INTERNATIONAL DESIGN CONFERENCE - DESIGN 2000
Dubrovnik, May 23-26, 2000

DESIGN OF OIL-LESS HELICAL TWIN SCREW MACHINES

Nikola Stošiæ, Ian K Smith and Ahmed Kovaèeviæ

Keywords: *Screw compressor performance, low viscosity liquid injection, involute profile rotors*

Abstract: A twin screw machine is described as oil-less when the working chamber between the contacting helical rotors and the casing, which is usually flooded by oil, operates with no oil or a very small quantity of oil injected in it. Such a mode of operation has been shown to be viable in two experimental projects in which three screw machine prototypes were successfully tested. Namely:

i) A water injected compressor
ii) An oil-less screw expander
iii) A single rotor pair screw expressor.

It is particularly valuable when such machines are required to run with no leakage to the environment or where mixing of the working and auxiliary fluids is not allowed. The design principles applied are described as well as the results obtained from test measurements on these machines. The latter, which confirm the feasibility of the concept, are included in the paper.

1. Introduction

Screw machines operate in two major modes of operation: dry and oil-flooded. In the dry mode, synchronising gears, through which they are driven, avoid contact between the screw rotors. Such machines can then compress or expand dry gas without the need for entrained oil. However, the synchronising gears must be lubricated. Hence, seals must be inserted on the rotor shafts between them and the working chamber to prevent inward leakage of the oil. These are of complex design and reduce machine efficiency. Therefore, oil-free screw machines are expensive both to manufacture and to operate.

In the oil-flooded mode, the rotors are in continuous contact and are lubricated by a large amount of oil injected directly into the working chamber near the start of the compression process. This must be sufficient for both the rotors and their bearings to withstand the torque transfer and end loadings caused by the severe pressure loads under which these machines operate. In this mode of operation, manufacture is simple and cheap. However, the presence of oil in large quantities causes high rotor drag forces. This reduces the rotational speed possible with high efficiency. To minimise the adverse effects of oil injection, continuous efforts have been made to investigate its influence on screw machine processes, as reported by [Stošiæ at al 1992], while many attempts have been made to maintain safe rotor contact with only a small quantity of injected oil or with oil of reduced viscosity. Such oil-less operation of screw machines has often resulted in their failure.
Failures of screw compressors when operating with a small quantity of oil or using a low viscosity

injection liquid, such as water as the cooling and sealing medium, are usually due to either unsuitable rotor profiles or surface material properties [Kauder and Daemgen 1994]. A considerable effort has therefore been made to develop screw machine rotor profiles which permit sustained oil-free or oil-less operation.

This paper describes a programme to achieve this by means of rotor profiles formed from a rack generation procedure [Stoš iæ 1998].

In the later stages of the project, rotors of 3/5 configuration with a main rotor diameter of 128 mm were used for a screw expressor design. The expressor is a machine in which two-phase expansion and vapour recompression of a refrigerant occur simultaneously with no external drive in a manner analogous to that of a turbocharger. Similarly, the expander and expressor operated successfully with only a small quantity of oil in the refrigerant which was R-113. These results confirmed that continuous operation of such machines is possible.

2. Design Principles of Oil-Less Helical Screw Machines

A programme for the development of screw compressors and expanders is being carried out at City University London.

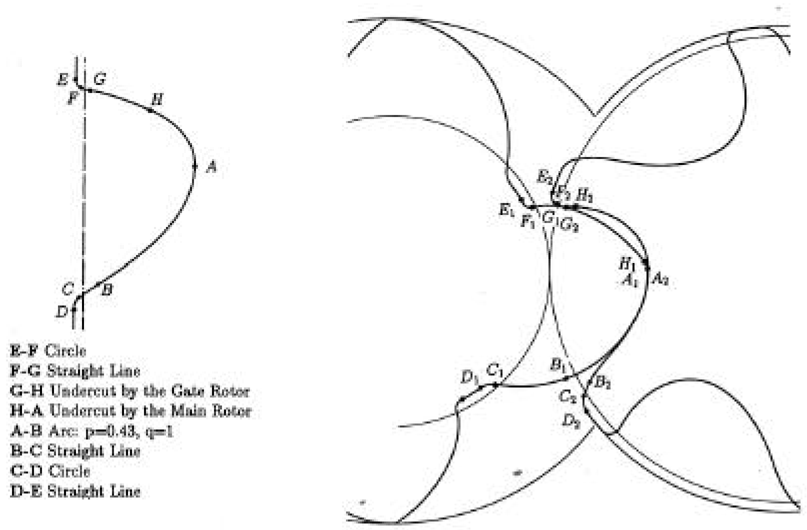

Fig. 1 'N' Helical Screw Rotor Profile

This includes the design of rotor profiles most likely to permit oil-less or oil free operation. The key to this was to produce both male and female lobes by a rack generation procedure developed by one of the authors [Stoš iæ and Hanjaliæ 1997]. In this case both have the form of a straight line in the region of the contact band. This gives them an involute form in that area which implies that the contact between the two rotors is the best possible. A further feature of this rotor profile is that there is very little torque transmission through the female rotor and hence the contact forces between the rotors and stresses on the rotors are small. The rotors also have other advantages including a very large cross sectional flow area, strong gate rotor lobes and a clearance distribution such that in the event of hard contact between the rotors, they will not seize. Patent applications have been filed for the family of profiles thus generated [Stoš iæ 1997].

Their suitability for oil-less operation was first checked by using such rotors with a 5/6 configuration in an oil injected compressor with a male rotor diameter of 128 mm. Water was used as the injected fluid in place of oil and the compressor was run for 150 hours with coated rotors and 5 hours with uncoated rotors. This has already been described in detail [Stoš iæ et al 1997]. No rotor wear was detected in either case.

At this time a twin screw machine was being designed to expand liquid refrigerant in two-phase flow. Previously timing gear had been used to avoid rotor contact in such machines in view of the low viscosity of the refrigerant and the need to minimise its oil content to enable the flash expansion of the liquid to proceed efficiently. The pressure difference required in this machine was less than half that obtained in the water injected compressor tests. It was therefore decided to use such rotors to dispense with the timing gear in the expander and rely on the presence of liquid refrigerant to act as a coolant should any heat be generated by rotor contact.

Advances in rolling element bearing design [Jakobson 1996] led to the decision to use liquid refrigerant as the bearing coolant and thereby completely eliminate oil from the expander system. Internal shaft seals then would not be required to prevent oil from the gears or bearings leaking into the refrigerant. Only one seal was then required and this was of the external type used in refrigeration compressors to prevent leakage through the shaft drive.

A 5/6 rotor configuration was chosen for this design with a male rotor diameter of 128 mm and an L/D ratio of 1.65 which was similar in character to that used in the air compressor. This has already been reported [Smith et al 1999]. The result of these decisions was to produce a very simple low cost machine. In performance tests it ran for over 200 hours with no detectable wear in the rotors.

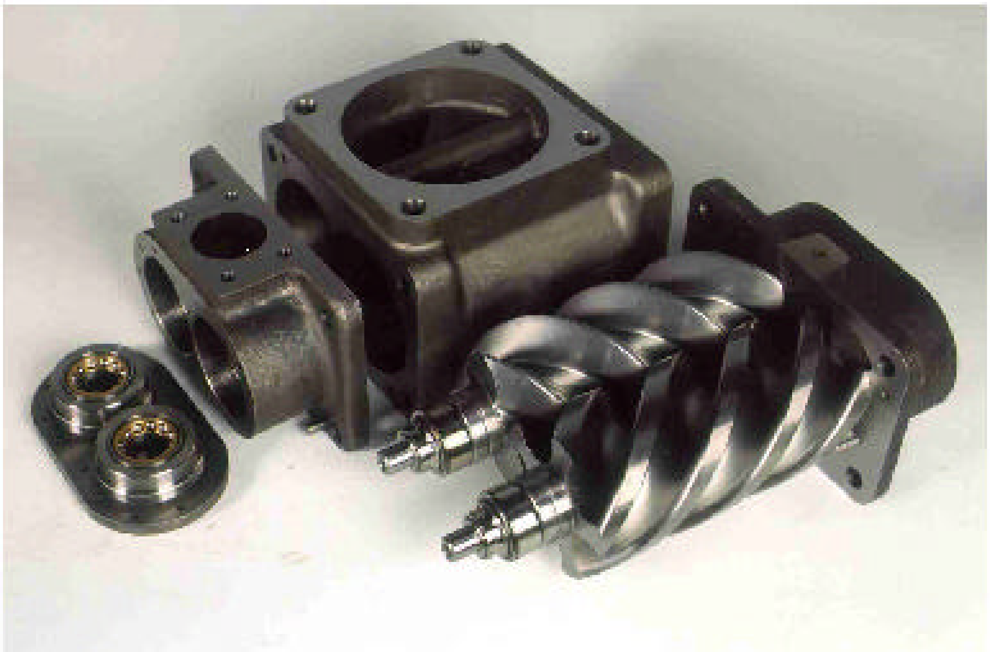

Fig. 2 Components of an Oil-less Screw Machine

Finally the experience gained in the compressor and expander designs was used to design an expressor unit based on an idea first proposed by the authors [Smith and Stoš iæ 1995]. This was to recover work from the expansion process in a refrigeration vapour compression cycle by means of a sealed unit

containing a twin screw expander driving a twin screw compressor. Work recovered from the two-phase
expansion of the liquid would then be used to recompress part of the vapour formed during expansion directly. The concept was developed further and the sealed expressor unit described in this paper used only a single pair of rotors for both the expansion and compression processes, as shown in Fig 2.

Apart from cost saving, there are a number of reasons for an expressor being superior to a coupled expander and compressor. Firstly, by performing the expansion and compression processes within sealed unit, no external drive shaft is required and hence the shaft seals required for a separate expander and compressor and any mechanical losses associated with them are eliminated. Secondly, the axial forces associated with expansion and compression act in opposite directions. The unit is thus balanced and the mechanical losses associated with the thrust bearings are negligible. In addition, since there is only one set of bearings in a two rotor expressor, the losses associated with them can not be attributed to both processes. Hence, if these are allocated to the expansion process, then the vapour recompression may be assumed to take place without any effective mechanical losses or vice versa. Finally, the kinetic energy imparted to the vapour at the end of expansion is retained at the start of compression, thereby further reducing the fluid dynamic losses associated with the compression process.

In the case of the single rotor pair expressor a 3/5 lobe configuration was chosen but as in the case of the expander, the same "N" profile was used for them. By this means, as in the expander which preceded it, timing gear, seals and an oil lubrication system were eliminated

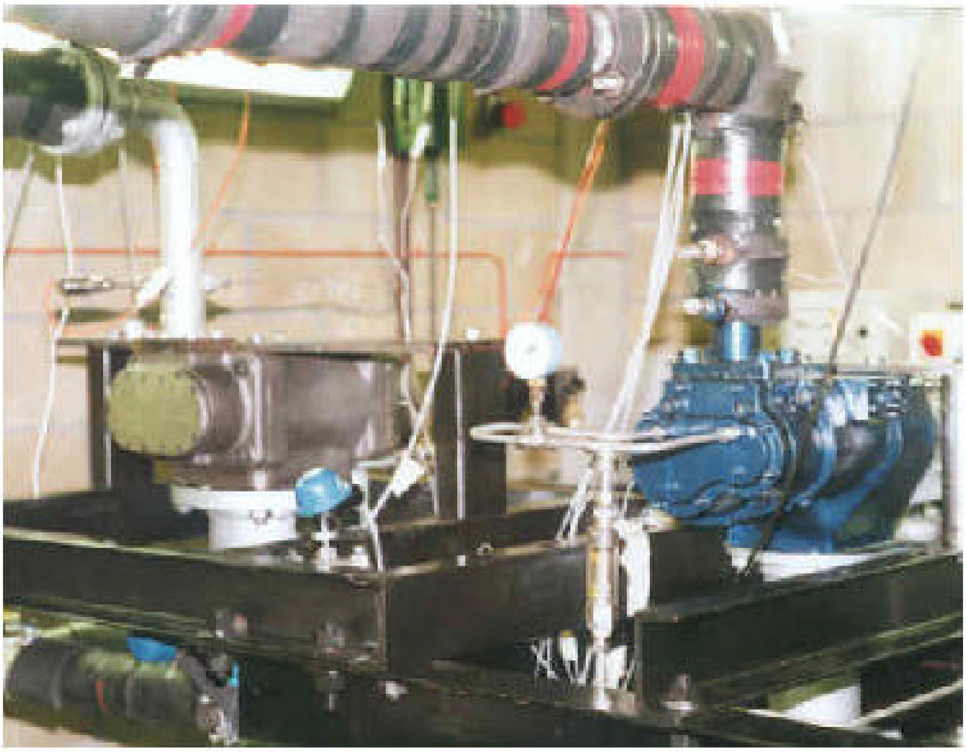

Fig. 3 The Expressor and Expander in the Test Rig

Some special features were required in the rotor design to make one pair equally suitable for both expansion and compression with an adequate interval between the end of expansion and the beginning of vapour recompression to permit the low pressure liquid and excess vapour to be discharged. To obtain high efficiency in both the expansion and compression processes, the profiles were modified to produce sealed contact between both sets of surfaces. To permit the liquid and excess vapour to escape, the wrap angle was greatly increased. The rotor line of contact with the casing could then be displaced axially by at least one lobe pitch length after expansion was complete without any significant change in the trapped volume. This would permit a large helical port to be formed in the casing at the end of the expansion process through which discharge could take place before recompression began.

The rotors were manufactured and paired by a specialist rotor manufacturer and to obtain the maximum benefit from them, all other components were designed to take advantage of their special features. Thus, special care was taken to minimise the flow losses in the suction and discharge ports. The ports were positioned in the housing to let the fluid flow with the fewest possible bends and the fluid velocity was kept low by making the flow area as large as possible. The cast iron casing, which was carefully dimensioned to minimise its weight, contained a reinforcing bar across the suction port to improve its rigidity at higher pressures.

The entire expressor was designed by the authors and its components were manufactured under their direction. The whole unit was then assembled and initially tested at City University.

3. Experimental Investigation

Both the expressor and the expander were installed and tested in the same rig. They were arranged to form two parallel loops within a sealed system through which refrigerant was circulated by means of a pump with compressed liquid heated and expanded vapour condensed in shell and tube type heat exchangers. A detailed description of this is given by [Smith et al 1999]. The expander was coupled to a dynamometer to measure the power output while the expressor performance was estimated from the delivery rate of compressed vapour. The two machines were installed side by side as shown in Fig 3 but could not be operated simultaneously. The fluid contained in the test loop was R-113.

After approximately six hours of running, the expressor unit was stripped down and the rotors and bearings inspected. Also, the specific gravity of the refrigerant was measured. From this it was found that the working fluid contained approximately 3% oil. This was the result of earlier test work on expanders with timing gear and oil lubricated bearings where the shaft seals had proved to be inadequate and the oil had been absorbed by the This residual oil appeared to be insufficient to affect the vapour-liquid equilibrium line and hence had no adverse effects on performance. However, a considerable residue of oil was found in the bearings. It was concluded that local evaporation of the liquid refrigerant used for lubricating them was responsible for this deposit. Consequently, even though the bearings used were only of plain steel, they showed no signs of wear. Similarly, the rotors, though uncoated, were virtually unmarked. The mechanical design of the unit therefore appears to be robust. Both the expander and the expressor run smoothly and quietly with continuous oil-less operation and the experimental programme for their development is continuing.

Test results so far obtained from the expressor show that the concept of combining expansion and compression in one set of rotors is a valid one and that the measured machine performance is quite close to that predicted. When these are complete it will be installed in an industrial chiller unit to determine its reliability when operating in an oil-less environment and what modifications are required to the chiller control system in order to obtain stable operation over its entire operating range.

4. Conclusions

The method of twin screw rotor profile generation developed by the authors over a long period has been shown to be highly suitable for the design of machines to operate in an oil-less environment. This has been confirmed by tests on both coated and uncoated rotors used for compression, expansion and the simultaneous action of both processes . Unlike existing conventional methods of rotor profile design, which are only known to a limited number of individuals, the rigorous mathematical procedure on which such profile generation is based has the additional advantage of simplicity. This permits rotors for any required application to be developed by non-specialist mechanical designers.

Using this procedure an experimental expressor has been designed, built and tested. Results obtained from it confirm that such rotors can run without timing gear or oil lubrication even when both surfaces of the rotors are used to provide a seal against internal fluid leakage.

Literature

Jacobson B, 1996: *Ball-bearing lubrication in refrigeration compressors. Proc 1996 International Compressor Conference at Purdue, Purdue University, West Lafayette, IND, USA, July 23-26, pp 103-108*

Kauder K. and Daemgen U, 1994: *Wasserspritzung in Schraubenkompressoren (Water Injection into Screw Compressors) "Schraubenmaschinen 94" VDI Berichte Nr. 1135 Duesseldorf*

Smith I. K. and Stošiæ N, 1995 *The Expressor: An efficiency boost to vapour compression systems by power recovery from the throttling process. AES-vol.34, Heat pump and refrigeration systems design, Analysis and applications, ASME, p 173*

Smith I. K, Stošiæ N, Aldis C. A. and Kovaèeviæ A, 1999: *Twin Screw Two-Phase Expanders in Large Chiller Units, Proceedings of IMechE Conference 'Compressors and Their Systems' London, 13-15 September*

Stošiæ N, Milutinoviæ Lj, Hanjaliæ K. and Kovaèeviæ A, 1992: *Investigation of the Influence of Oil Injection Upon the Screw Compressor Working Process, Int. Journal of Refrigeration, Vol. 15, No 4, 1992.*

Stošiæ N. and Hanjaliæ K, 1997 *Development and Optimization of Screw Machines with a Simulation Model, Part I: Profile Generation, ASME Proceedings, Journal of Fluids Engineering, Vol 119, p 659*

Stošiæ N, 1997: *Plural screw positive displacement machines. Int Pat Application No: WO 97/43550*

Stošiæ N, Smith I. K, Kovaèeviæ A and Aldis C. A, 1997: *The design of a twin-screw compressor based on a new rotor profile. Journal of Engng Design, Vol 8, No.4 pp389-399*

Stošiæ N. Smith, I. K, Brasz J. J. and Sishtla V, 1998: *The performance of a screw compressor with involute contact rotors in a low viscosity gas-liquid mixture environment, "Schraubenmaschinen 98" VDI Berichte Nr. 1391, pp 279-292.*

Stošiæ N, 1998: *On Gearing of Helical Screw Compressor Rotors, Proceedings of IMechE, Journal of Mechanical Engineering Science, Vol 212, p 587*

Professor dr Nikola Stošiæ, dipl.ing.

Royal Academy of Engineering Chair in

Positive displacement Compressor Technology

Centre for Positive Displacement Compressor Technology
City University London
Northampton Square, London EC1V 0HB, England
tel: +44 20 7447 8925, fax: +44 20 7477 8566,
e-mail: *n.stosic@city.ac.uk*