

[54] REFRIGERATION PLANTS

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62/510, 418/97

[51] Int. Cl..... F25b 1/10

[58] Field of Search 62/84, 192, 193, 468, 470,
62/510, ; 418/9, 10, 97, 201

[56]

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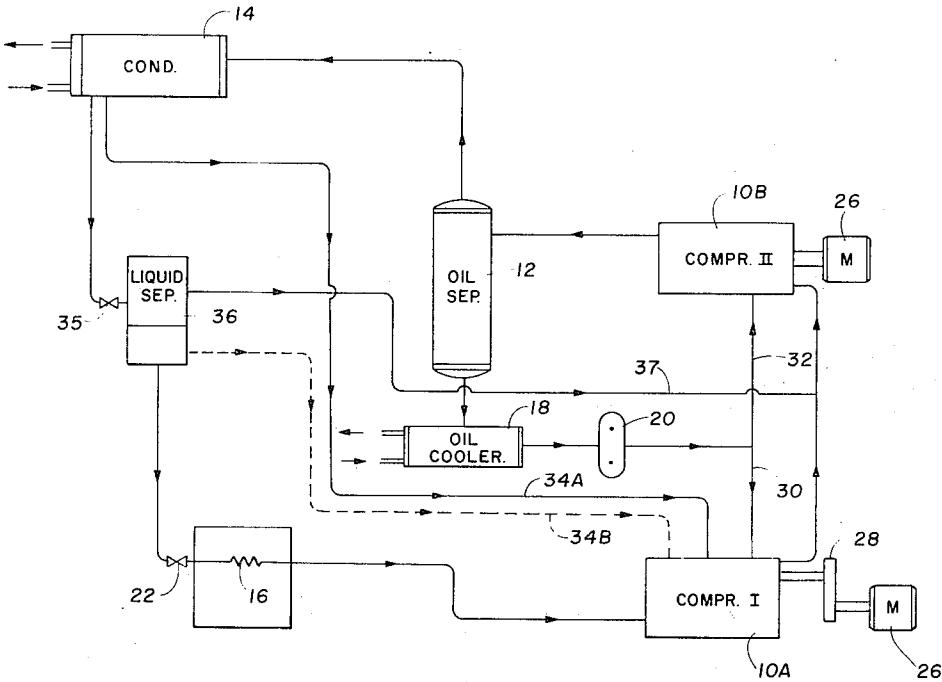
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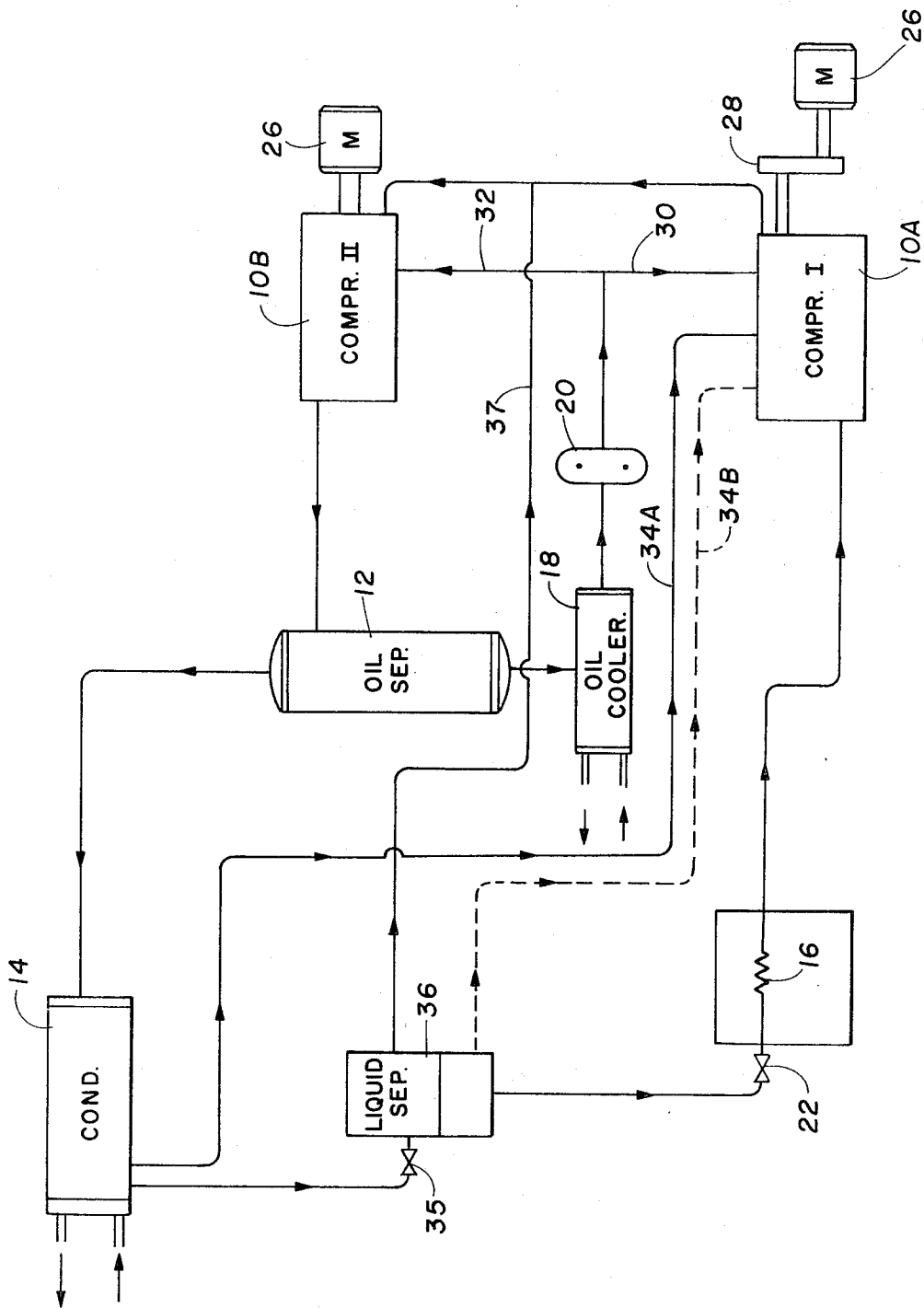
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ABSTRACT

Refrigeration plant comprising two stage compression. The low pressure compressor or booster is of the screw compressor type running at high speed with respect to the high pressure compressor and provided with oil injection, restricted to the amount necessary for lubrication of the contact surfaces of the rotors.

7 Claims, 1 Drawing Figure





REFRIGERATION PLANTS

The present application is a continuation-in-part of my previous application Ser. No. 352,731, filed Apr. 19, 1973, now abandoned.

This invention relates to a compression unit and more particularly to such a unit for use in refrigeration plants of the type including a condenser, an evaporator, and a compression unit including a first compressor means forming a low pressure stage and a second compressor means forming a high pressure stage. The invention also relates to a method of operating a refrigeration plant.

In large refrigeration plants such as for freezing houses the refrigerant may be compressed from say 1.0 atm abs to 18 atm abs. Already at an early stage of the development of such plants it proved necessary to accomplish this compression in two stages in order to obtain an acceptable efficiency. It was also found that it was sufficient to use as the first stage a compressor having a relatively low pressure ratio of for instance 3:1 thereby reducing the volume supplied to the compressor of the second stage to substantially one third of the gas volume leaving the evaporator, the compressor of the first stage thus serving as a booster for the second stage or main compressor. The two compressors may be of the same or different types.

In recent years compressors of the meshing screw rotor type having been used to an ever increasing extent within the refrigeration field i.e. due to the fact that in relation to their size the screw rotor compressors are capable to handle large gas volumes under high pressure ratios. In the heavy-duty design the compressors are provided with means for injecting relatively large amounts of oil into the working chamber for cooling, sealing and lubricating purposes.

The large oil quantities give rise to ventilation losses which, as is well known, increase strongly progressively with the speed. This fact makes it necessary to operate oil-injected (oil-flooded) screw compressors at rather moderate speeds, for instance giving the tips of the male rotor lands a velocity of about 30 m/sec.

Hitherto, when screw compressors have been used in both stages of the compressor unit of a refrigeration plant they have either both been of the oil-injected heavy-duty type or the compressor of the first stage or booster has been of the dry synchronized type. In this connection reference is made to "The development of oil-injected screw compressors for refrigeration" by P. D. Laing and E. J. Perry, page 8, a paper presented before the Institute of Refrigeration at London, Feb. 6, 1964. In both cases the efficiency of the booster has been rather low, for instance of the order of 55%.

The invention has for an object to provide a method of operating a refrigeration plant such as to obtain a high adiabatic efficiency of the booster and thereby of the compressor unit as a whole and to make it possible to use a screw rotor compressor of simple and space-saving design as booster.

A meshing-screw compressor includes two intermeshing rotors disposed on parallel axes within a housing. In such compressors one of the intermeshing rotors is of the male rotor type and the other is of the female rotor type; a male and a female type rotor constituting a set. When the rotors are considered in a plane transverse to the rotor axes, the lands and grooves of the male rotor are located mainly outside the pitch circle of the rotor and formed with substantially convex

flanks while the lands and grooves of the female rotor are located at least mainly inside the pitch circle of the female rotor and formed with substantially concave flanks. The rotors are located within a working chamber formed in a housing having low pressure and high pressure ports and cylindrical and end walls sealingly enclosing the rotors.

According to a first aspect of this invention a compression unit comprises a first and second meshing-screw compressor forming respectively a low and high pressure compression stage means for driving the first and second compressor such that the peripheral velocity of the male rotor of the first compressor is higher than that of the male rotor of the second compressor and means for supplying lubricating oil to the first compressor in an amount at least not considerably exceeding the amount necessary to satisfactorily lubricate the relatively movable contact surfaces.

According to a second aspect of the invention a method of operating a refrigeration plant comprising an evaporator, a condenser and compression unit comprising a first and a second meshing-screw compressor forming respectively a low and a high pressure compression stage comprises the steps of driving the first compressor means with a speed such that the peripheral velocity of its male rotor is higher than that of the male rotor of the second compressor means and supplying lubricating oil to the first compressor means in an amount at least not considerably exceeding what is necessary to satisfactorily lubricate the relatively movable contact surfaces.

According to a third aspect of this invention a refrigeration plant comprises a condenser, an evaporator and a compression unit according to the above mentioned first aspect of this invention.

Since there is only a small oil quantity present within the first compressor the oil does not cause any appreciable ventilation losses. It is therefore possible to have the first compressor running at a much higher speed than an oil-flooded compressor for instance with a male rotor peripheral velocity of the order of 70 m/sec resulting in smaller dimensions. Further, due to the low pressure ratio the load on the radial bearings is also low permitting the use of antifriction bearings of sufficiently small diameters to have room between the rotor shafts but nevertheless of acceptable life.

Preferably, the first compressor is driven at such a speed that the peripheral velocity of the male rotor is at least twice that of the male rotor of the second compressor means.

In order to drive the low pressure compressor or booster at the desired high speed it is in many cases suitable to provide a gearing between the prime mover (such as an electric motor) and the booster. However, the costs of such a gearing are more than compensated for by the fact that the size of the compressor is decreased.

The oil quantity supplied to the working chamber of the booster is so small that it has substantially no cooling effect. Therefore, in most cases it is necessary to cool the booster in some other way, preferably by drawing off liquid refrigerant from the high pressure section of the plant and injecting said liquid refrigerant into the grooves of the rotors of the first compressor means at such a stage of the compression and at a rate such that the temperature of the gaseous refrigerant is prevented from rising above a desired discharge tem-

perature. If desired, the gaseous refrigerant discharged from the booster may be further cooled in an inter-cooler before it enters the second or main compressor.

An embodiment of the invention will now be described by way of example with reference to the accompanying drawing which diagrammatically illustrates a refrigeration plant operating according to the invention.

The plant shown in the drawing includes a compressor unit comprising a first or low-pressure compressor 10A and a second or high-pressure compressor 10B. In conventional manner the plant further comprises an oil separator 12, a condenser 14, a liquid separator 36, an evaporator 16, an oil cooler 18 and an oil pump 20. The compressors, the oil separator, the condenser, the liquid separator and the evaporator are connected in series to form a closed circuit for the refrigerant, a first stage throttle valve 35 and a second stage throttle valve 22 being provided in the conduit between the condenser 14 and the liquid separator 36 and between the liquid separator 36 and the evaporator 16, respectively. Gaseous refrigerant is returned from the liquid separator 36 to the suction side of the high-pressure compressor 10B through a conduit 37.

Both compressors 10A and 10B are of the meshing screw rotor type. Each compressor is driven by an electric motor 26 for instance of the induction type. The motors are in this case assumed to be of the same rated speed. The compressor 10A is driven by its motor via a step-up gear 28 having an output speed for instance two or three times higher than the input speed. The compressor 10B is driven directly by its motor.

In the embodiment shown oil from the oil pump 20 is supplied to both compressors 10A and 10B through conduits 30 and 32, respectively. The oil quantity supplied to the compressor 10A is restricted so that it at least not considerably exceeds what is necessary to satisfactorily lubricate the relatively movable contact surfaces. Simultaneously a certain sealing effect may automatically be obtained. The compressor 10B is of the oil-flooded type in which the oil does not only have a lubricating and sealing effect but also serves to cool the refrigerant during its compression. Oil flooded screw rotor compressors are well known and shown in several U.S. Pat. Specifications, for instance in Specifications U.S. Pat. Nos. 3,129,877 and 3,432,089. In such compressors the mass of the oil flow through the compressor is larger than the mass of the gas flow therethrough. Thus, the oil quantity supplied to the compressor 10B is several times larger than that supplied to the compressor 10A.

In order to cool the gaseous refrigerant compressed by compressor 10A liquid refrigerant is supplied to the compressor from the condenser 14 through a conduit 34A. The liquid refrigerant is injected into the working chamber of the compressor at such a stage of the compression and at a rate such that the temperature of the gaseous refrigerant is prevented from rising above a desired discharge temperature the valve of which is determined by calculations so as to suit the actual thermodynamic cycle. Simultaneously the temperature of the compressor is kept at such a level that the thermal deformations of the rotors and the housing do not cause any problems. As an alternative liquid refrigerant may be supplied to the compressor 10A from the liquid sep-

arator 36 through a conduit 34B indicated in the drawing by a dotted line.

The rotors of the compressor 10A are journaled in antifriction bearings. In the drawing are also shown inlets and outlets for a cooling fluid, for instance water, circulated through the condenser 14 and the oil cooler 18.

One or both compressors may be provided with slide valves for capacity control and the compressor 10B may be cooled also by means of admittance of liquid refrigerant in addition to the cooling by means of injected oil for instance in a similar manner as disclosed in our copending U.S. application Ser. No. 314,993 now U.S. Pat. No. 3,811,291, issued May 21, 1974.

A screw compressor 10A of 25 cm diameter meshing screw rotors is driven so that the tip speed of the male rotor is about 70 m/sec, i.e., at a speed of about 5,300 rpm. The quantity of oil injected is then about 5 liters/minute, if a refrigerant such as ammonia is used at the working conditions -40°C evaporating and -10°C condensing temperatures. This means a mass flow ratio between the injected oil and the sucked in gas of about 0.1 to 1. The compressor 10B may have meshing screw rotors of 23 cm diameter, driven at 2,500 rpm, i.e., a tip speed of male rotor of about 30 m/sec. This compressor is oil flooded and oil is introduced by pump 20 at the rate of about 200 liters/minute meaning a mass flow ratio between the injected oil and the sucked in gas of about 4 to 1.

The relative quantities of oil can be gauged from the foregoing example. These quantities are not critical. It is desirable that an oil film is available on the relatively mating surfaces of compressor 10A.

The injection of liquid refrigerant from the condenser 14 into the working chamber is done at a location leading to an introduction of the liquid refrigerant into the working chamber when the pressure in said working chamber has reached the discharge pressure of the compressor.

Injection of liquid refrigerant from the liquid separator 36 into the working chamber is done at a location permitting the liquid refrigerant to be introduced into the working chamber before the pressure in said working chamber has reached the discharge pressure of the compressor. In this manner the pressure of the liquid refrigerant will be sufficient for injecting the refrigerant into the working chamber against the pressure prevailing in said chamber during this stage of the compression as disclosed in our above-mentioned U.S. Pat. application Ser. No. 314,993.

A suitable injection quantity is about 10 percent of the refrigerant flowing from evaporator 16.

I claim:

1. A method of operating a refrigeration plant including a condenser, an evaporator, a first compressor means forming a low pressure stage and a second compressor means forming a high pressure stage each said compressor means being of the meshing screw rotor type having a first rotor driven by a prime mover and a second rotor driven by direct contact between its lands and the lands of the first rotor, comprising the steps of driving the first compressor means with a speed such that the peripheral velocity of its male rotor is higher than that of the male rotor of the second compressor means and supplying lubricating oil to the first compressor means in an amount at least not considera-

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bly exceeding what is necessary to satisfactorily lubricate the relatively movable contact surfaces.

2. A method as defined in claim 1 and further comprising the steps of drawing off liquid refrigerant from the high pressure section of the plant and injecting said liquid refrigerant into the grooves of the rotors of the first compressor means at such a stage of the compression and at a rate such that the temperature of the gaseous refrigerant is prevented from rising above a desired discharge temperature.

3. A method as defined in claim 1, characterized in that the first compressor means is driven at a speed such that the peripheral velocity of its male rotor is at least about two times that of the male rotor of the second compressor means.

4. A method as defined in claim 2, characterized in that the first compressor means is driven at a speed such that the peripheral velocity of its male rotor is at least about two times that of the male rotor of the second compressor means.

5. A method as defined in claim 2, characterized in

that said liquid refrigerant injected into the grooves of the rotors of the first compressor means is drawn off from said condenser and injected into the working chamber at a location leading to an introduction of the liquid refrigerant into the working chamber when the pressure in said working chamber has reached the discharge pressure of the compressor.

6. Method as defined in claim 1 wherein the step for supplying lubricating oil to the first compressor means (10A) comprises:

supplying lubricating oil in an amount to provide only a film of oil on the relative mating surfaces of the first compressor means.

7. Method as defined in claim 1 wherein the step for supplying lubricating oil to the first compressor means (10A) comprises:

supplying lubricating oil in an amount sufficient to provide a lubricating coating film of oil on the relative mating surfaces of the first compressor means, but not in substantial excess for such a coating film.

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UNITED STATES PATENT OFFICE
CERTIFICATE OF CORRECTION

Patent No. 3,848,422 Dated November 19, 1974

Inventor(s) Lauritz Benedictus SCHIBBYE

It is certified that error appears in the above-identified patent
and that said Letters Patent are hereby corrected as shown below:

Change Foreign Application Priority Data to read:

--Apr. 27, 1972 Great Britain.....19655/72--.

Signed and sealed this 14th day of January 1975.

(SEAL)
Attest:

McCOY M. GIBSON JR.
Attesting Officer

C. MARSHALL DANN
Commissioner of Patents