Multifunctional Screw Compressor Rotors

Nikola Stosic, Ian K. Smith and Ahmed Kovacevic

Centre for Positive Displacement Compressor Technology,
City University, London EC1V OHB, U.K.
N.Stosic@city.ac.uk

Abstract: *For many years the authors have been investigating the use of twin screw machines to fulfil both the expansion and compression. One of potential advantages of screw machines over other types of positive displacement machine is their ability to perform both the compression and expansion functions simultaneously, using only one pair of rotors. A further feature is the use of the rotors which seal on both contacting surfaces so that the same profile may be used both for the expander and the compressor sections. By using the same profile for both, the compressor and expander 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. Moreover, in such a case, by proper location of the machine ports, pressure loads and thereby, mechanical friction losses will be less than if the two functions are performed in separate machines. Fields of application of such machines are replacement of the throttle valve in refrigeration and air conditioning plants, high-pressure application, fuel cells, multistage compression or expansion and, really, any other application where simultaneous compression and expansion are required. One example of such unusual, but convenient application is the compressor capacity control by partial expanding of compressed gas.*

Keywords*: Screw Compressor, Screw Expander, Compression and Expansion on one Rotor Pair*

1. INTRODUCTION

Screw compressors are reliable and compact machines and consequently they comprise a substantial portion of all positive displacement compressors now sold and of those currently in operation.. The main reasons for this success are the development of novel rotor profiles, which have drastically reduced internal leakage, and advanced machine tools, which can manufacture the most complex shapes to tolerances of the order of 3 microns at an acceptable cost. Rotor profile enhancement is still the most promising means of further improving screw compressors and rational procedures are now being developed both to replace earlier empirically derived shapes and also to vary the proportions of the selected profile to obtain the best result for the application for which the compressor is required. In addition, improved modelling of flow patterns within the machine can lead to better porting design. Also, more accurate determination of bearing loads and how they fluctuate enable better choices of bearings to be made. Finally, if rotor and casing distortion, as a result of temperature and pressure changes within the compressor, can be estimated reliably, machining procedures can be devised to minimise their adverse effects.

Screw machines operate on a variety of working fluids, which may be gases, dry vapour or multi-phase mixtures with phase changes taking place within the machine. They may involve oil flooding, or other fluids injected during the compression or expansion process, or without any form of internal lubrication. Their geometry may vary depending on the number of lobes in each rotor, the basic rotor profile and the relative proportions of each rotor lobe segment. It follows that there is no universal configuration which would be the best for all applications. Hence, detailed thermodynamic analysis of the compression process and evaluation of the influence of the various design parameters on performance is more important to obtain the best results from these machines than from other types which could be used for the same application. A set of well defined criteria governed by an optimisation procedure is therefore a prerequisite for achieving the best design for each application. Such guidelines are also essential for the further improvement of existing screw machine designs and broadening their range of uses. *Fleming et al, 1998* gives a good review of screw compressor modelling, design and application.

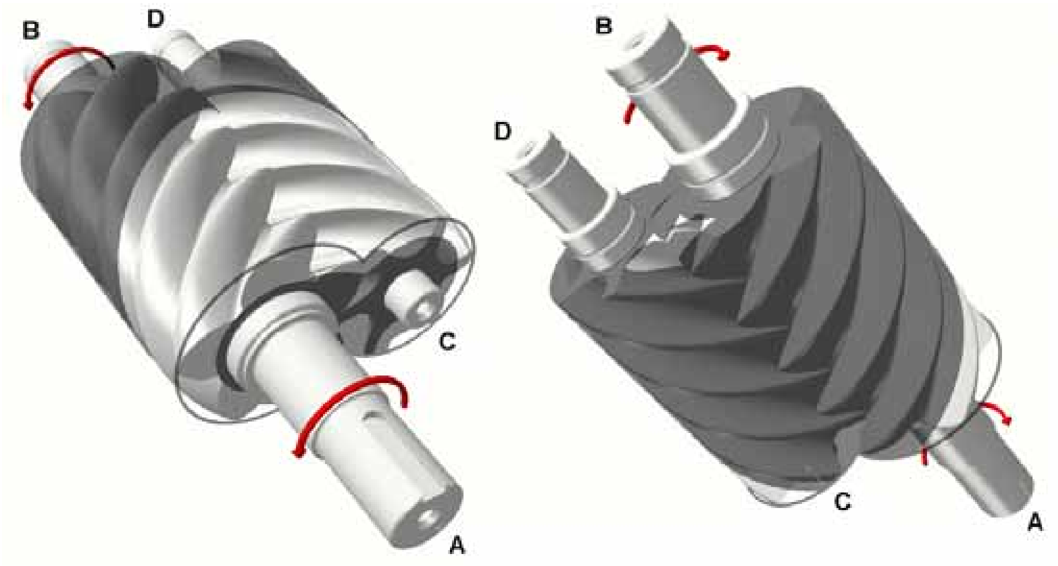
An important feature of screw machines is that if the direction of rotation of the rotors is reversed, then gas will flow into the machine through the high pressure port and out through the low pressure port and it will act as an expander. The machine will also work as an expander when rotating in the 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 further feature is the use of a single pair of the rotors which seal on both contacting surfaces so that the same profile may be used both for the expander and the compressor sections. By using the same profile for both, the compressor and expander 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. More information about new conceptr in screw machines can be found in *Stosic et al 2003*.

2. SIMULTANEOUS COMPRESSION AND EXPANSION ON ONE PAIR OF ROTORS

This paper is concerned with a multifunctional use of the screw machine, the multistage compression or multistage expansion and the simultaneous compression and expansion.

Screw compressors are positive displacement rotary machines which consist, essentially, 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.

Examination of Fig 1, shows 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. Moreover, the machine will work as an expander when rotating in the same direction as a compressor provided if 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.



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

Figure 1 Screw Machine Main Components

In Fig 2 the arrangement of a screw machine 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 layout by which the fluid enters and leaves this combined compressor-expander is critical and from this, load on the machine bearings is reduced.

As is shown in Fig 2, high pressure gas enters the expander port at the top of the casing, near the centre, and are expelled from the low pressure port at the bottom of the casing at one end. The expansion process causes the temperature to drop. However, here the fall in pressure is used to recover power and causes the rotors to turn. Air enters the low pressure compressor port, at the top of the opposite end of the casing, 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 atmosphere. Ideally, there is no internal transfer of fluid within the machine between the expansion and compression sections which each take place in separate chambers.

If the same machine presented in Fig 2 is used as a two stage compressor, only the ports of the second stage will exchange their places. The low pressure port of the second stage will be located on the top of the machine and the high pressure discharge will be at the machine bottom. This offers a compact two stage machine which may be used either in the oil flooded or dry operation mode. A similar analysis is valid for a two stage expander.

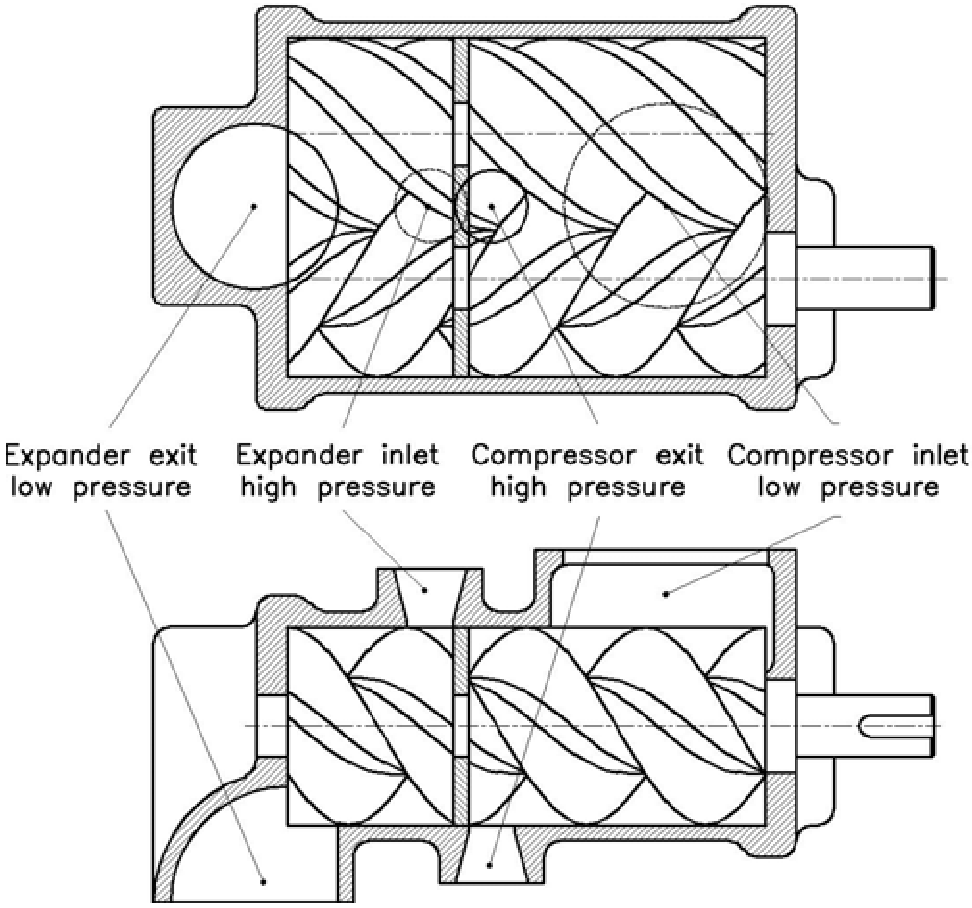


Figure 2 View of the multifunctional rotors acting simultaneously as compressor and expander

3. DESIGN CHARACTERISTICS OF MULTIFUNCTIONAL SCREW ROTORS

If used for the combined compression and expansion, a further feature of the proposed multifunctional rotors must be utilized, the rotors must form a full 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 small clearance space between the two rotor functions. By using the same profile for both, the compressor and expander, the rotors can be milled or ground in a single cutting operation and then separated by machining a parting slot in them on completion of the lobe formation. Such rotors are presented in Fig. 3.

Additionally the expansion section can contain a capacity control such as a slide or lifting valve at suction to alter the volume passing through it at part load, in a manner identical to capacity controls normally used in screw compressors. In the fuel cell application this will control the cell pressure, while the air flow will be controlled by variable machine speed.

As already indicated, a major problem with screw 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. A full balancing of axial forces and partial balancing of radial forces will increase the machine efficiency compared with two units operation independently.

If the rotors are used for multistage compression, they can retain their profile shape common for screw compressors with a small blow hole on one side and a relatively large one on the opposite side.

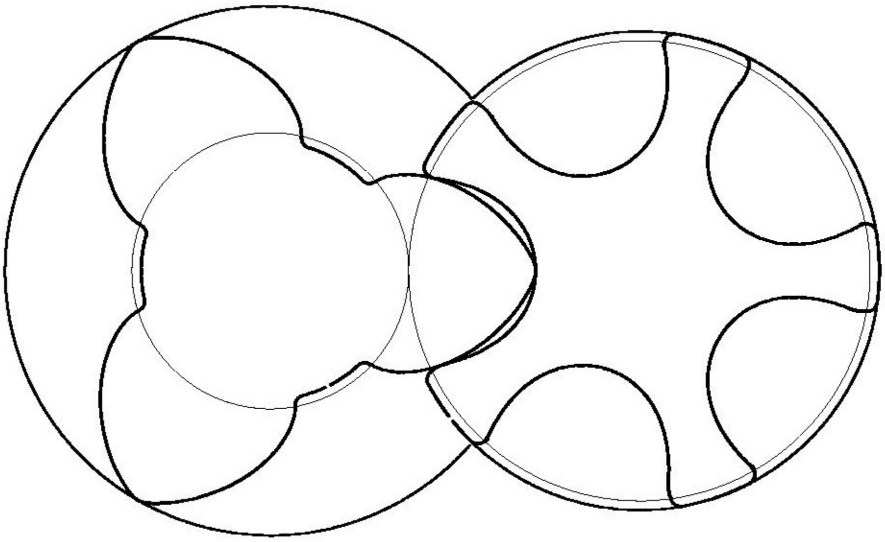


Figure 3 Compressor-Expander Rotors

4. BALANCING FORCES ON COMPREESOR–EXPANDER ROTORS

An important novelty of the compressor expander arrangement on one pair of rotors is in the positioning of the ports. Because the high pressure ports of such machine 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 of a combined compressor and expander in the high pressure application indicates the extent of the advantages, which are possible from this arrangement.

A refrigerator uses 2.75 m^3/min CO_2 as a working fluid which l3aves evaporator and enters the compressor as dry saturated vapour at a suction pressure of 35 bar to leave the compressor and enter the condenser at a discharge pressure of 100 bar. The compressor 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. The force calculations showed that 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. A similar calculation was performed for the expander rotors and their corresponding bearing forces. 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. The bearing forces, which would result, if the compressor and expander rotors were machined on the same shafts with the high pressure ports in the middle and the low pressure ports at each end were as following. The male 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. 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.

5. APPLICATION EXAMPLES OF MULTIFUNCTIONAL SCREW ROTORS

Several examples of application of multifunctional rotors are presented here. The expressor for simultaneous expansion of refrigerant liquid and compression of its vapour, dry fuel cell compressor expander for simultaneous compression of air and expansion of fuel cell reaction products, high pressure oil flooded compressor for CO_2 and a two stage oil flooded air compressor.

5.1 The Expressor

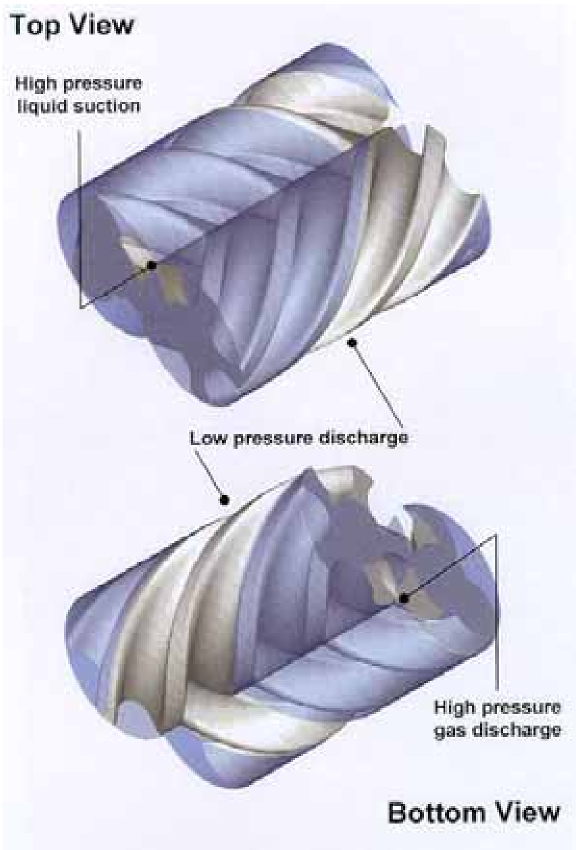


Fig. 4 The expressor principle

*A*n introductory report on a means of replacing the throttle valve in vapour compression systems was published by *Brasz et al, 2000*. Power is recovered from the two-phase expansion process and used directly to recompress a portion of the vapour formed during the expansion. Both the expansion and recompression processes are carried out in a self-driven machine with only one pair of rotors and no external drive shaft. The principle of simultaneous expansion and compression on the same pair of screw rotors is illustrated in Fig. 8. Unlike rotors that perform compression or expansion only, these have to seal on both sides, as shown in Fig 9. The authors have called such a device an "expressor" and, as built and tested, it operated as a process lubricated totally sealed unit without the need for lubricating oil, internal seals or timing gears. A complete unit is shown in Fig 10.

The test results indicate that the overall expansion-compression efficiency of the expressor is of the order of 55%. This corresponds roughly to 70% expansion efficiency and 80% compression efficiency. The simplicity of the expressor design, together with its promising performance, indicate that it should be a highly cost effective component in large commercial chiller systems.

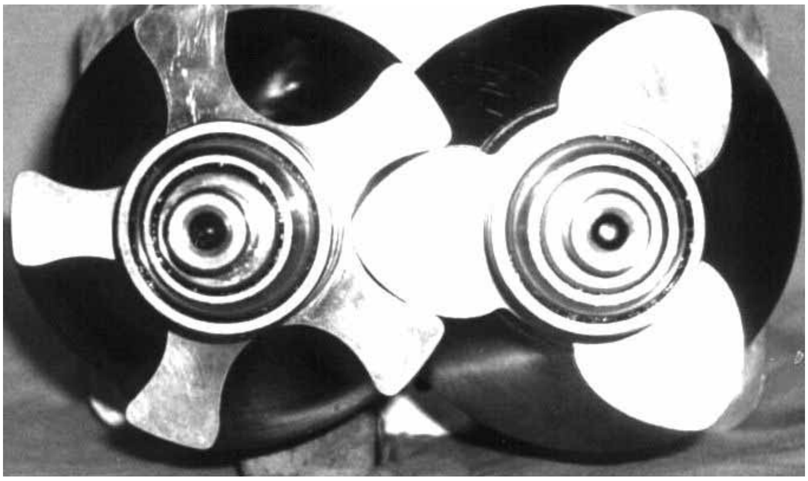


Fig. 5 Expressor rotors sealed on both sides

Further studies are being performed to determine the best built in volume ratios for the expansion and compression processes and to improve liquid-vapour separation during low pressure discharge.

Fig. 6 The expressor prototype before assembling

5.2 Fuel Cell Compressor-Expander

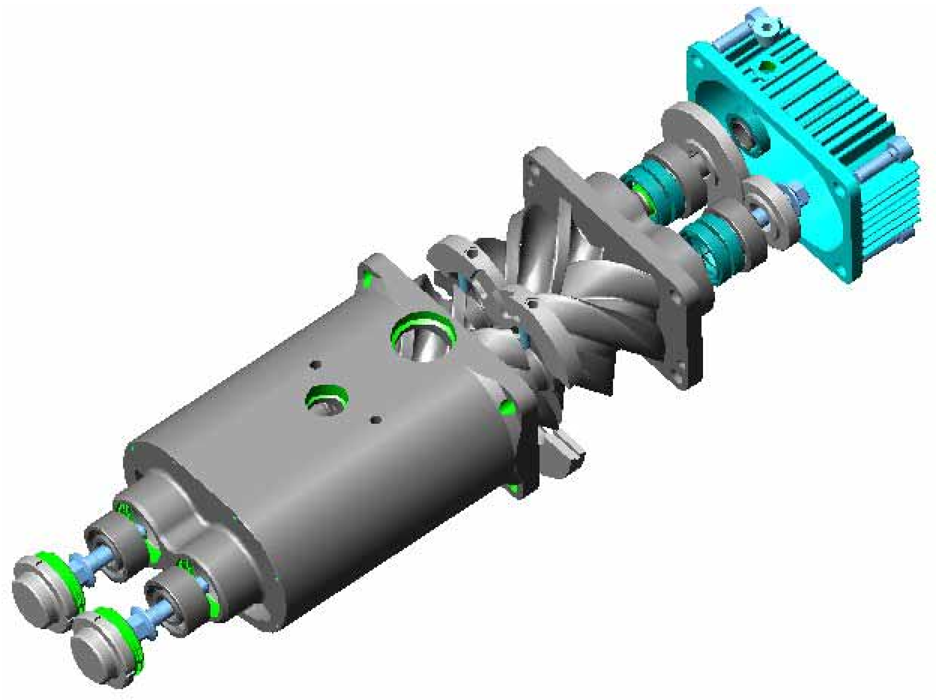


Figure 7 Expanded View of the Compressor-Expander for Fuel Cell Application

An expanded view of the compressor expander for fuel cell application is given in Fig 6. Both, the expansion and compression sections are clearly visible being separated by the central plate which contains high pressure ports. The rotors of the fuel cell compressor expander are presented in Fig. 3.

Figure 8 Expanded View of the Compressor-Expander for Fuel Cell Application

A prototype of a similar machine was manufactured and experimentally investigated as a replacement of a throttle valve in a refrigeration plant. Such a machine is presented in Fig. 8. This was a direct contact machine with rotors lubricated by process fluid, while the fluid cell compressor-expander might be either dry or water injected. In both cases rotor synchronization by oil lubricated gears, as well as oil lubricated bearings are recommended.

5.3 High Pressure Screw Compressor

Recent interest in natural refrigerants, has resulted in more intensive 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 of up to 50 bars required across the compressor and the large efficiency losses associated with the throttling process. To overcome the throttle losses, a combined compression with a recovery of work from the expansion process is proposed. Furthermore, 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. The balanced rotor concept, requires the rotors to seal, as shown in Fig 3, but requires the compression and expansion sections to be separated completely, as shown in Fig 2.

Apart from reducing pressure loads, the simultaneous compression and expansion processes reduce throttling losses in CO_2 vapour compression system plants significantly. 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. Analysis presented in Stosic, 2002 shows that the coefficient of performance will be improved by both these factors, recovering of the throttle loss and reducing of mechanical loses because of lower rotor loads 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. More information on this application can be found in *Stosic et al 2002*.

5. 4 Two-Stage Compressor

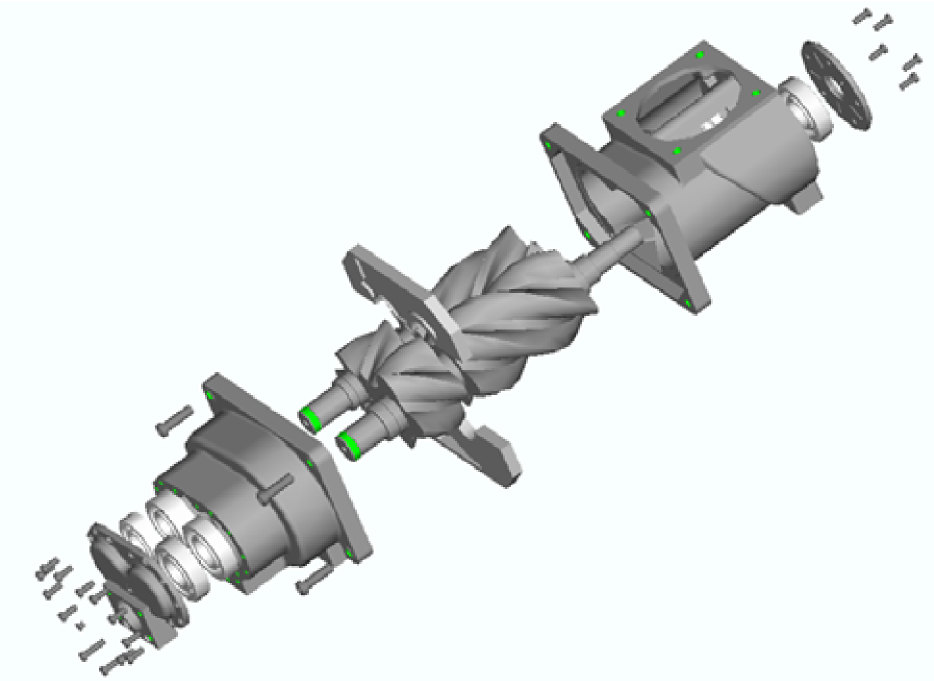


Figure 9 Expanded View of the two-stage compressor on one pair of rotors

5. 5 Capacity Control by Expanding of the Compressed Gas Surplus

An interesting application of multifunctional rotors is being proposed here. This is a means of the compressor capacity control by expanding of a flow surplus in the expander part of the combined compressor expander. It is known that screw compressors are subjected to various capacity controls, including there the suction throttling, variable compressor speed and variable compressor suction volume. Throttling is inefficient, while variable speed devices and sliding valves are expensive, therefore the proposed principle, although burdened by combined efficiencies of the compressor and expander process, may have its chance.

6. CONCLUSION

Although the screw compressor is now a well developed product, greater involvement of engineering science in the form of computer modelling and mathematical analysis at the design stage, makes further innovations possible. The use such machines for expansion as well as compression, leads to possibilities of combining both processes in the same machine and thereby extending their range of application.

Namely, an important feature of screw machines is that it 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, which is effectively the same as reversing the direction of rotation relative to the ports. By using the same profile for both, the compressor and expander 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. As a result a lower rotor loads will be obtained for the same machine duty. This system may equally be used with conventional refrigerants, or indeed, wherever there is need for combined expansion and compression processes or even if a combined expansion compression process is established only to reduce the rotor loads.

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