with the newer, soon to be required, HFC refrigerants, such as R-410A, which operate at high pressures and temperatures in conventional heating and cooling systems. For example, typical high end pressures for R-410A range from 600 to 800 PSI in conventional systems. In the present invention, high end pressures for R-410A drop to 450 PSI. Similar pressure drops occur with R-22, which typically runs about 100 PSI less than R-410A on the high pressure end. Thus, one aspect of the present invention is a simple method of modifying existing heating and cooling systems by replacing the conventional metering device with an orifice coupler or providing an orifice within the conduit system as described herein to maintain the required pressure differential between the high and low pressure ends of the system as well as removing all other refrigerant flow restrictions while allowing for optimal refrigerant mass flow.

[0039] Another feature of the present invention is that all compressor lubricating oils may be employed (both miscible and non-miscible with the refrigerant). In conventional heat exchange systems, POE is recommended if not required for use with high pressure HFC refrigerants, such as R-410A, since it does not break down at high temperatures like some of the non-synthetic oils, such as mineral oil. The present invention, however, will work well with mineral oil and other non-synthetic oils, such as akylbenzene, for example, even when the high pressure HFC refrigerants are used.

[0040] The following examples are not intended to limit the scope of the invention, but are intended to illustrate the various aspects of the invention

EXAMPLE 1

[0041] In September 1999, performance tests were performed by Intertek Testing Services (Cortland, N.Y.) on a Model HTS 120A-1B heat pump pool heater charged with R-22 refrigerant. The pool heater included a thermostatic expansion valve as the refrigerant metering device. The system included a ZR 67 Copeland brand compressor. Table 4 lists the conditions and results of the test.

TABLE 4

	Standard Rating Test	Low Temperature Test	Max. Operating Conditions Test
Air Side			
Ambient Temperatures, ° F.			
Dry Bulb	80.60	50.20	80.80
Wet Bulb	70.95	44.35	71.00
Pool Side			
Water Temperatures, ° F.			
Entering	80.15	79.95	104.90
Leaving	84.15	82.85	108.70
Water Flow, gpm	40.05	40.05	40.00
Electrical Characteristics			
Voltage, volts	230	230	230
Current, amps	24.9	22.8	31.4
Power Input, watts	5,415	4,930	6,850
Refrigerant Circuit			
Temperatures, ° F.			
Discharge at compressor	171.5	162.5	198.5
Liquid at TXV	102.5	95.0	124.5
Vapor at Evaporator	69.5	34.5	71.0
Suction at Compressor	69.5	36.0	71.0
Refrigerant Circuit			
Pressures, PSIG			
Discharge at compressor	238	215	311
Suction at compressor	82	53	88
Barometric Pressure (in. Hg.)	28.71	28.76	28.77
Test Conditions			
Heating Capacity (BTU/hr)	80,350	58,500	
Coefficient of Performance (COP)	4.34	3.47	

EXAMPLE 2

[0042] In August 2000, performance tests were performed by Intertek Testing Services (Cortland, N.Y.) on a Model HTS 120A-1D heat pump pool heater charged with R-410A refrigerant. The pool heater included an orifice coupler disposed between the evaporator and condenser to maintain the pressure differential between the condenser side and evaporator side of the heater. The orifice size was 0.136 inch (29 drill size). The system included a ZR 67 Copeland brand compressor. Table 5 lists the conditions and results of the test.

TABLE 5

	Standard Rating Test	Low Temperature Test	Spa Conditions Test
Air Side			
Ambient Temperatures, ° F.			
Dry Bulb	80.60	50.10	80.50
Wet Bulb	71.00	44.20	71.15
Pool Side			
Water Temperatures, ° F.			
Entering	80.25	80.00	103.75
Leaving	86.25	83.60	109.15
Water Flow, gpm	44.90	45.05	45.05
Electrical Characteristics			
Voltage, volts	230	230	230
Current, amps	39.1	32.4	46.7
Power Input, watts	8,590	7,120	10,200
Refrigerant Circuit			
Temperatures, ° F.			
Discharge at compressor	144.5	122.5	163.0
Liquid at TXV	88.5	91.5	110.0
Vapor at Evaporator	51.0	27.5	55.0
Suction at Compressor	50.0	27.0	54.0
Refrigerant Circuit			
Pressures, PSIG			
Discharge at compressor	383	324	474
Suction at compressor	131	84	151
Barometric Pressure (in. Hg.)	28.86	28.86	28.86
Test Conditions			
Heating Capacity (BTU/hr)	135,060	81,570	122,010
Coefficient of Performance (COP)	4.60	3.35	3.50

EXAMPLE 3

[0043] In October 2000, performance tests were performed by Intertek Testing Services (Cortland, N.Y.) on a Model HT 115A-1B heat pump pool heater charged with R-22 refrigerant. The pool heater included an orifice coupler disposed between the evaporator and condenser to maintain the pressure differential between the condenser side and evaporator side of the heater. The orifice size was 0.128 inch (30 drill size). The system included a ZR 67 Copeland brand compressor. Table 6 lists the conditions and results of the test.

TABLE 6

	Standard Rating Test	Low Temperature Test	Spa Conditions Test
Air Side			
Ambient Temperatures, ° F.			
Dry Bulb	80.60	50.05	80.50
Wet Bulb	71.05	44.30	71.15
Pool Side			
Water Temperatures, ° F.			
Entering	79.90	80.00	103.95
Leaving	84.60	82.85	108.25
Water Flow, gpm	45.00	45.00	44.95

TABLE 6-continued

	Standard Rating Test	Low Temperature Test	Spa Conditions Test
Electrical Characteristics			
Voltage, volts	230	230	230
Current, amps	28.7	23.8	33.8
Power Input, watts	6,250	5,130	7,430
Refrigerant Circuit Temperatures, ° F.			
Discharge at compressor	136.5	105.0	157.0
Liquid at TXV	81.0	84.5	100.5
Vapor at Evaporator	NA	NA	NA
Suction at Compressor	55.0	31.5	57.0
Refrigerant Circuit Pressures, PSIG			
Discharge at compressor	266	216	352
Suction at compressor	92	59	96
Barometric Pressure (in. Hg.)	28.98	28.98	28.98
Test Conditions			
Heating Capacity (BTU/hr)	106,310	63,520	96,950
Coefficient of Performance (COP)	4.98	3.62	3.82

EXAMPLE 4

[0044] An HTS 120A-1D heat pump pool heater was charged with CO_2 refrigerant. The system included a ZR 67 Copeland brand compressor and an orifice coupler was disposed in-line with the conduits connecting the evaporator and the condenser. No other metering devices were employed. The heat pump was used to heat water satisfactorily.

EXAMPLE 5

[0045] A chiller was charged with R-22. The system included a nominal 67,000 BTU compressor and an orifice coupler disposed in-line with the conduits connecting the evaporator and the condenser. No other metering devices were employed. The system was used to cool water very satisfactorily (92,000 BTU's was achieved).

I claim:

1. A vapor compression heat exchange system comprising:

a. a compressor, an evaporator, and a condenser;

b. a charge of refrigerant;

c. a series of conduits in communication with said compressor, said condenser, and said evaporator, wherein said conduits are adapted for carrying said refrigerant through said compressor, said condenser, and said evaporator of said heat exchange system;

d. said series of conduits including at least one conduit connecting said condenser and evaporator and through which said refrigerant is carried from said condenser to said evaporator, said at least one conduit having an internal diameter; and

e. a fixedly open orifice disposed in line with said at least one conduit connecting said condenser and evaporator, thereby defining an evaporator side and a condenser side of said system, said orifice having an internal diameter smaller than the diameter of said at least one conduit for creating a pressure differential between said condenser side and said evaporator side of said system during operation.

2. The heat exchange system of claim 1, further comprising a coupler secured to said at least one conduit, said coupler having an inner channel in communication with said at least one conduit, and wherein said open orifice is disposed within said inner channel of said coupler.

3. The heat exchange system of claim 2, wherein said internal diameter of said orifice is from about 0.120 inch to about 0.25 inch.

4. The heat exchange system of claim 1, wherein said orifice is disposed within said at least one conduit.

5. The heat exchange system of claim 4, wherein said internal diameter of said orifice is from about 0.120 inch to about 0.25 inch.

6. The heat exchange system of claim 1, wherein said system is designed to heat air or liquid in a confined space.

7. The heat exchange system of claim 1, wherein said system is designed to cool air or liquid in a confined space.

8. The heat exchange system of claim 1, wherein said refrigerant is selected from the group of hydrofluorocarbons, hydrochlorofluorocarbons, and carbon dioxide.

9. The heat exchange system of claim 8, further comprising a coupler secured to said at least one conduit, said coupler having an inner channel in communication with said at least one conduit, and wherein said open orifice is disposed within said inner channel of said coupler.

10. The heat exchange system of claim 9, wherein said internal diameter of said orifice is from about 0.120 to about 0.25 inch.

11. The heat exchange system of claim 8, wherein said orifice is disposed within said at least one conduit.

12. The heat exchange system of claim 11, wherein said internal diameter of said orifice is from about 0.120 to about 0.25 inch.

13. The heat exchange system of claim 9, wherein said system is designed to heat air or liquid in a confined space.

14. The heat exchange system of claim 9, wherein said system is designed to cool air or liquid in a confined space.

15. The heat exchange system of claim 11, wherein said system is designed to heat air or liquid in a confined space.

16. The heat exchange system of claim 11, wherein said system is designed to cool air or liquid in a confined space

17. The heat exchange system of claim 8, wherein said refrigerant is a hydrofluorocarbon.

18. The heat exchange system of claim 17, further comprising a coupler secured to said at least one conduit, said coupler having an inner channel in communication with said at least one conduit, and wherein said open orifice is disposed within said inner channel of said coupler.

19. The heat exchange system of claim 17, wherein said orifice is disposed within said at least one conduit.

20. The heat exchange system of claim 17, wherein said hydrofluorocarbon is R-410A.

21. The heat exchange system of claim 8, further including a non-synthetic lubricating oil within said compressor.

22. The heat exchange system of claim 22, wherein said refrigerant is a hydrofluorocarbon.

23. The heat exchange system of claim 18, wherein said lubricating oil is mineral oil.

20. The heat exchange system of claim 18, wherein said hydrofluorocarbon is R-410A.

21. The heat exchange system of claim 20, wherein said lubricating oil is mineral oil.

22. The heat exchange system of claim 1, wherein said refrigerant is R-22.

23. The heat exchange system of claim 22, further comprising a coupler secured to said at least one conduit, said coupler having an inner channel in communication with said at least one conduit, and wherein said open orifice is disposed within said inner channel of said coupler.

24. The heat exchange system of claim 23, wherein said internal diameter of said orifice is from about 0.120 inch to about 0.25 inch.

25. The heat exchange system of claim 22, wherein said orifice is disposed within said at least one conduit.

26. The heat exchange system of claim 25, wherein said internal diameter of said orifice is from about 0.120 inch to about 0.25 inch.

27. A heat pump suitable for heating swimming pools and spas comprising:

a. a compressor, an evaporator, and a condenser;

b. a charge of refrigerant;

c. a series of conduits in communication with said compressor, said condenser, and said evaporator, wherein said conduits are adapted for carrying said refrigerant through said compressor, said condenser, and said evaporator of said heat exchange system;

d. said series of conduits including at least one conduit connecting said condenser and evaporator and through which said refrigerant is carried from said condenser to said evaporator, said at least one conduit having an internal diameter; and

e. a fixedly open orifice disposed in line with said at least one conduit connecting said condenser and evaporator, thereby defining an evaporator side and a condenser side of said system, said orifice having an internal diameter smaller than the diameter of said at least one conduit for creating a pressure differential between said condenser side and said evaporator side of said system during operation.

28. The heat pump of claim 27, further comprising a coupler secured to said at least one conduit, said coupler having an inner channel in communication with said at least one conduit, and wherein said open orifice is disposed within said inner channel of said coupler.

29. The heat pump of claim 28, wherein said internal diameter of said orifice is from about 0.120 inch to about 0.25 inch.

30. The heat pump of claim 27, wherein said orifice is disposed within said at least one conduit.

31. The heat pump of claim 30, wherein said internal diameter of said orifice is from about 0.120 inch to about 0.25 inch.

32. The heat pump of claim 27, wherein said refrigerant is selected from the group of hydrofluorocarbons, hydrochlorofluorocarbons, and carbon dioxide.

33. The heat pump of claim 32, further comprising a coupler secured to said at least one conduit, said coupler having an inner channel in communication with said at least one conduit, and wherein said open orifice is disposed within said inner channel of said coupler.

34. The heat pump of claim 33, wherein said internal diameter of said orifice is from about 0.120 to about 0.25 inch.

35. The heat pump of claim 33, wherein said orifice is disposed within said at least one conduit.

36. The heat pump of claim 35, wherein said internal diameter of said orifice is from about 0.120 to about 0.25 inch.

37. The heat pump of claim 32, wherein said refrigerant is R-410A.

38. The heat pump of claim 32, further including a non-synthetic lubricating oil within said compressor.

39. The heat pump of claim 37, further including a mineral oil within said compressor.

40. The heat pump of claim 1, wherein said refrigerant is R-22.

41. The heat pump of claim 40, further comprising a coupler secured to said at least one conduit, said coupler having an inner channel in communication with said at least one conduit, and wherein said open orifice is disposed within said inner channel of said coupler.

42. The heat pump of claim 41, wherein said internal diameter of said orifice is from about 0.120 inch to about 0.25 inch.

43. The heat pump of claim 40, wherein said orifice is disposed within said at least one conduit.

44. The pump system of claim 43, wherein said internal diameter of said orifice is from about 0.120 inch to about 0.25 inch.

45. A method of modifying a vapor compression heat exchange system to increase mass flow of refrigerant during subsequent operation of said system, said method comprising:

a. removing existing refrigerant metering devices from said heat exchange system, said system comprising:

  i. a compressor, an evaporator, and a condenser;

  ii. a series of conduits in communication with said compressor, said condenser, and said evaporator, wherein said conduits are adapted for carrying said refrigerant through said compressor, said condenser, and said evaporator of said system; and

  iii. said series of conduits including at least one conduit connecting said condenser and evaporator and through which said refrigerant is carried from said condenser to said evaporator, said at least one conduit having an internal diameter; and

b. introducing a fixedly open orifice in line with said at least one conduit connecting said condenser and evaporator, thereby defining an evaporator side and a condenser side of said system, said orifice having an internal diameter smaller than the diameter of said at least one conduit for creating a pressure differential between said condenser side and said evaporator side of said system during operation.

46. The method of claim 45, wherein said introducing of said orifice comprises securing a coupler to said at least one conduit, said coupler having an inner channel in communication with said at least one conduit, and wherein said orifice is disposed within said inner channel of said coupler.

47. The method of claim 46, wherein said internal diameter of said orifice is from about 0.125 inch to about 0.25 inch.

48. The method of claim 45, wherein said orifice is disposed within said at least one conduit.

49. The method of claim 48, wherein said internal diameter of said orifice is from about 0.125 inch to about 0.25 inch.

50. The method of claim 45, wherein said refrigerant is replaced with a hydrofluorocarbon.

51. The method of claim 50, wherein said refrigerant is R-410A.

52. The method of claim 51, wherein said introducing of said orifice comprises securing a coupler to said at least one conduit, said coupler having an inner channel in communication with said at least one conduit, and wherein said orifice is disposed within said inner channel of said coupler.

53. The method of claim 52, wherein said internal diameter of said orifice is from about 0.125 inch to about 0.25 inch.

54. The method of claim 51, wherein said orifice is disposed within said at least one conduit.

55. The method of claim 54, wherein said internal diameter of said orifice is from about 0.125 inch to about 0.25 inch.

56. The method of claim 45, wherein said refrigerant is replaced with a carbon dioxide.

57. The method of claim 50, wherein said system includes a non-synthetic lubricating oil within said compressor.

58. The method of claim 57, wherein said lubricating oil is mineral oil.

59. The method of claim 57, wherein said refrigerant is R-410A.

60. The method of claim 58, wherein said refrigerant is R-410A.

61. The method of claim 56, wherein said introducing of said orifice comprises securing a coupler to said at least one conduit, said coupler having an inner channel in communication with said at least one conduit, and wherein said orifice is disposed within said inner channel of said coupler.

62. The method of claim **61**, wherein said internal diameter of said orifice is from about 0.125 inch to about 0.25 inch.

63. The method of claim 56, wherein said orifice is disposed within said at least one conduit.

64. The method of claim **63**, wherein said internal diameter of said orifice is from about 0.125 inch to about 0.25 inch.

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