REFRIGERATION COMPRESSOR CAPACITY AND LOADING CONTROL MEANS

Fig. 1

Fig. 2
REFRIGERATION COMPRESSOR CAPACITY AND LOADING CONTROL MEANS

Bendt Wegge Larsen and Knud Vagn Valbjorn, both of Nordborg, Denmark, assignors to Danfoss A/S, Nordborg, Denmark, a corporation of Denmark

Continuation of application Ser. No. 352,464, filed Mar. 17, 1964. This application July 1, 1966, Ser. No. 562,969

Claims priority, application Germany, Mar. 19, 1963, D 41,165

7 Claims. (Cl. 230—22)

This is a continuation of a U.S. application Serial No. 352,464, filed March 17, 1964, now abandoned.

This invention relates generally to refrigeration and air-conditioning compressor and more particularly to a new and improved compressor capacity and loading control means.

Hermetic refrigeration and air-conditioning compressors must be constructed for long operating lives of the component parts thereof which, of course, cannot be easily replaced. The construction of such compressors requires faultless operation under extreme working conditions. The constructions of such compressors requires taking into consideration the relationship of capacity, suction temperature, condenser temperature, power output of the driving motor and operating speed thereof. Moreover, the refrigeration or air-conditioning load is generally subject to considerable variation during an operating period.

The construction of the compressor must therefore take into consideration the highest and lowest evaporation temperatures and corresponding suction pressure at which the compressor is expected to work. The volume of the pressure strokes and number of compression strokes must be such that a sufficient quantity of refrigerant is delivered at a sufficiently high suction pressure to allow liquifying it under these various operating conditions. On the other hand, the controlling factors in the construction of the motor driving a refrigerating compressor are governed by the highest evaporation temperature or the suction pressure corresponding to this temperature. The motor must deliver sufficient power to take care of the highest load it encounters. The motor must deliver the same amount of power at all times in units where provision is not made, by the use of motor controls, for variable power delivery by the motor. Thus, in order to maintain the motor output reduced and yet sufficient to drive the compressor at its highest operating suction pressure provision is generally made for loading control means by use of unloaders effective during the start of the compressor. These unloaders do not function as capacity control means.

A feature of a refrigeration compressor according to the invention is the provision of a compressor capacity and loading control means in which a cylinder port is provided in the individual compressor cylinder and a bypass valve for the individual port is biased in a direction for closing the port and is operable to an operative position opening the port in response to pressure developed internally in the cylinder thereby to by-pass refrigerant in a gaseous state from internally of the cylinder to externally thereof, for example into a capsule of a hermetic compressor, during the initial phase of travel of the piston during a pressure or compression stroke. The bypass valve is biased to a closed position by expansible and contractible means constructed to permit the bypass valve to open for bypassing or discharging a fraction of the refrigerant from the cylinder to the exterior thereof only when pressure internally of the cylinder exceeds the biasing force of the expansible and contractible means during a phase of the compression stroke of the piston which is generally closer to the start of the individual compression stroke than to the end of the compression stroke thereby to control loading and capacity of the compressor and to permit driving the compressor with less power input than would otherwise be required in the absence of the bypass valve.

The invention takes advantage of the fact that the weight of refrigerant delivered by the compressor increases with the increasing evaporation temperature on the one hand because of the higher suction pressure and on the other hand because of the higher density of the refrigerant. It is therefore possible to use much smaller driving motor according to the invention than was heretofore possible. The motor thereby cheaper and optimum compressor loading and capacity control are more easily maintained than heretofore.

The periods in which hermetic compressors operate at the maximum value of the evaporation temperature at which they are designed to operate are very rare particularly with respect to low and medium back pressure compressors. Compressors generally operate at their rated lower or medium evaporation temperatures. This provision has to be made for operating at the maximum value of the evaporation temperature and therefore high suction pressures. The capacity of the motor must provide for operating under maximum load conditions for the motor when the evaporation temperatures are highest. Thus during the usual operating conditions of such compressors the motor losses are considerable.

According to the invention the power output requirements of the motor can be decreased to such an extent that the range of power output of the motor provided corresponds substantially with the power requirements of the usual operating conditions and in those cases where high suction pressures obtain the compressor capacity and loading control means according to the invention provides for unloading the cylinder so that the motor power is sufficient for driving the compressor during compression of the gaseous refrigerant and the compressor will still deliver a sufficient volume of refrigerant to have the compressor function properly.

Tests indicate that it is useful to not always bypass the same volume of refrigerant by unloading means if the suction pressure justifies the use of an unloader. On the contrary, the volume of refrigerant being bypassed should bear a certain relation to the suction pressure, that is to say it should be greater or less according to whether the suction pressure increases or decreases. This variable control, for example, can be accomplished in refrigerating apparatus by use of a throttle valve instead of a bypass valve. A throttle valve could, for example, vary the volume of the refrigerant unloaded or supplied with increasing suction pressure. Another possibility is keeping the cross-section area of a bypass valve or port constant, however, controlling the valve in such a way that its open time or period of time in which the valve is open to bypass or discharge refrigerant increases with increasing suction pressure.

This last-mentioned technique is realized by the invention in a simple arrangement by controlling a valve opening and closing a cylinder port, for example in dependence upon the difference between piston displacement pressure and suction pressure. This type of capacity and loading control is based on the fact that the pressure of the cylinder of a reciprocating compressor, for example, during a compression stroke as the suction pressure increases. Thus, by controlling a bypass valve and causing it to open sooner during a compression stroke as the higher suction pressure obtains the desired operating characteristics of the invention can be realized.
a simple form of a bypass valve comprising a fluid-filled valve which closes a cylinder port and maintains it closed so long as a given suction pressure obtains or is not exceeded and when this suction pressure is exceeded the cylinder port is opened and maintained opened during a period of time proportionate to the increase of suction pressure relative to the first-mentioned given suction pressure which is a minimum suction pressure. According to the invention, the bypass valve arrangement operates only during an initial phase of the compression stroke and its effectiveness is curtailed or otherwise rendered ineffective during the remainder of the compression stroke as soon as the piston passes the cylinder port.

The bypass valve according to the invention comprises a fluid-containing element, for example a diaphragm or bellows, that expands and contracts to close and open a cylinder port or opening having a fixed or constant area in the cylinder wall of a refrigerant compressor. The fluid-containing diaphragm or bellows device is preferably filled with a gasifiable fluid which keeps the valve closed so long as the suction pressure is lower than the internal pressure of the actuating bellows or diaphragm. If the suction pressure exceeds a given pressure then the internal pressure of the cylinder during the initial phases of compression strokes of the compressor will exceed the balance pressure at which the valve is closed and the valve is opened. The point or suction pressure at which the valve opens is determined for example by the fluid in the actuating element and the extent to which the actuating element is filled. The actuating element preferably closes the valve at a suction pressure corresponding to an evaporator-temperature within a range from about -8°C to about -20°C. This insures a valve arrangement which opens at a well defined suction pressure and increases the extent of opening to the greatest possible opening as the suction pressure increases.

Other features and advantages of the capacity and loading control means according to the present invention will be understood as described in the following specification and appended claims, in conjunction with the following drawings in which:

FIG. 1 is a diagrammatic longitudinal section of a compressor cylinder provided with a bypass valve according to the invention and illustrates the corresponding compressor indicator diagram; and

FIG. 2 is a diagram for illustrating the relationship of motor power and compressor suction pressure.

While the invention is particularly applicable to hermetically sealed refrigeration compressors and will be described as applied to hermetic reciprocating compressors those skilled in the art will recognize that the invention is equally applicable to reciprocating and other compressors for refrigeration and air conditioning apparatus which are not hermetically sealed. Moreover, as hereinafter employed the term bellows or bellows means comprises fluid-containing diaphragm devices and the like.

According to the drawings, and more particularly to FIG. 1, in which a cylinder 1 of a reciprocating compressor, not shown, is illustrated diagrammatically as being provided with a valve plate 2. The valve plate is illustrated as a solid plate, however, it will be understood that it is provided with the usual intake and discharge valves for controlling intake and discharge of a gaseous refrigerant during the intake stroke and compression stroke of the piston 3 reciprocable in the bore 4 of the cylinder 1. The cylinder wall is provided with a port 5 having a fixed dimension or area and a valve seat 7 circumferentially of the port and over which is disposed a pressure-actuated valve 8 which is biased to a closed position by an expanisible and contractible element or actuating element 9.

The loading or biasing element 9 exerts a pressure or biasing force on the valve 8 in a direction for sealing the valve, as hereinafter explained, on the side of the port 5. The pressure developed internally of the cylinder during the compression stroke of the piston 3 is exerted on the valve 8 in a direction for opening it. When the valve is open it allows refrigerant in a gaseous condition to flow into a suction chamber illustrated diagrammatically as a space designated 6 externally of the cylinder for example a space internally of a capsule, not shown.

The suction pressure must rise from a given pressure to a higher pressure by an increase of pressure P before the valve 8 will lift during compression as later explained. During operation, for example, if the condenser pressure or pressure of the gas when compressed to a value necessary for liquefication is P₂, for example and a suction pressure is assumed to be P₁ as illustrated in an indicator or p-V diagram corresponding to the operating compressor in FIG. 1, and the suction pressure P₁ which has a lowest value for example a suction pressure P₁, the operation of the compressor is such that the pressure-volume diagram or indicator card having the ordinate P for pressure and abscissa designated V designating the volume, is that illustrated in FIG. 1 resulting in a work curve I. The compressor thus functions in the usual manner at lower suction pressures. If the compressor is operating with a suction pressure P₁ the bypass valve 8 is not operated since the pressure developed internally of the cylinder during the initial phase of the compression stroke before the piston 3 moves past the point is insufficient to overcome the resistance of the element 9.

Assuming that the suction pressure increases an increment to a higher suction pressure P₂ then during the compression stroke and more particularly the initial phase of thereof an indicator card work curve III is developed. During this condition of operation the bypass valve is pressure-actuated and partly opens after a given travel of the piston 3 and before it passes the port 5 or its compression stroke travel so that gaseous refrigerant is discharged from the interior of the cylinder to externally thereof. In the event that the suction pressure is high, for example P₃, the valve 8 is operated immediately on the compression stroke and all the refrigerant up to the time the piston closes the port 5 is bypassed so that an indicator card curve II is developed instead of an indicator curve III as illustrated on the indicator card. Thus from study of the indicator card or diagram of a compressor provided with bypass valve according to the invention it can be seen that a considerable amount of work or energy is saved during each compression stroke when the suction pressure exceeds a selected limit. This is achieved in view of the fact that not only the absolute value of the compression pressure increase with the increasing suction pressure but the speed or rate at which the compression pressure increases. During this operation the difference between piston displacement pressure or compression pressure and suction pressure which is sufficient to overcome the element 9 of the bypass valve obtains at a time during which the piston has not yet closed off the valve port so that refrigerant is delivered from internally of the cylinder to the exterior thereof into the suction space. The interval or time during which the bypass valve functions to discharge gas refrigerant will increase as the suction pressure rises since the valve will lift proportionately sooner during a compression stroke.

From a study of the indicator card it is apparent that the limiting values of the suction pressure range at which the valve 8 becomes operative and ineffective are indicated by the indicator card curves I and II and it can be understood that the time at which the valve 8 opens and therefore the refrigerant is bypassed from a zero value at the suction pressure P₁ to a respective value at a given suction pressure, for example, P₂. Thus the bypass valve constructed according to the invention not only achieves open-closed control thereby functioning as an unloader or loading control means in dependence upon the suction closing the valve and changes control of volume and therefore capacity control in dependence of the suction pressure so that the unloading and capacity
control of the compressor are obtained by the invention. The expansible and contractible element 9 comprising the actuating element for actuating the valve to a closed and an opened position comprises a bellows or diaphragm. This element 9 contains a fluid comprising a gaseous medium under pressure therein and is made of a suitable deflectable material to allow expansion and contraction. The fluid in the element 9 may be any suitable fluid but preferably is a refrigerant and particularly a gasifiable refrigerant corresponding to the refrigerant being employed in the compressor to which the invention is applied. In this manner any escape of actuating fluid into the compressor will not contaminate the refrigerant. This is particularly important in hermetic compressors for example.

Those skilled in the art will recognize that by controlling the interval pressure developed in the bellows 9 the point at which the valve 8 opens and closes can be suitably determined. Thus by suitably determining the amount of refrigerant in the chamber 10 the bypass valve 11 can be opened before the pressure is determined and its operation pressures can be preset and determined. That is to say the operation pressures at which the valve is effective will be determined.

The power consumption $N$ of a motor driven reciprocating compressor varies in dependence upon the suction pressure $P_s$, as illustrated in FIG. 2. The power consumption of a compressor without the provision of the invention is illustrated and reaches a maximum value $N_{\text{max}}$. The same type compressor provided with a bypass valve according to the invention has a lower power consumption curve $N_2$ in this range of suction pressures as illustrated by a solid line and the power consumption reaches a maximum value illustrated as $N_{2\text{max}}$. Thus in studying the diagrams of FIG. 2 it is apparent that by provision of the bypass valve according to the invention, the power consumption curve is greatly reduced. It can be seen, therefore, that in order to drive a refrigeration compressor provision of the bypass valve according to the invention permits a large reduction in the size of compressor employed since less power input is required for driving the compressor.

In FIG. 2 the full line curve $N_2$ shows for all suction pressures smaller values than the curve $N_1$. In the range of upper suction pressures, which is between the points $c$ and $d$, a characteristic curve is obtained which would correspond with a compressor with a stroke which would only reach up to the valve port. Between points $b$ and $c$, a transition range exists which is based on partial opening of the valve. Between points $a$ and $b$ the characteristic curve has a locus which is close to the locus of the characteristic curve $N_1$, however, somewhat lower. This is because at the lower suction pressures the motor works close to the no-load idling range. The smaller motor in this range produces smaller losses than the larger motor. Moreover, the suction gas or refrigerant is therefore less overheated, and enters the compressor more readily. This results in the lower suction-pressure range in a better refrigerating output at a reduced motor power.

The suction pressure $P_1$ of FIG. 1 would lie between points $a$ and $b$ in FIG. 2, the suction pressure $P_2$ between points $b$ and $c$ and the suction pressure $P_3$ between points $c$ and $d$. The maximum output required from the motor, $N_2\text{max}$, is considerably less than the output $N_1\text{max}$ which would apply to a motor without a bypass valve. By way of example: a motor compressor, which hitherto required a ½-H.P. motor, would be sufficiently powered if provided with an arrangement according to the invention with a ¼-H.P. motor. In this connection, not only a smaller type of a motor is obtained, but also a smaller power consumption covering the entire operating range, a higher specific output, a reduced gas-exit temperature, a reduced capsule-temperature and less changes of the winding temperature.

Although the invention has been described with respect to a reciprocation compressor, for example a domestic hermetic compressor, it will be understood that it is not limited to reciprocatory compressors and the invention can be provided on other types of compressors, for example rotary and centrifugal compressors and reciprocating compressors which are not hermetic.

While a preferred embodiment of the invention has been shown and described, it will be understood that many modifications and changes can be made within the true spirit and scope of the invention.

What we claim and desire to secure by Letters Patent is:

1. A refrigerant compressor having means for compressing a gaseous refrigerant, compressor capacity and loading control means comprising means defining a by-pass port of constant area for bypassing refrigerant under pressure from interiorly of said compressor to exteriorly thereof, a needle valve for opening and closing said port, said needle valve being essentially unaffected by the inertial forces of the refrigerant passing therethrough, bel lows means for applying a biasing force on said valve in a direction for closing said port and biasing said valve against internal gaseous pressure in said compressor, and said bellows means and port being disposed and jointly effective to allow said refrigerant to be discharged through said port in dependence upon a given range of instantaneous differences in pressure between instantaneous compression pressures in said compressor and instantaneous suction pressure and essentially independently of the rate of change of the compression pressure for maintaining the output pressure of said compressor substantially constant regardless of increases and decreases of suction pressure and for reducing the work required of said compressor necessary to maintain compressor capacity and said output pressure constant.

2. A refrigeration compressor according to claim 1, in which said bellows means comprises a fluid filled bellows for applying a biasing force on said valve effective to allow lifting of said valve and outflow through said port a fraction of the refrigerant in said compressor under pressure as a function of the suction pressure of said compressor and proportionate to the increase of suction pressure over a given suction pressure.

3. A refrigeration compressor having a cylinder, a piston reciprocably driven in said cylinder a refrigerant in a gaseous state, compressor capacity and loading control means comprising a bypass valve arrangement comprising, means on said cylinder defining a port disposed for bypassing refrigerant from internally of said cylinder to exteriorly thereof only during a portion of individual compression strokes of said piston, a needle valve for opening and closing said port, said needle valve being essentially unaffected by the inertial forces of the refrigerant passing therethrough, fluid-filled bellows constantly applying a biasing force on said valve in a direction for closing said port and biasing said valve against internal pressure developed in said cylinder during a given travel of said piston during an initial phase of the compression strokes in which the internal pressure in said cylinder exceeds said biasing force, said biasing being essentially independent of the rate of said travel of said piston.

4. In a refrigeration compressor having means for compressing a gaseous refrigerant, compressor capacity and loading control means comprising means defining a by-pass port of constant area for bypassing refrigerant under pressure from interiorly of said compressor to exteriorly thereof, a needle valve for opening and closing said port, said needle valve being essentially unaffected by the inertial forces of the refrigerant passing therethrough, a fluid containing bellows for applying a biasing force on said valve in a direction for closing said port and biasing said
valve against internal gaseous pressure in said compressor,
and said fluid-containing bellows and port being disposed
and jointly effective to allow said refrigerant to be dis-
charged through said port in dependence upon a given
range of instantaneous difference pressures between in-
stantaneous compression pressures in said compressor and
instantaneous suction pressure and essentially independ-
ently of the rate of change of the compression pressure for
maintaining the output pressure of said compressor sub-
stantially constant regardless of increases and decreases
of suction pressure and for reducing the work input to said
compressor necessary to maintain compressor capacity and
said output pressure substantially constant.

5. In a refrigeration compressor according to claim 4,
in which said fluid-containing bellows contains refrigerant
for controlling expansion and contraction thereof.

6. In a refrigeration compressor according to claim 5,
in which said refrigerant in said bellows comprises a
refrigerant similar to the refrigerant in said compres-

7. In a refrigeration compressor according to claim 4,
in which said bellows comprises a fluid responsive to tem-
perature and pressure effective to close said valve at a suc-
tion pressure corresponding to an evaporator-temperature
within a range from about —5° C. to about —20° C.

References Cited by the Examiner

UNITED STATES PATENTS

774,503 11/1904 Yeakley 230—28
1,607,657 11/1926 Whitehead 230—27
1,965,420 7/1934 Lipman 230—22
2,006,584 7/1935 De Puy 230—22
2,208,428 7/1940 Nicolet 230—30
2,486,617 11/1949 Soberg 103—44
2,562,584 7/1951 Soberg 103—44
3,048,022 8/1962 Clary 230—22
3,269,645 8/1966 Valbjorn 230—22

MARK NEWMAN, Primary Examiner.
W. J. KRAUSS, Assistant Examiner.