Cost Effective Small Scale ORC Systems for Power Recovery from Low Enthalpy Geothermal Resources

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ABSTRACT

Small scale geothermal power generation has, until recently, been inhibited by the relatively high cost of the power plant. Developments in the design and manufacture of screw machines have made it possible to modify standard mass produced air compressors to act as expanders in place of turbines in ORC units of up to approximately 500kWe output. Such machines have adiabatic efficiencies close to those of turbines of similar output but with a number of advantages. These include, very low cost, the ability to enhance the cycle efficiency by admitting wet vapor, direct coupling to standard 3-phase generators, and the elimination of traditional lubricating systems. Combining them with standard heat exchangers has made it possible to manufacture small ORC systems for cost effective power production at outputs of as little as 20-50kWe. An experimental unit, containing all these features, has been designed, built and tested. Preliminary results showed good agreement with performance predictions with the screw expander achieving an adiabatic efficiency of 74% at an output of only 22kW. This compares well with turbines of similar power. A detailed design study has shown that a 50 kW air cooled system, receiving heat from brine at only 100°C can be built and installed for a total cost of only $1500-2,000/kWe of net output, if water cooled, and approximately $2,500/kWe of net output, if air cooled. Extrapolating these figures to a 200kW unit, it is estimated that up to approximately 500kWe output, such screw driven ORC units are 30% cheaper than recently publicized low cost, turbine driven systems.

Introduction

ORC systems for recovering power from geothermal heat are now well developed. However, in order to make them more cost effective, efforts have been increasingly directed to units of ever larger sizes. More recently, Brasz (1) has shown that centrifugal compressor driven air conditioning chiller units can be converted to ORC systems at a relatively low cost, mainly, by adapting the compressor to operate, in reverse, as a radial inflow turbine. By this means it was claimed and has since been demonstrated (2) that units of as low as 200kWe power output can be built and installed cost effectively, even when operating with very low brine temperatures.

Heat sources of the order of only 250 kW, mainly as waste heat from small IC engine generator sets, are widely available. As fuel prices rise, power recovery from them would be very attractive commercially, if ORC units of the order of 20-50 kWe output could be built and installed for an economic price. At such outputs, the current use of turbines as expanders has many disadvantages, since they rotate at very high speeds, and hence require high ratio reduction gearboxes or expensive direct coupled electrical generators, as well as relatively costly lubrication systems.

A closer examination was therefore made of alternative types of expanders and the most promising of these was found to be the twin screw type. The use of screw expanders in ORC systems has been proposed before (3) and a small number of these have been constructed and operated. One of the most widely publicized of these units was a geothermal plant in Birdsville, Australia (4). In those cases, the plant designers lacked expertise in the engineering science required for the optimum design of such machines. Consequently, many of the potential advantages in their use were not realized.

This paper describes the combined efforts of a US company that recognized the potential for small scale ORC units, and a UK university, where research and development on screw expanders and compressors has proceeded for over 25 years, to produce a low cost ORC system. Although, primarily intended for engine waste heat recovery, these units are equally suitable for geothermal power generation.

Screw Expanders

Twin screw machines are mainly used as compressors, for air and refrigeration applications. As a result of much development, when operating in the oil flooded mode, they can now attain adiabatic efficiencies of up to 90%. This is comparable to that of the best aerodynamic compressors. However, unlike turbo machines, they are manufactured in large numbers and, especially when operating as air compressors, are relatively inexpensive. Their mode of operation has already been described to a GRC meeting (3). Mainly, it is worth repeating that, unlike turbo machines, in which the blading and stage requirements are completely different for expansion and compression, as shown in Figure 1, a screw compressor only requires adjustment of the shape of the high pressure port, to obtain the optimum built-in volume ratio, and reversal of the direction of rotation of the rotors, to operate equally well as an expander.

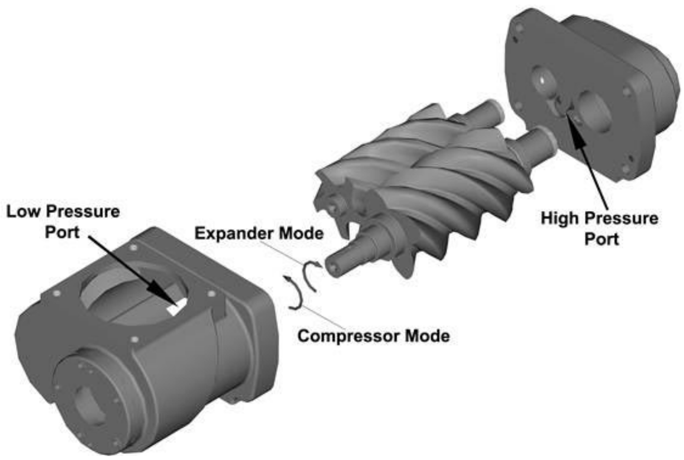

Figure 1. Twin Screw Machine as Compressor or Expander.

Rotor Profiles

Given the clearance between the rotors and the rotors and the casing, which is determined, mainly by manufacturing limitations, the most important feature of a twin screw machine, which determines flow rates and efficiencies, is the rotor profile. The earliest machines used a symmetric profile, as shown in Figure 2. This is very simple, with the male rotor shape constructed from only three circles, the centers of which are positioned either at the rotor centre or on its pitch circle. The female rotor profile then mirrors this with an equivalent set of circles.

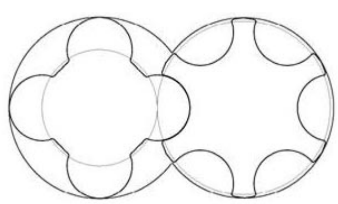

Figure 2. Symmetric circular profile.

The symmetric profile has a very large blow-hole area which creates significant internal leakage. This excludes it from any application where a high or even a moderate pressure ratio is involved.

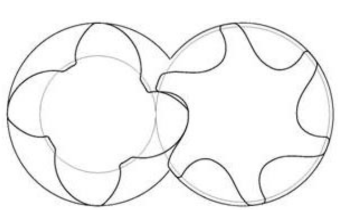

Figure 3. Rack generated 'N' Profile.

Since the introduction of the symmetric circular profile, many improved profiles have been developed and are in use. In this area, the "N" profile (5,6), as shown in Figure 3, was developed at City University. This has many advantages over any other known and is now used by some of the world's leading screw compressor manufacturers. In particular, its application raises the adiabatic efficiencies of these machines, especially at lower tip speeds, where gains of more than 10% over other profiles, in current use, have been recorded.

Rotational Speeds

Screw expanders are classified as positive displacement machines and though capable of higher rotational speeds than reciprocating, vane or scroll expanders, their optimum tip speeds are roughly one order of magnitude less than those of turbo machines. Accordingly, in most cases, they can be coupled directly to a standard 3-phase generator, without an intermediate reduction gearbox. This, not only saves in cost but also gives them an approximately 5% transmission efficiency advantage over turbines.

In the case of power outputs in the 20-50 kW range, the machines are rather small and 2-pole generators are not generally available. Hence for direct coupling, they are limited to rotational speeds of 1500/1800 rpm. This results in tip speeds of the order of only 8 – 15 m/s. These are low for optimum results. Although better efficiencies are attainable at higher tip speeds, the expander would then need a reduction gearbox, which would require oil lubrication. As will be shown, screw ORC expanders can operate without a separate oil lubrication system and therefore this is a feature to be avoided. At small power outputs a compromise solution is possible by coupling the expander to the generator with a belt drive. However, even at low tip speeds efficiency predictions, based on computer simulations and backed by experimental data, indicate that adiabatic shaft efficiencies of the order of 70% or more are possible. Thus at these low outputs, eliminating transmission losses, by direct coupling, results in overall expander-generator efficiencies, comparable to those attainable from small turbo-generator sets.

Optimizing the Cycle Efficiency with Wet Expansion

Typically, working fluids used or proposed for ORC systems are refrigerants, such as R124 (Chlorotetrafluorethane), R134a (Tetrafluoroethane) or R245fa (1,1,1,3,3-Pentafluoropropane), or light hydrocarbons, such as isoButane, n-Butane, isoPentane and n-Pentane.

As can be seen from the temperature-entropy diagram, shown in Figure 4, a common characteristic of these fluids is that as expansion proceeds, the working fluid entering the expander, initially as dry vapor, leaves it with some degree of superheat.

When the expander is a turbine, the working fluid entering it must be in the dry vapor phase in order to maintain a high efficiency and to prevent blade erosion by any entrained mist of droplets. However, one of the potential advantages of screw expanders is that they will operate with identical and even improved efficiencies if the working fluid is wet,

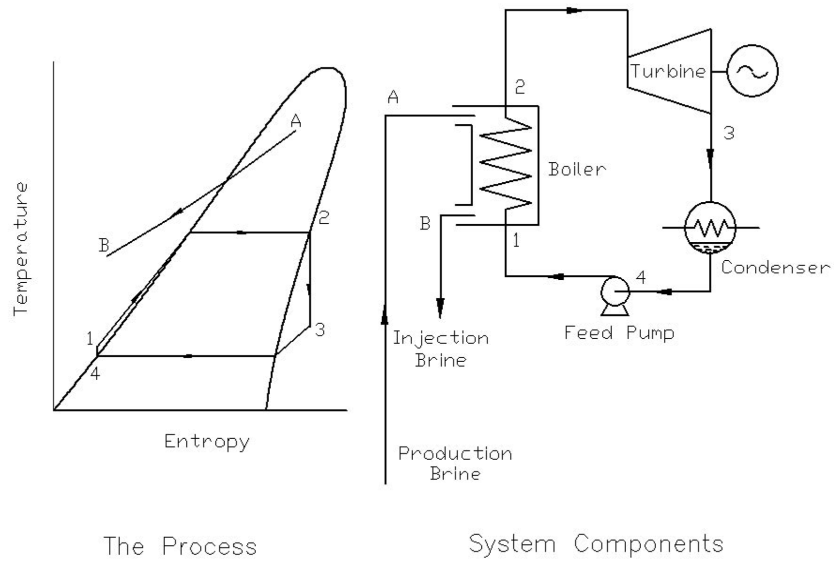

Figure 4. ORC System with Turbine as Expander.

since the presence of liquid seals the gaps between the rotors and the casing and enhances the lubrication of the meshing screw rotors. In that case, the Rankine cycle may be modified, as shown in Figure 5, to eliminate wasteful superheat at the expander exit and, given limiting brine temperatures, to raise the evaporating temperature.

Since less heat is required per unit mass flow to only partially evaporate the working fluid, given the same supply of heat, the mass flow rate of the working fluid is thus increased as its wetness at the expander inlet increases. Also, reducing the need for desuperheat after expansion tends to increase the specific enthalpy drop in the expander. Thus, all these effects tend to increase the power output. However, in the absence

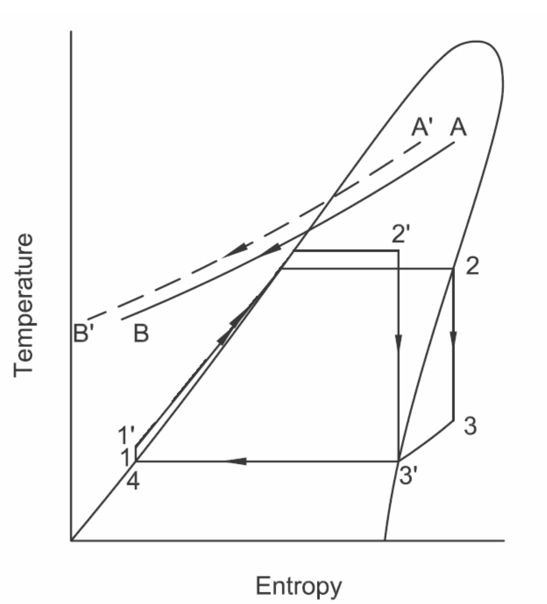

Figure 5. The effect of varying expander inlet dryness fraction.

of superheat at the end of expansion, there is a more general trend for the specific enthalpy drop in expansion to decrease as the initial dryness fraction of the vapor decreases and this decreases the power output. The net effect of these conflicting trends is that, overall, by correct choice of the initial dryness of the fluid entering the expander, the cycle efficiency can be increased.

To determine the optimum condition of the working fluid entering the expander, a large cycle simulation program was used, which has been developed by the authors over a period of many years. The input to this program is limited to defining the heat source mass flow rate, thermal capacity, its allowed temperatures at the inlet to and exit from the boiler and the permitted boiler pinch point value, together with the coolant thermal properties, initial temperature, temperature rise and condenser pinch point. Initial values for the boiler exit temperature and dryness fraction (where applicable) and condensing temperature are then assumed internally and a cycle analysis is performed taking account of internal pressure losses and pump, fan, expander, generator and motor efficiencies. The resulting output is then input to MINUIT (7), a standard multivariable minimization routine, which then alters the values of the boiler and condenser conditions until the optimum output is obtained. The expander inlet condition, thus derived, is dependent on the working fluid selected and the temperature range of the heating source and coolant but it generally leads to an optimum with an inlet dryness fraction of the order of 0.8-0.95, depending on the choice of working fluid, the inlet pressure and the condensing temperature. This inlet value corresponds closely to the vapor leaving the expander in the nearly dry saturated condition. The power output and, hence the efficiency of the cycle are then of the order of 3 -10% more than when the fluid enters the expander as dry vapor.

Since desuperheating the working fluid needs a larger heat transfer surface than condensing, and increased power output leads to reduced heat rejection, the effect of increasing the expander inlet wetness can lead not only to greater plant output but also, to reduced heat exchanger surface area.

From a cost/benefit consideration the ability to operate with wet vapor at the expander inlet clearly improves the economic viability of the system; since the power output is increased while the heat exchanger surface (and cost) is lowered. Thus CAPEX ($/kW) is lowered.

Lubrication System

The admission of wet vapor to the expander can also be used to simplify and reduce the cost of the ORC lubrication system by dissolving or otherwise dispersing approximately 1-2% oil by mass in the working fluid inventory, as described by Smith et al (8, 9). The oil is then transported by the working fluid, in the liquid phase through the boiler to enter the expander, where it will lubricate the rotors as the liquid working fluid evaporates during the expansion process. Also, some of the oil enriched, pressurized working fluid leaving the feed pump,

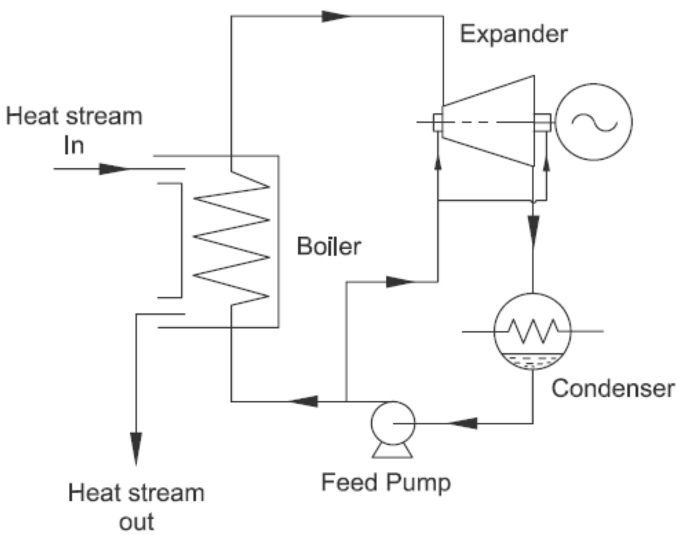

Figure 6. ORC with process lubricated bearings.

prior to entering the boiler, can be distributed to the bearings, which are of the rolling element type, where frictional heating will evaporate it off to leave sufficient oil to lubricate them. One possible arrangement for this is shown in Figure 6.

This entirely eliminates the need for the separator, oil circulating pump and heat exchanger, needed for an oil flooded expander lubrication system or the even more costly oil storage tank, pump, heat exchanger, shaft seals and timing gear needed for oil free machines. Moreover, tests (8) revealed that, after a few hours of running, the bearing housings were found to be full of oil even when the initial fraction of oil dissolved in the refrigerants, was hardly detectable.

Mass produced screw compressors modified to act as expanders and lubricated in this manner, are roughly 80% cheaper than purpose designed turbines performing the same function. An even more significant improvement in the CAPEX than that achieved from the gain in power output attributed to wet vapor admission is thereby achieved.

Boiler Design and Control

It is common practice, in a vapor power plant, to divide the boiler into two sections. In the feed heating first stage, the liquid is heated up to its boiling point. Evaporation follows in the second stage. At the evaporator exit, the vapor so formed is separated from any liquid and passes to the expander, while the residual liquid is returned to the evaporator inlet. Both sections are normally of the shell and tube type. Separation usually takes place in an external vessel, as shown in Figure 7, but it can also be within the main shell of the boiler casing. The normal means of ensuring that only dry vapor flows to the expander is by means of a float control valve, the rise and fall of which generates a signal either to open or close a bypass valve that recirculates some of the liquid, leaving the feed pump back to the pump inlet, or to vary the feed pump speed.

When the vapor leaving the boiler is wet, a simpler arrangement would be to use a single pass system, in which cold liquid enters at one end and leaves with the desired degree of wetness at the other end, without any external recirculation. This would then eliminate the need for a separator and the boiler could be of the less expensive plate fin type. A further reduction in CAPEX is thereby achieved but an alternative boiler control system is then needed to control the exit dryness fraction.

A number of arrangements for this are possible (10), one of which is shown in Figure 8. This is based on throttling a small sample flow from the main boiler delivery pipe directly to the condenser. Since the working fluid is never required to leave the expander as wet vapor, it follows that the throttled sample must always be superheated when it attains the condenser pressure. In that case, the exit enthalpy from the boiler can be estimated only from pressure and temperature

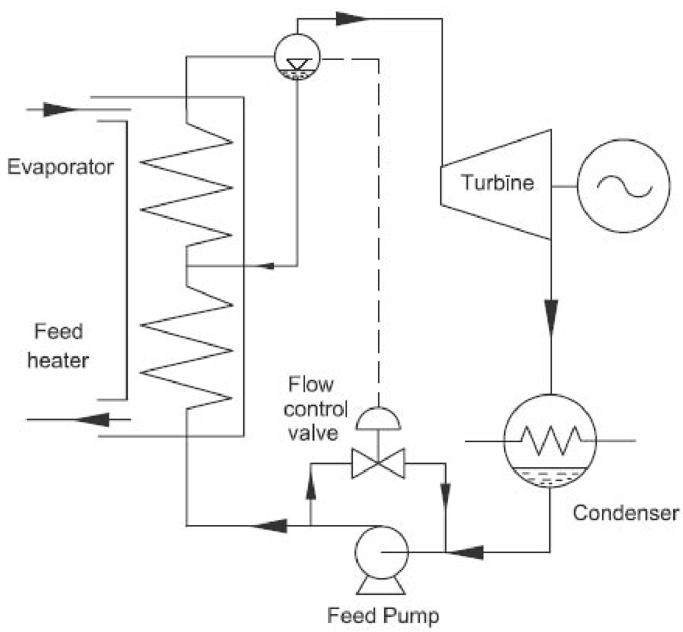

Figure 7. ORC flow control system.

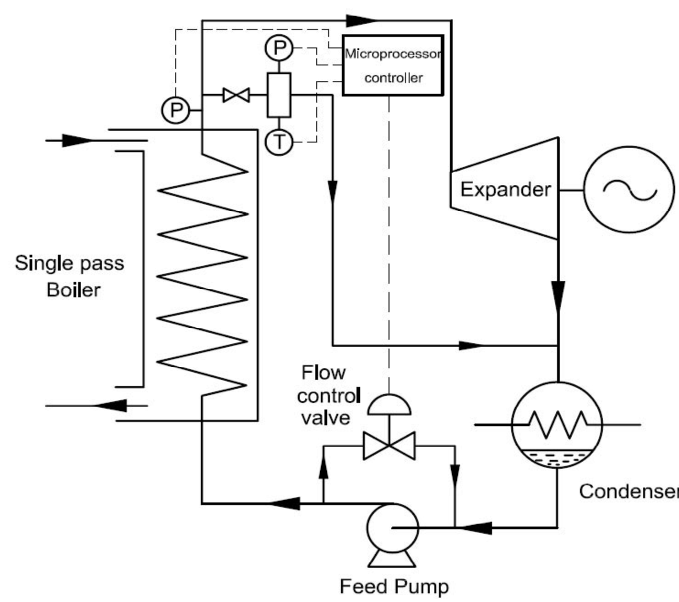

Figure 8. Microprocessor Control System.

measurements of the throttled fluid. Thus, given the boiler discharge pressure, its exit dryness can be estimated from this value. Accordingly, using a microprocessor to generate the signal from the pressure and temperature inputs, the required boiler exit dryness fraction can be maintained by recirculating some of the feed pump delivery or by varying the feed pump speed to achieve the right degree of superheat in the throttled fluid. This, again, results in further reduction in CAPEX.

Design Study

To determine the suitability of such a system for small outputs, a typical case of heat recovery from geothermal brine at 100°C was considered. The boiler pinch point temperature difference was taken as 5°C. A water cooled system was assumed with water entering the condenser at 21°C (65°F) and incurring a 9°C temperature rise in the condenser in which the pinch point temperature difference was taken as 4°C. The working fluid was assumed to be refrigerant 245fa.

The results of the study showed that the ORC system would operate with an optimum output when the fluid enters the expander 86% dry at a temperature of 75°C and leaves it as wet vapor, 98% dry, at 34°C. A standard 159 mm diameter oil injected screw compressor, modified to act as an expander and directly coupled to a generator rotating at 3600 rpm would then develop a gross electrical power output of 25 kWe with an expander adiabatic efficiency of 78%.

Demonstration Unit

Although most of the novel features, described in this paper, have been validated experimentally on different occasions, they were never included, together, in a single unit. An existing demonstration unit was therefore modified to incorporate all of them. In this case it was required to produce net output from a flow of hot water at only 90°C with a cooling water supply at 15°C, using R124 as the working fluid. For this purpose, a standard oil flooded air compressor, with "N" profile rotors, of which the male rotor diameter was 102 mm, was adapted to operate in the expander mode. For flexibility of testing, the machine was coupled to a generator, rotating at 1800 rpm, using a vee belt drive. A photograph of the demonstration unit is shown in Figure 9. Preliminary tests showed that the machine ran smoothly and quietly with process fluid lubrication, as described. After strip examination, oil was found to have accumulated in the bearing housings and, due to wet vapor admission, traces of oil were found at the high pressure inlet port with the rotors uniformly covered with a film of oil. Thus, the predicted behaviour of the lubrication system was fully realized. Initial performance tests then showed that the expander generated approximately 22 kW of shaft power with an adiabatic efficiency of 74%, when admitting vapor with an inlet dryness fraction of 0.95. This agreed well with expander performance predictions obtained from simulation studies, using proprietary screw machine design software, developed over a period of more than 25 years (11).

Figure 9. ORC Demonstration Unit.

Cost Estimates

Based on the results obtained, a highly detailed costing study was carried out for a 50kW unit, in which all component costs were obtained from vendors' price quotes. Using scaling methods normally applied to chemical plant design, it was concluded that units in the 25 – 500kW output range can be built and installed for a total cost in the range of $1,500 to $2,000/kWe, depending on size, if water cooled, and about $2,500/kWe when air cooled. These figures were compared with costs quoted for a 200 kW turbine driven unit, developed from a reversed air conditioning system (1,2) and it was concluded that as a result of the cost savings possible, using modified mass produced air compressors, together with the other advantages described in this paper, equivalent screw units would be approximately 30% cheaper.

Conclusions

By taking full advantage of the potential of screw machines as expanders, it has been shown that ORC units can be constructed for power recovery from low temperature heat sources with outputs of as little as 20 kWe, at an economically viable cost and these should be substantially cheaper than turbine driven units at outputs of up to approximately 500kWe.

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