A TWIN SCREW COMBINED COMPRESSOR AND EXPANDER FOR CO_2 REFRIGERATION SYSTEMS

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ABSTRACT

Recent interest in natural refrigerants has created a new impetus for studies of CO_2 as a working fluid in vapour compression systems for refrigeration and air conditioning. Two major drawbacks to its use are the very high pressure differences required across the compressor and the large efficiency losses associated with the throttling process. To overcome the throttle losses, a number of proposals have been made for various types of positive displacement machine, mainly of the vane type, which combine compression with some recovery of work from the expansion process. However, how well they operate with high pressure differences across the vanes has not been confirmed.

For many years, the authors have been investigating the use of twin screw machines to fulfil both the expansion and compression processes when using more conventional halocarbon refrigerants. These have many potential advantages over other types of positive displacement machine. Unfortunately, when applied to CO_2 the huge bearing forces associated with the pressure distribution within them have hitherto made them appear to be unsuitable.

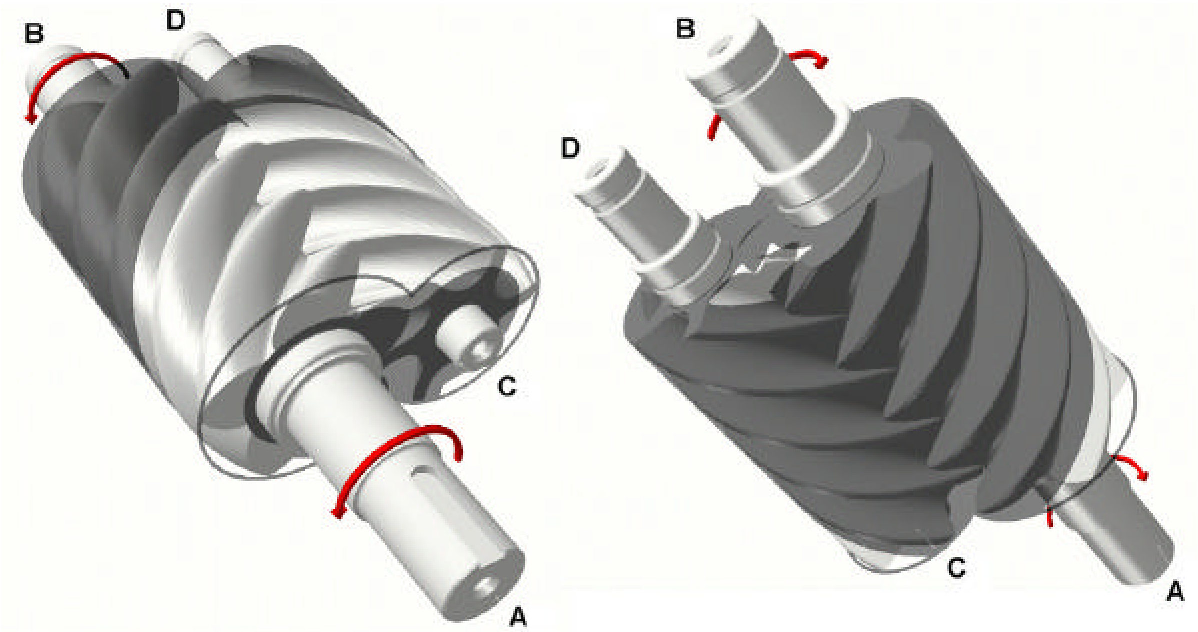
In this paper, it is shown how the rotor forces created by the compression and expansion processes can be partially balanced in order to eliminate the axial forces and reduce the radial bearing forces. The disadvantages of twin screw compressors for such high pressure applications are thereby reduced. The balanced rotor concept is also applicable to vapour compression systems using more conventional refrigerants and even for high pressure gas compression.

INTRODUCTION

Screw compressors are reliable and compact machines and consequently they comprise the majority of all positive displacement compressors now sold and of those currently in operation. One of the main reasons for their success is the advances in manufacturing techniques which enable compressor rotors to be manufactured with very small clearances at an economic cost. Internal leakages have thereby been reduced to a fraction of their values in earlier designs. However, pressure differences across screw compressor rotors impose heavy loads on them and create rotor deformation, which is of the same order of magnitude as the clearances between the rotors and the casing. Consequently the working pressure differences at which twin screw machines can operate reliably and economically are limited. Current practice is for a maximum discharge pressure of 85 bar and a maximum difference between suction and discharge of 35 bar. Rinder, 1999, presented a comprehensive analysis of these effects and Arbon, 2001, gave a good review of current trends in the design, manufacture and use of high pressure screw compressors. CO_2 (R744) in refrigeration cycles requires both maximum pressures and pressure differences beyond these limits. Accordingly, hitherto screw compressors have not been considered for this purpose.

Other types of positive displacement compressors are used today for compression in CO_2 cycles. Typically, these are single and two-stage reciprocating compressors, as described by Pitla et al, 2000. A vane compressor study was presented by Fukuta et al, 2001. In such applications the authors concentrate on either the thermodynamic aspects of the CO_2 cycle or mechanical design aspects of its compressors. In considering twin screw machines capable of operation over the pressure range required in CO_2 refrigeration systems, the following factors must be taken into account

Screw compressors are positive displacement rotary machines which essentially consist of a pair of meshing helical lobed rotors contained in a casing. Together, these form a series of working chambers, as shown in Fig 1, by means of views from opposite ends and sides of the machine. The dark shaded portions show the enclosed region where the rotors are surrounded by the casing and compression takes place, while the light shaded areas show the regions of the rotors which are exposed to external pressure. The large light shaded area in Fig 1a) corresponds to the low pressure port. The small light shaded region between shaft ends B and D in Fig 1b) corresponds to the high pressure port. Admission of the gas to be compressed occurs through the low pressure port which is formed by opening the casing surrounding the top and front face of the rotors. Exposure of the space between the rotor lobes to the suction port, as their front ends pass across it, allows the gas to fill the passages formed between them and the casing. Further rotation then leads to cut off of the port and progressive reduction in the trapped volume in each passage, until the rear ends of the passages between the rotors are exposed to the high pressure discharge port. The gas then flows out through this at approximately constant pressure.



1a) View from Front and Top 1b) View from Bottom and Rear

Figure 1 Screw Compressor Main Components

An important feature of such machines, which can well be appreciated from examination of Fig 1, is that if the direction of rotation of the rotors is reversed, then gas will flow into the machine through the high pressure port and out through the low pressure port and it will act as an expander. The machine will also work as an expander when rotating in the same direction as a compressor provided that the suction and discharge ports are positioned on the opposite sides of the casing to those shown since this is effectively the same as reversing the direction of rotation relative to the ports. When operating as a compressor, mechanical power must be supplied to shaft A to rotate the machine. When acting as an expander, it will rotate automatically and power generated within it will be supplied externally through shaft A.

As already indicated, a major problem with these machines is that the pressure difference between entry and exit creates very large radial and axial forces on the rotors whose magnitude and direction is independent of the direction of rotation. It is normal practice to have bearings on each end of the rotors and these have to withstand both the radial and axial loads induced by the pressure difference. As a result, some of the power transmitted through the rotors is lost in bearing friction. More importantly, the space, in which the bearings must fit, is restricted by the centre line distance between the rotors. Since the required bearing size increases with the load, the maximum pressure difference by which it is possible to compress gases within one pair of rotors is thereby limited. Thus any means of reducing these bearing loads will extend the range of pressures and hence applications, for which such machines may be used.

A further limitation in the use of CO_2 in normal air conditioning and refrigeration systems is that the range of operating pressures and temperatures required are close to the critical point of CO_2. Hence the losses associated with throttling are much larger than those associated with conventional refrigerants. It follows that some recovery of power is required from the expansion process in order to achieve an acceptable coefficient of performance from a CO_2 cycle.

The machine described in this paper is designed to reduce the problem of high bearing loads associated with screw machines, and at the same time enable some power to be recovered from the expansion of the fluid between the cooler and evaporator in a CO_2 vapour compression cycle system.

BALANCING LOADS OF SCREW COMPRESSOR ROTORS

Consider the layout of a CO_2 refrigeration system, operating between an evaporating temperature of 0^oC and a cooler exit temperature of 40^oC as shown in Fig 2.

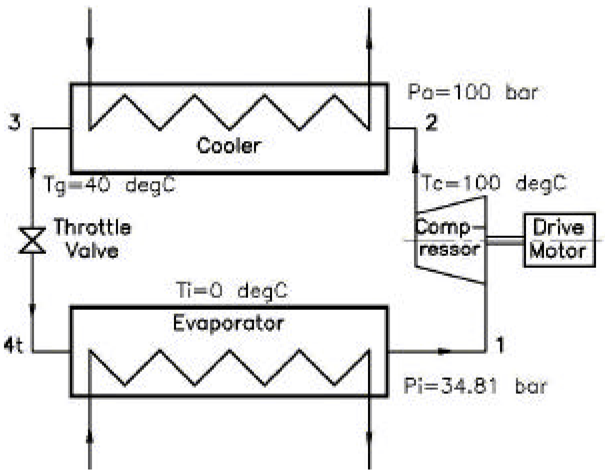


Fig 2a Conventional Refrigeration System

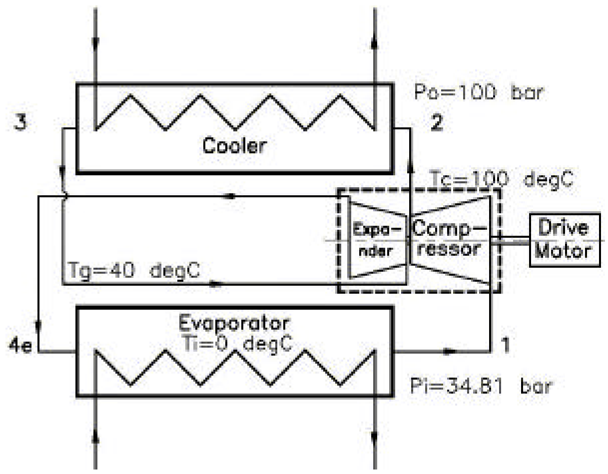


Fig 2b Refrigeration System with Balanced Rotor Compressor-Expander

As shown in Fig 2a, in a conventional system, CO_2 at approximately 35 bar has its pressure raised to 100 bar in a compressor. It then passes through the cooler where it is cooled in the supercritical state at approximately constant pressure until it reaches a temperature of 40^oC. The cooled dense fluid then passes through a throttle valve in which the pressure is reduced back to 35 bar. As a result of the pressure drop, it liquefies and part flashes into vapour, causing the liquid-vapour mixture to fall in temperature to 0^oC. The cooled liquid CO_2, together with the vapour formed during flashing, then passes through the evaporator, where it receives heat from the cold surroundings at approximately 35 bar and 0^oC until all the refrigerant is evaporated. The dry, or slightly superheated vapour then enters the compressor to complete the cycle. As can be seen, the required pressure rise across the compressor is 65.2 bar, which is beyond the limit of what is readily achievable in a single stage twin screw compressor.

Fig 2b shows the arrangement of a twin screw machine installed in a similar system which both compresses and expands the working fluid. In this case, the compressor rotor shafts are extended to include expander rotors on them, so arranged that each set of rotors is contained in a separate chamber within a single casing to form a combined compressor-expander machine. The processes of compression and condensation are identical to those described in the description of the system shown in Fig 2a. However, instead of passing through a throttle valve, the expanding fluid passes through the expander rotors, thus recovering some power from the expansion and thereby reducing the required mechanical power input to the compressor. The method by which the fluid enters and leaves this combined compressor-expander is critical and is shown in more detail in Fig 3. From this, it will be shown how the load on the bearings is reduced.

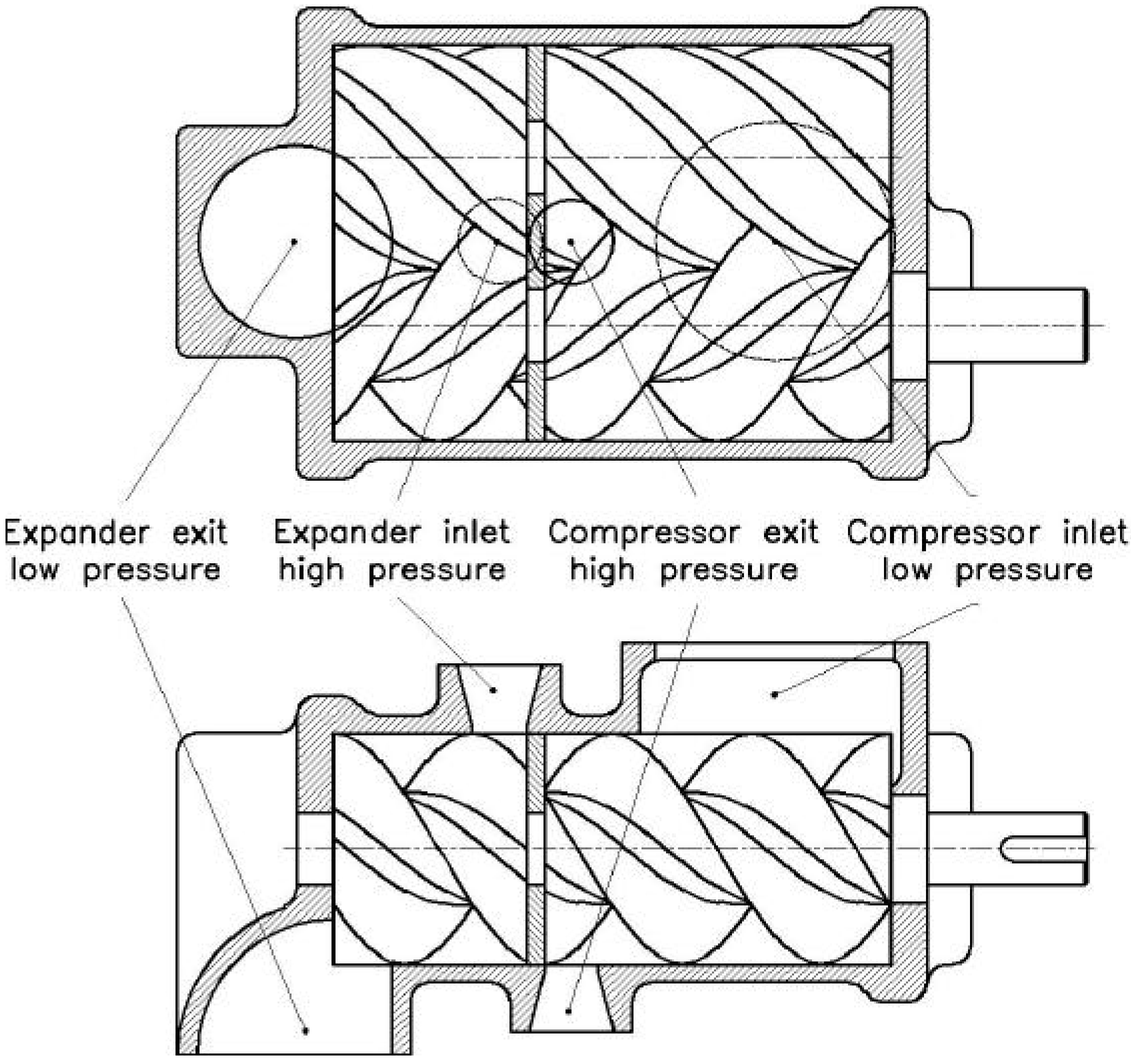


Figure 3 Schematic view of the balanced rotor compressor-expander

As is shown in Fig 3, high pressure dense fluid enters the expander port at the top of the casing, near the centre, and is expelled from the low pressure port at the bottom of the casing at one end, as a mixture of liquid and vapour. The expansion process causes the temperature to drop, as in passing through a throttle valve. However, here the fall in pressure is used to recover power and causes the rotors to turn. Vapour from the evaporator enters the low pressure compressor port, at the top of the opposite end of the casin g, is compressed within it and expelled from the high pressure discharge port at the bottom of the casing, near the centre, to be delivered to the cooler. Ideally, there is no internal transfer of fluid within the machine between the expansion and compression sections which each take place in separate chambers.

The main novelty of this arrangement is in the positioning of the ports. Because the high pressure ports are in the centre of the unit and arranged so that they are on opposite sides of the casing, the high pressure forces due to compression and expansion are opposed to each other and, more significantly, only displaced axially from each other by a relatively short distance. The radial forces on the bearings are thereby significantly reduced. In addition, since both ends of the rotors are at more or less equal pressure, the axial forces virtually balance out

The following example indicates the extent of the advantages, which are possible from this arrangement. Consider a refrigerator in which CO_2 leaves the evaporator at the rate of 2.75m^3/min as dry saturated vapour at a suction pressure of 35 bar to leave the compressor and enter the cooler at a discharge pressure of 100 bar. Exact calculations were carried out on a large simulation program to aid the design of twin screw machines. The results of this showed that for the compressor, the main or male rotor required would be 102mm in diameter with a length/diameter ratio of 1.5. The expander required to replace a throttle valve in this system would have a male rotor of the same diameter but with a length/diameter ratio of only 1.1.

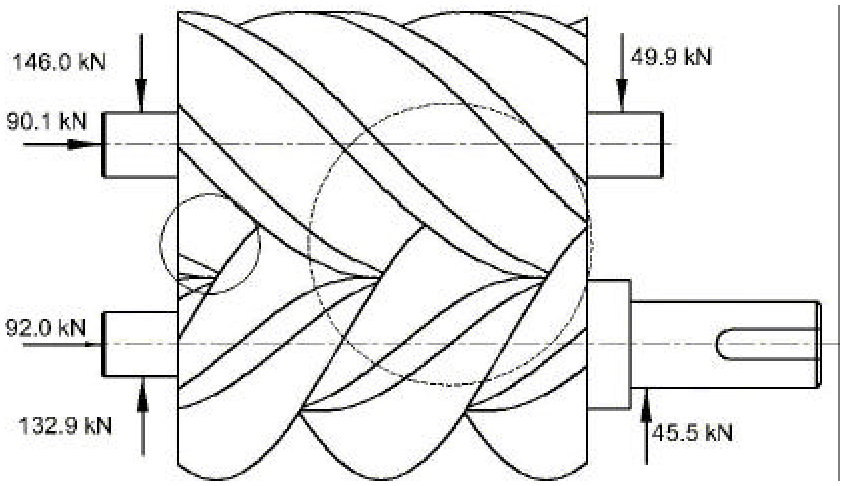


Figure 4 Bearing forces on compressor rotors

Fig 4 shows the compressor rotors and their bearing loads which must be resisted if the refrigeration system is designed with a conventional screw compressor drive. On the main rotor alone, there is an axial force of 92 kN and radial bearing forces of 132.9 kN at the high pressure end and 45,5 kN at the suction end.

In Fig 5 the expander rotors and their corresponding bearing forces are similarly shown. Here, the axial bearing load on the main rotor is 91,9 kN while the corresponding radial loads are 85.9 kN at the high pressure end and 34,1 kN at the low pressure end.

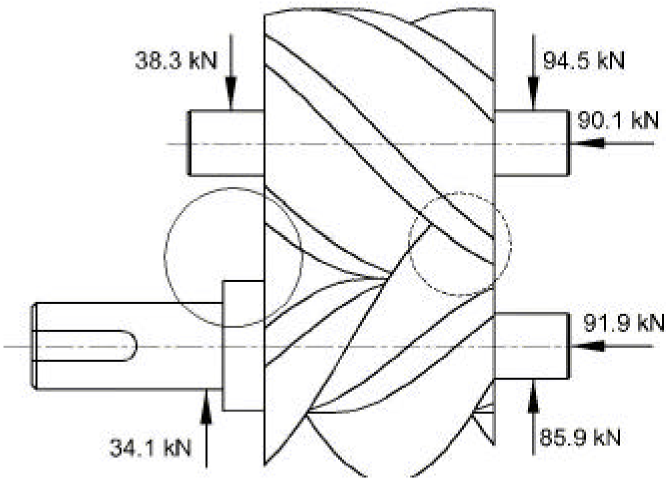


Figure 5 Bearing forces on expander rotors

Fig 6 shows the bearing forces, which would result, if the compressor and expander rotors are machined on the same shafts with the high pressure ports in the middle and the low pressure ports at each end. By this means the main rotor axial load has been reduced to 0.12 kN, which is negligible. The radial bearing loads are now 101 kN at the compressor end and 117 kN at the expander end.

More significantly, for the female rotor, which is weaker, the maximum bearing load has been reduced from 146 kN to 119 kN, which is 19% less.

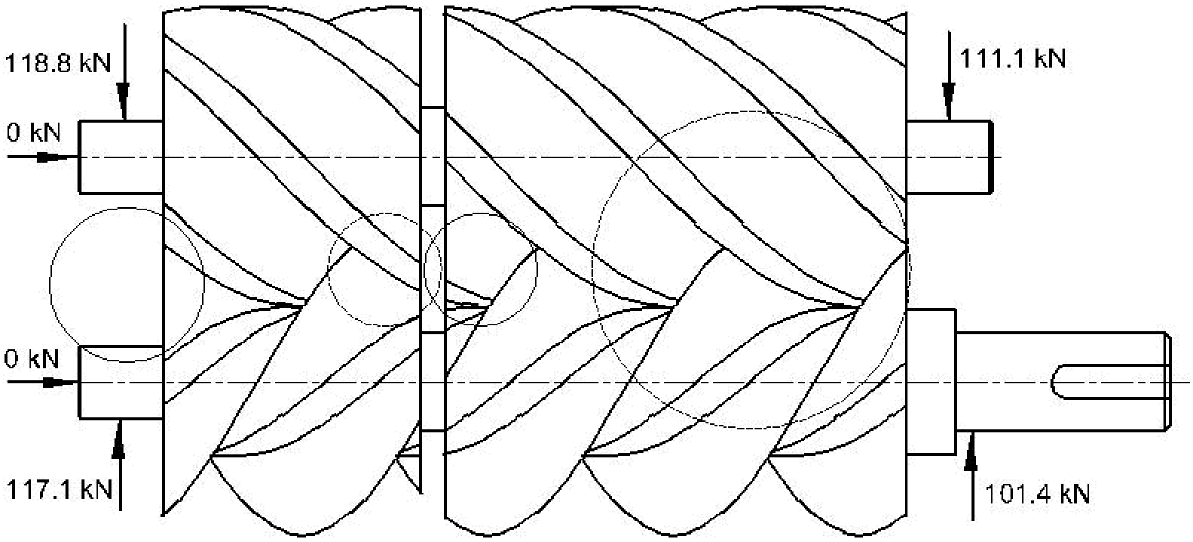


Figure 6 Bearing forces on balanced compressor-expander rotors

Thus the total bearing load on the male rotor alone has been reduced from 270.4 kN for the compressor to 218 kN for the combined compressor-expander. If both male and female rotors are included, then the total bearing load is reduced from 556 kN for the compressor alone to only 448 kN for the combined balanced rotors. This amounts to a total decrease in bearing load of nearly 20%. Design problems associated with high bearing loads in screw compressors for CO_2 systems are thereby reduced.

During recent years a number of other proposals have been made to use twin screw machines both to utilise the expansion process and to supply all or part of the work required for the compression process in a single machine. The first of these by two of the authors (Smith and Stosic), 1995, described an earlier concept of an 'expressor' to serve as a replacement of the throttle valve in classical refrigeration plants. Apart from other differences, it did not include any proposal to balance the rotors forces.

Olofsson, 1993, Shaw, 1999, Ohman, 2000 and Brasz, 2001 all describe inventions to carry out the expansion and compression processes in a single chamber using only one pair of rotors. These all require high pressure admission in a port located at the top of one end of the casing. Some of the fluid entering the expander leaves at an intermediate port, located in the bottom of the casing, while fluid remaining in the rotor pockets is transferred at low pressure from the expansion to the compression section. Compression then occurs along the remaining length of the casing leading to discharge of the compressed fluid through a high pressure port at the bottom of the opposite end of the casing. Almost equal high pressures at each end largely eliminate the axial load but the location of the high pressure ports at the extreme ends of the casing causes the radial bearings to take virtually the full high pressure loads of both the expander and the compressor, even though the ports are on opposite sides. Some differences may be seen in these concepts, depending on whether the expansion section is part of a separate compressor, which runs independently, or is part of the main compressor. However, none of these arrangements would be suitable for operating under the conditions described in this paper.

A further feature of the proposed balanced rotors is the use of rotors, which form a sealing line on both contacting surfaces so that the same profile may be used both for the expander and the compressor sections. In fact, since compression and expansion are carried out separately, the compressor and expander profiles could be different. However, this would make manufacture extremely difficult, due to the very small clearance space, which would be less than 10 mm, between the two rotor pairs. By using the same profile for both, the compressor and expander, the rotors can be machined or ground in a single cutting operation and then separated by machining a parting slot in them on completion of the lobe formation.

Additionally the expansion section can contain a capacity control such as a slide or lifting valve to alter the volume passing through it at part load, in a manner identical to capacity controls normally used in screw compressors. This would be in addition to any capacity or volume ratio control used for the compression section. This would then replace the throttle valve control system normally required in conventional vapour compression systems.

IMPROVEMENT IN THERMODYNAMIC PEFORMANCE OF THE CO_2 CYCLE

Fukuta et al, 2001, gave a comprehensive analysis of the CO_2 cycle. They recognized the problem of throttle loss and proposed improvements by the introduction of a compressor-expander in a vane type of machine for simultaneous compression of CO_2 vapour and expansion of the liquid. An enthalpy-entropy diagram of the idealised cycle with reversible compression and expansion of the CO_2 is shown in Fig 7.

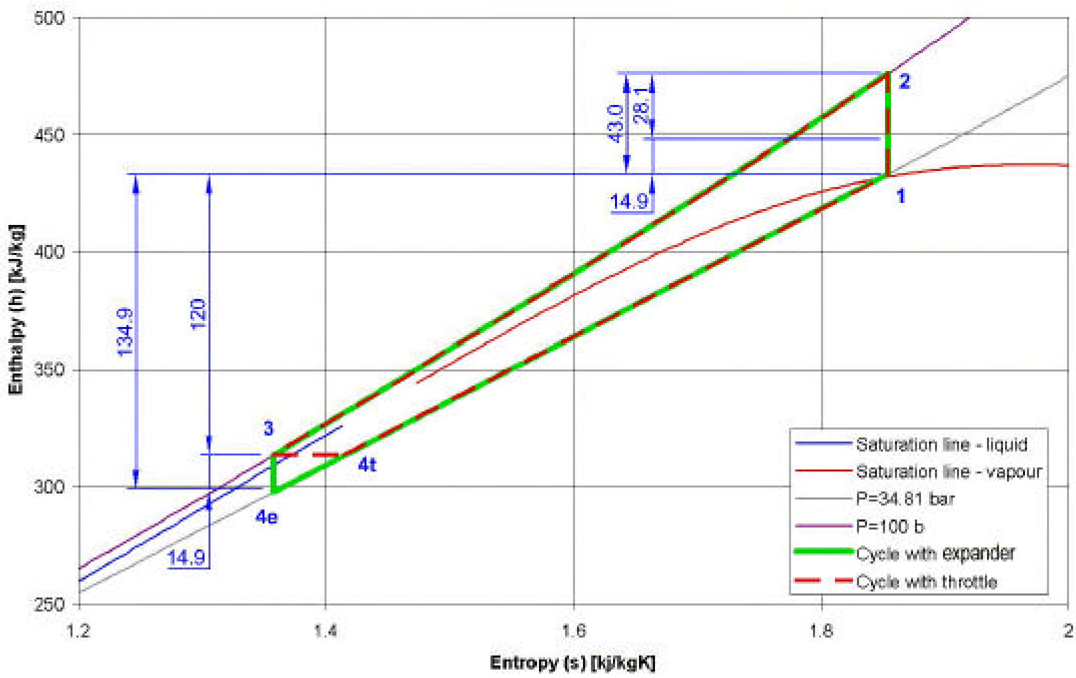


Fig 7: Enthalpy - Entropy Diagram of CO_2 Cycle

As can be seen, point 1 corresponds to vapour being admitted to the compressor, point 2 to discharge from the compressor and entry to the cooler and point 3 to exit from the cooler. If the fluid then passes through a throttle valve, isenthalpic expansion will lead to it entering the evaporator at point 4t. However, if it passed through the expander and work is extracted from it, then the expansion process will be adiabatic and the fluid will enter the evaporator at point 4e. The difference between these two processes is that work extraction reduces the specific enthalpy of the fluid entering the evaporator by 14.9 kJ/kg. This causes the same mass of fluid to enter the evaporator with less vapour and hence has the effect of increasing the refrigerating capacity of the plant by 11%.

At the same time, this recovery of 14.9 kJ/kg in the form of shaft work is used to reduce the external work input to the compressor, shown by the difference between points 1 and 2, from 43.0 kJ/kg to only 28.1 kJ/kg. Thus there is a saving in power input of 34.6%. The coefficient of performance will be improved by both these factors and thus be increased by 72% from 2.79 to a more acceptable 4.8. However, these figures are based on idealised work input and output. In a practical system, allowance would have to be made for the compression and expansion efficiencies, which would reduce the expansion work and increase the compression work. Nonetheless, an overall gain in coefficient of performance over the ideal cycle with a throttle valve should still be achievable by this means.

CONCLUSION

The pressure range over which a single stage screw compressor may be readily made to operate is limited to 65 bar and in special cases up to 85 bar. At high pressures, axial rotor loads have been reasonably successfully reduced by the inclusion of balancing pistons or their equivalent. However, high radial loads have up to now still been unavoidable. These must be reduced for use in high pressure CO_2 applications to avoid excessive rotor deflection and premature bearing wear. The combined compressor expander, as described in this paper fully balances the axial loads and reduces the radial bearing loads. Design problems associated with high bearing loads in screw compressors for CO_2 systems are thereby reduced.

Apart from reducing pressure loads, the simultaneous compression and expansion processes reduce throttling losses in CO_2 vapour compression system plants significantly.

Although, the balanced rotor configuration described in this paper was conceived specifically for high pressure CO_2 refrigeration systems, it may also be used with more conventional refrigerants, or indeed, wherever there is a need for combined expansion and compression processes. For a small penalty in overall efficiency, it may also be used in non refrigeration applications to permit high pressure compression in a single stage system with reduced rotor loads, even where there is no system requirement to recover power from the expansion process.

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