POWER RECOVERY FROM LOW COST TWO-PHASE EXPANDERS

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ABSTRACT

It is well known that power recovery from liquid geothermal brines could be considerably increased by means of efficient two-phase expanders. Hitherto proposals for them have not been economically viable either due to the low adiabatic efficiencies of the expanders or their relatively high manufacturing cost.

It has already been reported how, as a result of a long term research and development programme, carried out at City University, London, screw machines have been designed, built and tested there which can expand two-phase fluids with adiabatic efficiencies of more than 70%. A more recent and major advance in this work is that by the development of special rotor profiles and the use of up to date bearing technology, such machines have been successfully operated without external timing gear to prevent rotor contact, and with process fluid lubrication of the bearings. By this means the need for an oil lubrication system and internal seals has been eliminated.

These developments have led to a huge reduction in the cost of such machines and make it possible to consider the Trilateral Flash Cycle system as a viable alternative to Organic Rankine Cycle systems when considering geothermal power plant applications where binary systems are preferred to flash steam.

INTRODUCTION

Since the early nineteen seventies, when it was recognised that flash expansion to derive dry steam from liquid dominated geothermal resources resulted in a large loss in power recovery potential, efforts have been made to develop two-phase expanders to increase the recoverable power output.

In 1973, Sprankle [1] was granted a patent for the use of a screw machine to expand wet steam or even pressurised hot water as a means of recovering power from liquid or low dryness fraction geothermal brines. The temporary crisis in oil supplies at that time gave a strong incentive to investigate this device and alternative modes of expanding two-phase fluids in what was termed the “total flow” process. In parallel with the work on screw machines, other investigations were carried out on axial flow turbines [2], biphase turbines [3] and more recently on two-phase reaction turbines of the Hero type [4] in which to expand water steam mixtures more efficiently.
Apart from geothermal power applications, it was also recognised that throttling processes in

chemical process plant and in large vapour compression plant used for air conditioning and heat pump units could be made more efficient by power recovery from them using radial inflow turbines, as shown by Swearingen [5].

All practical demonstrations of these proposals were characterised by relatively poor expansion efficiencies. The highest values were for radial inflow turbines for which adiabatic efficiencies of 67% were claimed.

A screw expander is of the positive displacement type. It comprises a meshing pair of helical lobed rotors contained in a casing which together form a working chamber, the volume of which depends only on the angle of rotation.

The first test results obtained from small screw expanders, which operated on water or water/steam mixtures, are given by Steidel et al [6]. This was followed by further publications [7-9] in which some limited efforts were made to model the expansion process either by empirical curve fitting or highly simplified and limited modelling. The highest adiabatic efficiency obtained from these tests was only 53%.

Taniguchi et al, after unsuccessful attempts to incorporate screw expanders in engine driven heat pumps for district heating [10], carried out a detailed analytical and experimental study of two-phase screw expanders using R12 as the working fluid [11]. The method of analysis used was identical to that developed for screw compressor performance prediction and is based on the assumption of the non steady flow energy equation applied to the volume of fluid trapped between the rotors and the casing and quasi steady one dimensional flow assumptions for the fluid flow through all ports and leakage passages. The performance predictions appeared to coincide well with the measured values on a small scale unit with rotors of 81 mm diameter. These led to predictions of internal adiabatic efficiencies approaching 80% in larger machines which would be reduced by bearing and timing gear friction losses.

In 1981, the authors began an independent investigation of two-phase expansion processes based on the use of the screw machine as the prime mover. It was soon found that such expanders were unsuitable for the huge volume flow rates and volume ratios required for the expansion of water to normal condensing temperatures. From the outset, therefore efforts were concentrated on processes involving organic fluids where, for equal heat utilisation over the same temperature range, exit volume flow rates could be reduced by a factor of 10 or more and expansion ratios by two orders of magnitude. The work, which was carried out over a long period, has been published in detail [12,13]. The main results of the studies reported are as follows:

Provided that two-phase adiabatic expansion efficiencies of at least 75% could be attained, a Trilateral Flash Cycle (TFC) system using light hydrocarbons as working fluids, as shown in Fig 1, could recover almost double the power from a hot liquid stream than was possible from either flashed steam or indirectly heated simple Organic Rankine Cycle (ORC) systems. Subsequently, it was found that such a system had previously been proposed with a turbine as the expander [2]. However, the adiabatic efficiencies attainable with a turbine were too low to make it worthwhile.

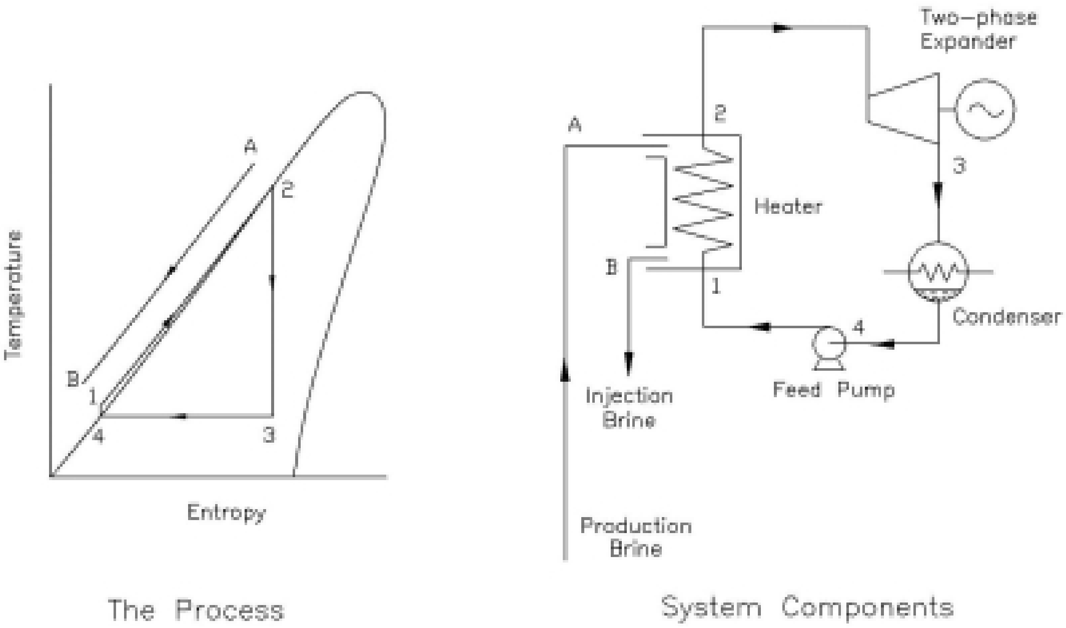


Fig. 1 Trilateral Flash Cycle (TFC) Power Recovery System

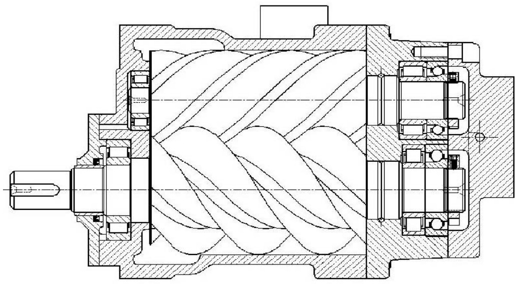
The key feature to obtaining high adiabatic efficiencies with positive displacement machines as expanders is for the built in volume ratio of the expander to be substantially less than the actual volumetric expansion ratio of the fluid being expanded [14]. By this means it is possible for screw machines to expand two-phase fluids with adiabatic efficiencies in excess of 75%.

RECENT IMPROVEMENTS IN SCREW MACHINE DESIGN

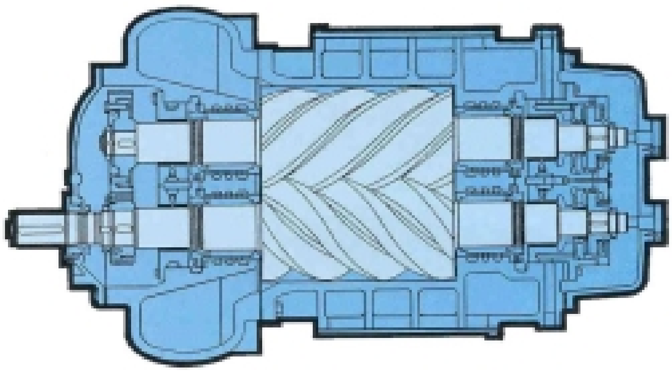
The main feature of a screw machine design, which determines its adiabatic efficiency, is the profile of the rotors. These must form a good seal both between each other and between the casing at all rotational positions in order to minimise internal leakage. The pioneering work which led to profiles with both effective sealing and a large flow area through which the working fluid could flow was carried out by the Swedish company, SRM, and most rotor designs used today are derivatives of these. The relative motion between the lobes in these profiles is a mixture of rolling and sliding. Thus direct contact between the lobe surfaces must be avoided to prevent seizure. A consequence of this has been the development of oil injected and oil free types as shown in Fig 2.

The oil injected machine relies on relatively large masses of oil injected with the compressed gas in order to lubricate the rotor motion, seal the gaps and reduce the temperature rise during compression. It requires no internal seals, is simple in mechanical design, cheap to manufacture and highly efficient. Consequently it is widely used as a compressor in both the compressed air and refrigeration industries.

In the oil free machine, there is no mixing of the working fluid with oil and contact between the rotors is prevented by timing gears which mesh outside the working chamber and are lubricated externally. In addition, to prevent lubricant entering the working chamber, internal seals are required on each shaft between the working chamber and the bearings. In the case of process gas compressors, double mechanical seals are used. Even with elaborate and costly systems such as these, successful internal sealing is still regarded as a problem by established process gas compressor manufacturers. It follows that such machines are considerably more expensive to manufacture than those which are oil injected.



Oil injected



Oil Free

Fig. 2 Oil Injected and Oil Free Compressors

A further feature of oil lubrication, which adds to the cost of an oil free machine and, to a more limited extent to the oil injected machine, is the system external to it which is required to supply the lubricating oil to the bearings and cool it before recirculation.

A high percentage of oil contained in the working fluid will inhibit the two-phase flashing process. Hence, screw machines must be of the oil free type if they are to expand two-phase fluids efficiently and all experiments reported on them acting in this mode hitherto were on machines of this type. Detailed studies of such expanders showed that for large units, as required for a TFC geothermal power plant, the unit cost of the expander alone would be of the order of US$1500/kW. This is of the same order of magnitude as the acceptable cost of the complete system, including pumps and heat exchangers. In the case of smaller units required for throttle valve replacement applications, the unit costs were even higher. It followed that, despite their good adiabatic efficiencies, two-phase screw expanders were not cost effective.

Apart from the interest in screw machines as expanders, there has been a continuing need to improve their operation as compressors. Thus, although the oil injected compressor is simple and cheap to make, the presence of oil in large quantities causes high rotor drag forces. This reduces the rotational speed possible with high efficiency. Oil injected compressors therefore require larger rotors than oil free machines to pass the same volume flow.

To minimise the adverse effects of oil injection, continuous efforts have been made to investigate its influence on screw machine processes, as reported by Stosic at al [15], while many attempts have been made to maintain safe rotor contact with only a small quantity of injected oil, with oil of reduced viscosity or even with water as the lubricant. Such efforts almost inevitably led to rotor seizure.

The main cause of the failures was considered to be either unsuitable rotor profiles or surface material properties [16]. A considerable effort has therefore been made to develop screw machine rotor profiles which permit sustained oil-free or oil-less operation.

Two recent developments have led to great simplification in two-phase screw expander mechanical design. Firstly, Mathematical studies of rotor profiling based on gear tooth generation theory, carried out by one of the authors [17-19], have led to the development of a new family of rotor profiles, as shown in Fig 3. These offer many advantages over more traditional shapes but from the viewpoint of the expander design, the most significant are the low rotor contact stresses between the male and female rotors and almost pure rolling motion along the contact band between them. The authors concluded that rotors with these profiles would not require timing gear provided that even a low viscosity fluid is present in the working chamber. This was confirmed by extensive testing in an air compressor designed for oil injection with rotors lubricated by water only [20].

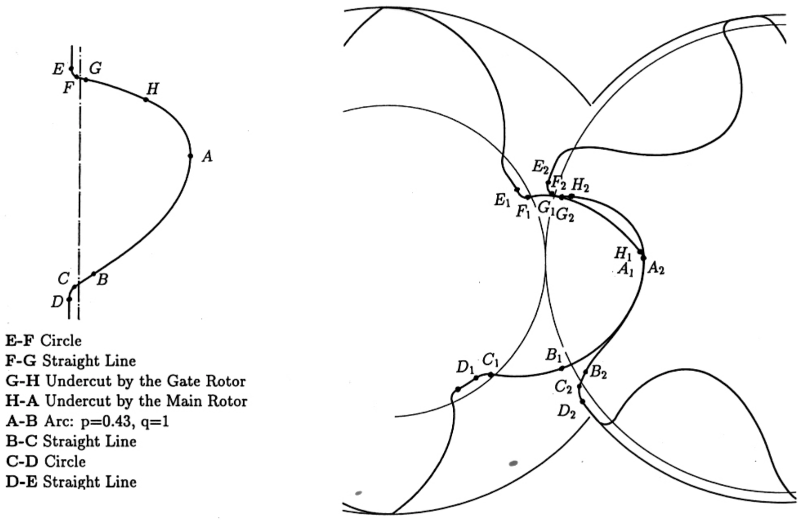


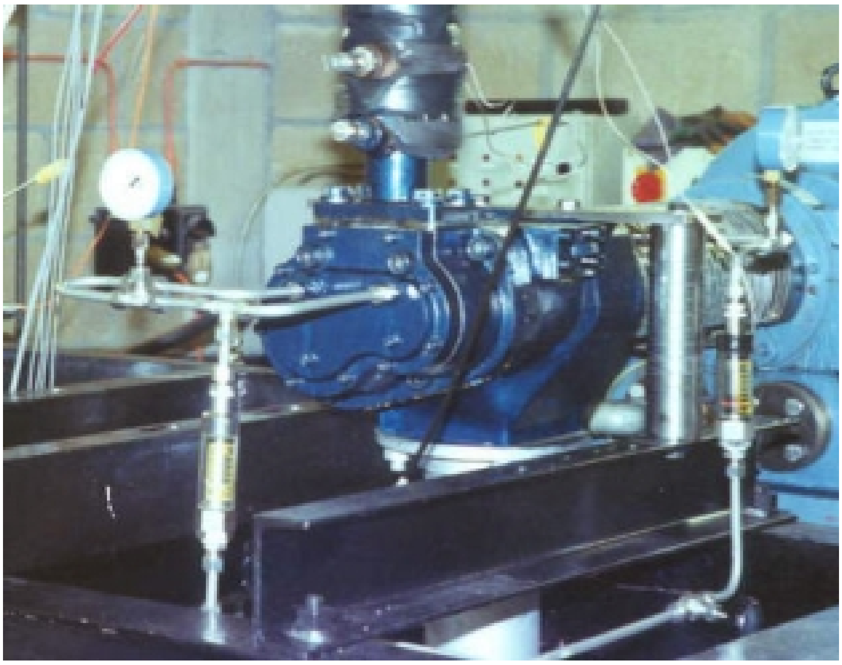
Fig. 4 'N' Screw Rotor Profile

Secondly, advances in the design of rolling element bearings [21], have shown that these may be used in refrigeration screw compressors to take both radial and axial loads without an oil supply, provided that there is a small trace of oil in the refrigerant. In the case of compressors, the oil would be present in the form of a fine mist but in the case of a two-phase expander, it would be wholly dissolved in the oil. It was not certain how the bearings would perform under these conditions. Experimental confirmation of their use in expanders was therefore required.

Experimental Investigation

An experimental screw expander was designed and built by the authors, without timing gear but with rotors based on their new “N” profile and which used standard steel rolling element bearings without oil lubrication. Liquid refrigerant, which contained less than 1% oil, was permitted to flow fairly freely through the bearings by means of drilled passages and no internal seals were included in it. The only seal used was located at the drive shaft exit and this was of the standard refrigerator compressor shaft type. This wholly process lubricated machine had the added advantage that additional pumps, heat exchangers and filters needed for the oil lubrication system were thereby eliminated.

The expander was installed in a test loop to complete 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 in [23]. The expander was coupled to a dynamometer to measure the power output and is shown, installed, in Fig 4. The fluid contained in the test loop was R-113.



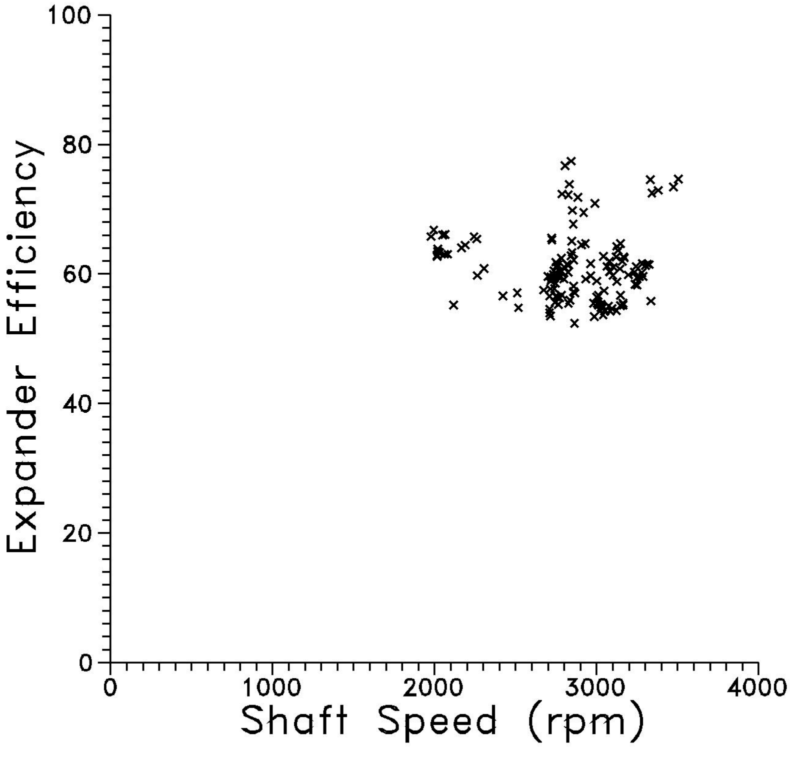


Fig. 5 Expander Test Results

The performance of this machine was satisfactory and although the design was not fully optimised from the thermodynamic point of view, peak adiabatic efficiencies of 76% were obtained, as shown in fig 5, the highest ever recorded for any type of two-phase expander. More importantly, it was found that local frictional heating in the bearings had the effect of evaporating the refrigerant while leaving the oil behind so that on strip down, the bearing housings were found to be full of oil. Consequently there was no detectable wear found either on them or the rotors after several hundred hours running.

The cost savings achieved by this design simplification were impressive. In 1984, the authors purchased their first screw expander which was simply a standard production oil free refrigeration compressor, made by Howden Compressors of Glasgow, Scotland, run in reverse. It had a swept volume of approximately 4.67 l/rev and cost US$72,000 at the then sterling/dollar rate of exchange. The experimental expander described in this paper was designed in its entirety at City University, London, and manufactured in 1997 as a one off unit. This had a swept volume of 1.56 l/rev and, excluding rotor tooling and pattern costs, which were not large, its manufacturing cost was only US$6,000. Allowing for the usual scaling factor of cost ratio = size ratio$^{0.65}$, then, apart from the further gains due to the effects of inflation over a period of 13 years, this amounts to a six fold reduction in price. There is little doubt that this figure could be further improved if such units were made on a batch production basis.

CONCLUSIONS

As a result of a long term research and development programme, it has now become possible to design and manufacture twin screw machines as efficient two-phase expanders at an economically viable cost. Although other applications for such devices may be possible, the most significant are in the Trilateral Flash Cycle system. This may now be considered as a possible contender for cost effective power recovery from liquid geothermal brine streams.

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