DEVELOPMENT OF A TWIN SCREW EXPRESSOR AS A THROTTLE VALVE REPLACEMENT FOR WATER-COOLED CHILLERS

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ABSTRACT

An introductory report is given on a novel means of replacing the throttling process in vapour compression systems. 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 twin screw machine with only one pair of rotors. These rotate, without the need for timing gear, in a process-lubricated, totally sealed unit which the authors have called an "expressor". First 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 give clear indications that it should be a highly cost effective component in large commercial chiller systems.

INTRODUCTION

Power recovery from the throttling process in vapour compression cycles could be a means of improving refrigeration and air conditioning system performance. Two-phase turbine technology has been applied and is currently commercially available on certain chillers. However, the cost of adding throttle loss recovery equipment based on the two-phase flow turbine technology prevents across-the-board use of two-phase flow turbines on water cooled chillers. The reason is not so much the cost of the turbine itself as its applied cost, i.e. the cost of the required controls (upstream and by-pass valves) and the required piping between the turbine and the evaporator and condenser vessels. This paper describes a program that has as its main objective to investigate whether alternative two-phase expansion technology (screw expansion) could result in a lower applied cost for throttle loss power recovery.

A major problem associated with any system of power recovery from the throttling process is how to utilise the power generated. The two main methods initially considered were either to link the expander to an independent electrical generator or to link the expander directly to the main compressor drive. Both have some disadvantages.

Electrical power generation involves the added cost of the generator, the need for constant speed control at part load and the fact that the power generated undergoes two additional energy transformations before it can be utilised to reduce the power input to the main compressor. Linking the expander directly to the main compressor drive is more attractive from an efficiency perspective but more expensive to implement. One reason for this is that if a long connecting drive shaft and shaft seals are to be avoided, the expander must be located close to the main compressor and contained in the same casing. Large bore piping is then needed to reduce pressure losses in transferring the liquid from the condenser discharge to the expander and then returning the expanded fluid back to the evaporator inlet. Another is because the expander needs controls for part load operation at constant speed.

THE EXPRESSOR CONCEPT

In 1995 two of the authors [1] proposed linking a twin screw expander to a twin screw compressor in a sealed unit to form an independent free running device. The expander would replace the throttle valve and the power thus recovered would recompress a portion of the vapour formed during the expansion process to deliver it directly to the condenser inlet, as shown in Fig 1. The concept was based on a long term investigation of twin screw machines operating as two-phase expanders. From this it had been shown that they had potentially higher adiabatic efficiencies than turbomachines developed for the same purpose and that this high efficiency could be maintained nearly constant over a wide range of speeds and loads. A key feature of the study was the demonstration that such a combined unit was capable of stable operation over a wide operating range. To emphasise the combination of expansion and compression processes in the same unit it was called an expressor.

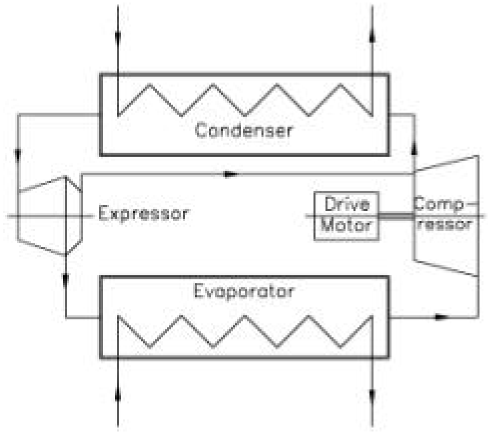

Fig. 1 Chiller unit with the expressor as altemative to the throttle valve

The expressor could be located directly between the condenser exit and the evaporator inlet with short straight pipe connections. Also, the speed of such a unit could vary according to the chiller unit duty without the need for additional controls.

A number of disadvantages were associated with this concept which made its manufacturing cost rather high. These were as follows:

The need for timing gear to avoid rotor contact without oil in the working fluid.
The high cost of seals to avoid gear and bearing lubricating oil mixing with the working fluid.
The need for two sets of rotors in which to carry out the expansion and recompression processes.

THE TWIN ROTOR EXPRESSOR

Beginning in 1996, the authors set out to determine whether an expressor unit could be developed at an economic cost despite the itemised disadvantages.

The starting point was to determine whether two-phase expansion and vapour compression could be achieved in twin screw machines without either the use of timing gears or oil entrainment in the working fluid. The basis of this was the use of advanced profile rotors developed by one of the authors [2] in which the relative motion between the male and female rotors was nearly pure rolling along the contact band and where the contact forces were very small. An extensive experimental programme was first carried out with a specially designed oil injected air compressor which ran for over 150 hours with only water as the rotor coolant [3]. This was followed up by the testing of a two-phase expander using the same profile rotors operating on R113 containing only a residual trace of lubricating oil [4].

Attention was given to recent developments in rolling element bearing design [5]. These led the authors to believe that the liquid component of the expanding two-phase working fluid would be sufficient to lubricate modern plain steel bearings. Should this be the case, then the two-phase expander would not require any internal seals since the whole machine would contain only the process fluid. Accordingly, the two-phase expander already mentioned [4] was built without internal seals and with process fluid lubricated rolling element bearings. The unit operated successfully and achieved a peak adiabatic expansion efficiency of 76%.

An interesting feature of the expander test programme was the presence of residual oil in the test loop. This was not admitted to the working fluid by design but was the result of earlier tests on two-phase expanders. These had all been built with internal seals. Several types were tried but all leaked and led to oil contamination of the working fluid. The oil was never wholly removed even after evaporating and recondensing the entire fluid inventory in a separate vessel. It was found that by allowing the liquid refrigerant to enter and leave the bearings fairly freely, the oil, though barely detectable in the bulk fluid, rapidly built up a high concentration there. It was concluded that bearing friction was causing the refrigerant to evaporate locally but leaving the oil in place.

The next stage of the programme was to determine whether the expansion and compression processes could be carried out in the same set of rotors by taking advantage of the principle that, when operating as an expander, after the maximum displacement is attained between a pair of twin screw rotors and their casing, further rotation leads to a decrease in the trapped volume. In normal expander operation, this region would be open to the discharge port, thereby leading to constant pressure discharge. Delay in exposing the volume of fluid, thus entrapped, to the discharge port would lead to its recompression.

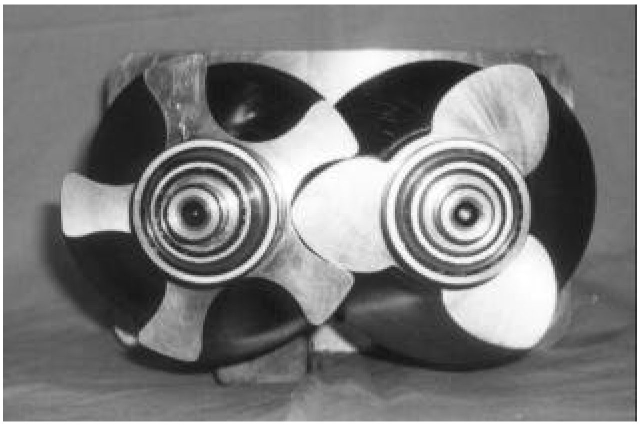

Fig 2 Expressor rotors

Two problems associated with this were the need for a sealing line between the rotors on both contact faces and a large enough port in the casing to enable virtually all the liquid to be discharged at the end of expansion process before recompression of the residual vapour started.

The sealing problem was solved by modifying the rotor profile to form a sealing line between both sets of contacting surfaces. The resultant shape is shown in Fig 2.

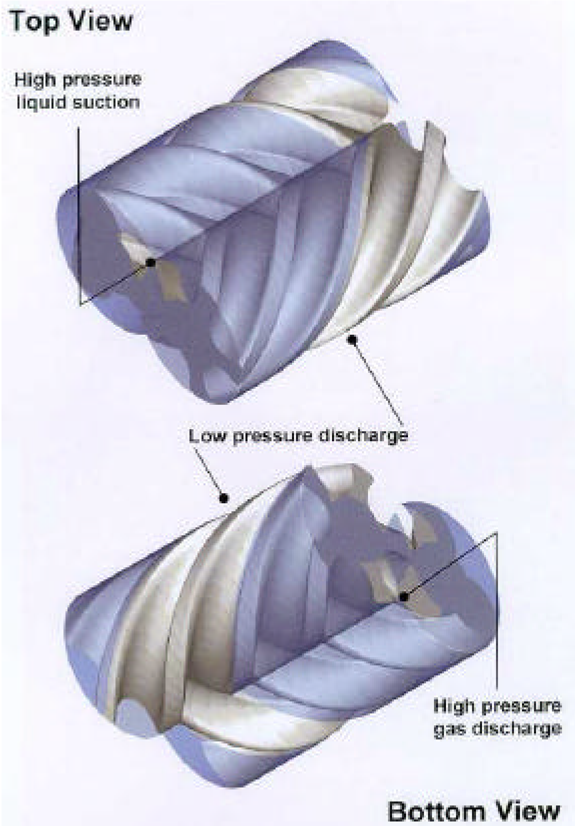

Fig. 3 Operating principle of a twin-rotor expressor

To permit the liquid and some of the vapour to discharge before recompression began, the rotor wrap angle was increased to over 450^{o}. The trapped volume between the rotors and the casing then remained virtually constant over a relatively large angle of rotation. A discharge port was then formed in the casing over the whole of this region without adversely affecting either the expansion or compression processes. The resulting configuration demonstrating this principle is shown in Fig 3.

Fig 4 The expressor

Views of the prototype machine, designed according to these principles, are shown in Fig 4 while a disassembled view of the components is shown in Fig 5.

Fig 5 Expressor parts

Apart from its mechanical simplicity and inherent low manufacturing cost, such a twin rotor expressor has the following advantages over the linking of a separate screw compressor and expander.

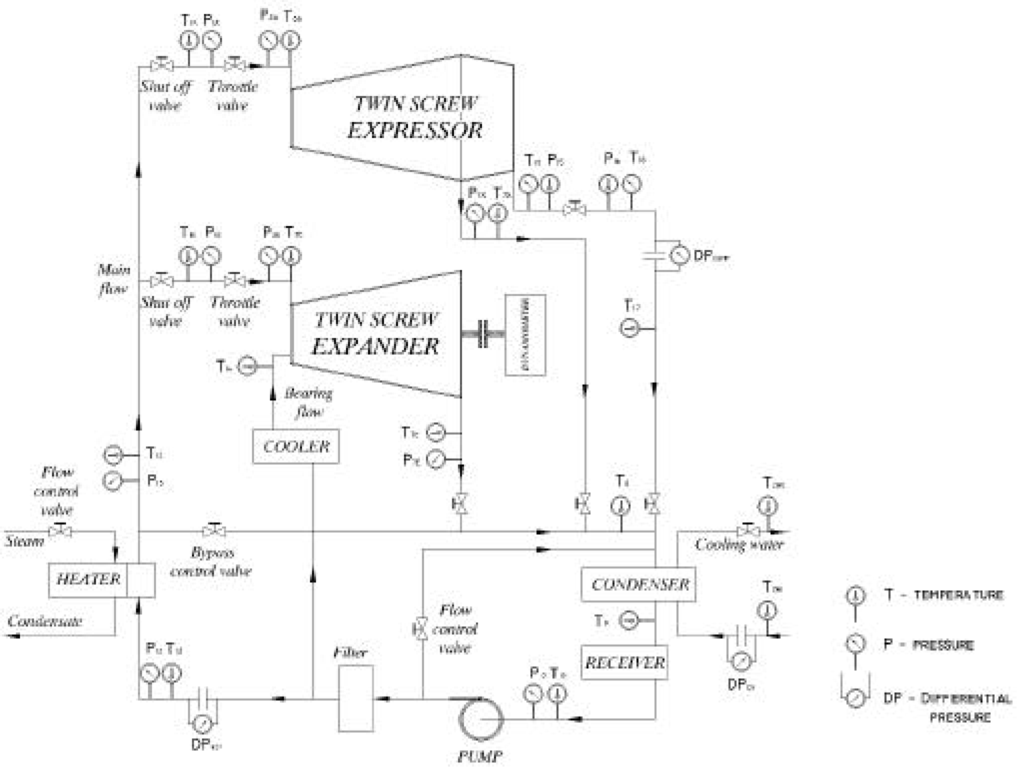

Fig 6 Layout of the expressor test rig

All shaft seals and the mechanical losses associated with them are eliminated.
Axial bearing loads are almost eliminated.
The mechanical losses associated with the bearings may be attributed either to the expansion or the compression process but not to both.
There is no entry suction loss to the compressor since the working fluid is already trapped within the compression volume at the end of expansion.

The twin screw expressor rotates freely according to the flow requirements. However, a key feature of the two-phase expansion process within it is that, as previously shown in [6], the expansion ratio attained by the working fluid increases at higher rotational speeds. It follows that at part load chiller operation, when both the mass flow rate and the pressure differential between condenser and evaporator are reduced, there is no need to include a control system to alter the built in volume ratio in order to maintain the expansion efficiency of the device. The speed reduction associated with reduced flow automatically allows for this. Hence good overall adiabatic efficiencies for the combined expansion and compression processes were anticipated for a machine without controls over the entire operating range.

THE DETERMINATION OF EXPRESSOR PERFORMANCE

A twin rotor expressor unit, as described in the previous section, was designed and constructed at City University, London, and by the time of writing this paper, preliminary test results had been obtained.

The experimental rig, on which the tests were carried out, is a modified version of an earlier rig used for testing two-phase expanders and is shown in Fig 6. The modification was made by forming a parallel loop for the expressor so that the same facility could be used to test either an expander or an expressor.

In its original form this rig was designed for two-phase power generation from working fluids operating at higher temperatures with condensation taking place at temperatures well above atmospheric. Refrigerant 113 was chosen as the working fluid for this purpose. When testing the expressor, working conditions had to correspond as closely as possible to those in a chiller operating on R134a. Thus identical inlet and exit pressures were maintained and mass flow rates of R113 were selected which it was estimated would correspond to those of R134a at the same rotational speeds. However, the working fluid temperatures were much higher than those of R134a in a chiller.

$$\eta_{Expressor} = \eta_{Expander} . \eta_{Compressor}$$

In order to evaluate the expressor performance, it was necessary to define an acceptable measure of its efficiency. This was taken to be:
Determining these values presented some problems, mainly because it is not possible to measure the torque transfer from the expansion to the compression sections in a single pair of rotors. Hence no direct estimate could be made of the actual enthalpy drop during the expansion process from which the actual power output and expansion efficiency are estimated. However equation [1] can be rewritten as:
Since the isentropic enthalpy change associated with expansion can be readily estimated and that of compression can be determined if the vapour inlet condition is known, the remaining parameters to be measured are the compressed vapour mass flow and discharge pressure.

$$\eta_{Expressor} = \frac{\dot{m}_{Compresion}}{\dot{m}_{Expansion}} . \frac{\Delta h_{Isentropic\,Compression}}{\Delta h_{Isentropic\,Expansion}}$$

The main difficulty associated with this procedure was that over the range of pressures required to simulate chiller operation, dry saturated R113 vapour at the compressor inlet would discharge as wet vapour at the end of compression. Even after throttling the compressed vapour, dry vapour could only be achieved if the liquid carry over was less than 4%. However, performance simulations of a complete chiller unit showed that a higher carry over of liquid in the expressor would hardly affect its efficiency since the excess liquid would simply recirculate through the condenser.

Efficiency estimates for the expressor, derived from the test results and based on the assumption of dry vapour flow after throttling the compressed vapour, were found to be too high to be credible. An alternative slightly simplified analysis was therefore used to process the measured data, based on the swept volume of the compressor and an assumed volumetric efficiency.

TEST RESULTS

Earlier studies [6] had shown that, for high adiabatic efficiencies, the built in volume ratio of a two-phase screw expander needed to be significantly less than the actual volumetric expansion ratio of the fluid passing through it. On this basis, the prototype unit was constructed with an estimated built in volume ratio of 2.85:1 to achieve an overall expansion ratio of 11.4:1 with R134a as the working fluid and approximately 12.9:1 with R113 working with identical inlet and exit pressures. Even at this low value of the built in volume ratio there was evidence of over expansion. A succession of tests was therefore carried out with the built in volume ratio progressively reduced to approximately 1.85:1.

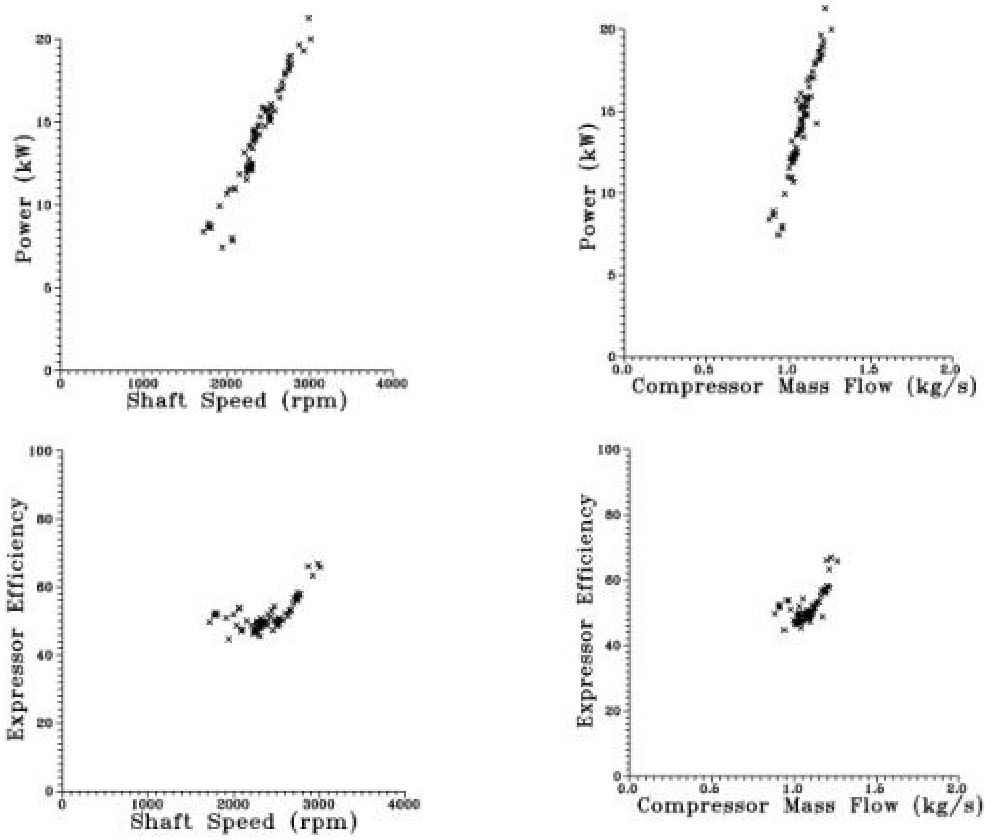

Fig 7 Expressor performance charts

The adiabatic efficiencies obtained from the final build are shown in Fig 7. As can be seen, efficiency values varied from approximately 60% at half load to 50% at the design point. The estimated power recovery over this range was roughly from 10-20kW. These values could probably be improved because pressure-volume measurements of the processes within the machine showed pressures 2 bar higher than the discharge value within the compression section prior to the opening of the vapour discharge port. At this stage it seems fairly certain that overall efficiency values at the design point of the order of 55% are attainable. This corresponds roughly to an expansion efficiency of 70% combined with a compression efficiency of 80%.

CONCLUSIONS AND FUTURE PROGRAMME

The twin rotor expressor is derived from a continuing investigation into the potential of twin screw machines to expand two-phase fluids efficiently which has been conducted over a period of eighteen years. The resulting machine, comprising a high efficiency sealed unit with only one pair of rotors, process lubricated and without the need for any seals or timing gear, is potentially a highly cost effective means of power recovery from the throttling process in vapour compression cycles. Further studies are needed to determine the best built in volume ratios for the expansion and compression processes and to improve liquid-vapour separation during low pressure discharge. A revised design to address these problems is currently under construction.

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