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COST EFFECTIVE SMALL SCALE ORC SYSTEMS FOR POWER RECOVERY FROM LOW GRADE HEAT SOURCES

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ABSTRACT

The growing need to recover power from low grade heat sources, has led to a review of the possibilities for producing systems for cost effective power production at outputs as little as 20-50kWe. It is shown that by utilizing the full potential of screw expanders instead of turbines, it is possible to produce Organic Rankine Cycle (ORC) systems at these outputs, which can be installed for a cost in the range of \$1500 to \$2000 /kWe of net output. This low capacity cost combined with the ORC's fuel-free specification results in a very favorable value proposition.

INTRODUCTION

The idea of using organic working fluids, to replace steam, in power generation systems, has been traced back as far as 1823 [1] and the technology for power recovery from low grade heat sources, such as geothermal and waste heat streams, using Organic Rankine Cycle (ORC) systems, is now well established. However, most systems of this type have only been cost effective for power outputs of the order of 1 MWe or more. More recently, Brasz [2] 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 is claimed that units of as low as 200kWe power output can be built and installed cost effectively. Since the thermodynamic efficiencies of these systems are generally only of the order of 10%, this implies that at least 2 MW of heat must be recoverable for such systems to be worthwhile. Low grade heat sources of this magnitude, which cannot be used for heating purposes more cost effectively, are relatively rare.

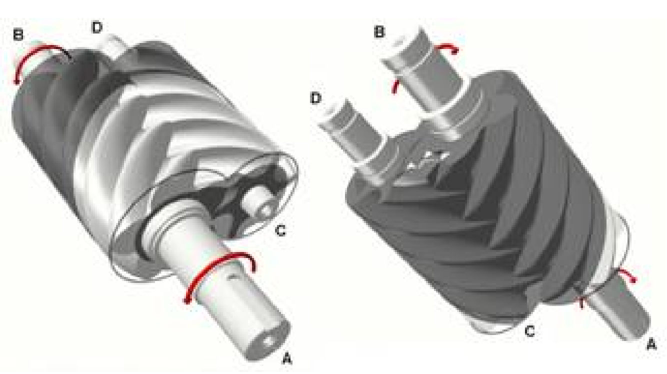
Rising fuel prices and the wide availability of heat sources of the order of only 250 kW, mainly as waste heat from small IC engine generator sets, have opened a considerable market for units of the order of 20-50 kWe output, if these could be built and installed for an economic price. At such outputs, the use of turbines as expanders has many disadvantages, since they rotate at very high speeds, and hence require high ratio reduction gearboxes as well as relatively expensive 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.

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. They are essentially comprised of a pair of meshing helical rotors, contained in a casing which surrounds them with clearances of the order of 50 microns (0.002”). The spaces between the lobes and the casing, together, form a series of working chambers, as shown in Fig 1, by means of views from the opposite ends and sides of the machine. The dark shaded portions show the enclosed region where the rotors are surrounded by the casing, 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. As they rotate, the volume trapped in the passages between the rotors and the casing changes. If fluid is admitted into this space, at one end of the rotors, its volume will either increase or decrease, depending only on the direction of rotation, until it is finally expelled from the opposite side of the rotors, at the other end. Power is transferred between the fluid and the rotor shafts from torque created by forces on the rotor surfaces due to the pressure, which changes with the fluid volume. Thus, unlike the mode of power transmission in turbo machinery, only a relatively small portion of the power transferred is due to dynamic effects associated with fluid motion. The presence of liquid in the machine, together with the vapor or gas being compressed or expanded, therefore, has little effect on its mode of operation or efficiency. Also, it should be noted that, unlike turbo machines, in which the blading and stage requirements are completely different for expansion and compression, a screw compressor only requires alteration to the high pressure port, to obtain the optimum built-in volume ratio and reversal of direction of rotation of the rotors, to operate equally well as an expander.



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

Fig 1: Screw Expander Main Components

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 Fig 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.

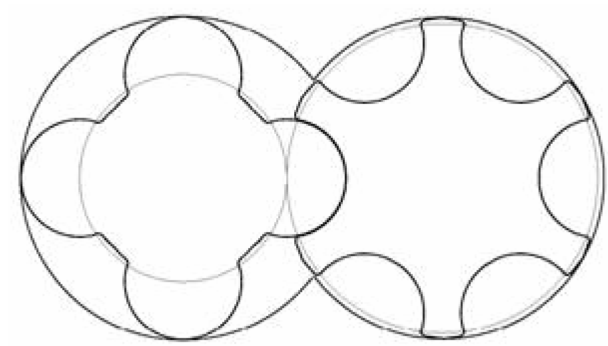


Fig 2: Symmetric circular profile

The female rotor profile then mirrors this with an equivalent set of circles

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. However, the symmetric profile performs surprisingly well in low pressure compressor applications.

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], developed at City University, 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 up to 10% over other profiles in current use have been recorded. A typical pair of “N” rotor profiles is shown in Fig 3, but in fact, it describes a whole family of shapes, the proportions of which can vary according to the machine duty.

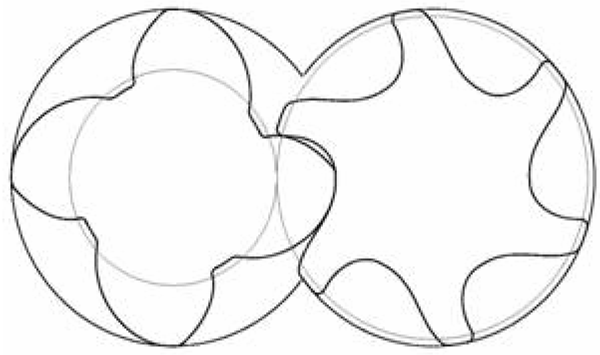


Fig 3: Rack generated 'N' Profile

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 generator, without an intermediate reduction gearbox. This, not only saves in cost but also gives them an approximately 5% 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. However, efficiency predictions, based on computer simulations and backed by experimental data, indicate that even at these rather unfavorable conditions, adiabatic shaft efficiencies of the order of 70% or more are possible.

When applied to the recovery of power from IC engine exhausts, further efficiency gains and cost savings are possible by linking the screw expander to the engine generator through a belt drive, in the same manner as the main engine generator unit. Under these circumstances, the rotational speeds can be increased and adiabatic shaft efficiencies in excess of 70% are possible even from the smallest units.

In the absence of a reduction gearbox, these values compare favorably with turbine efficiencies at these low outputs.

The limiting size of screw expander rotors that can be produced at an economic cost is of the order of 350-400 mm diameter. Depending on the working fluid selected and the temperature range of operation, such machines, directly coupled to 2-pole generators, have predicted maximum power outputs of the order of 1 MW with adiabatic shaft efficiencies in excess of 80%.

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 Fig 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.

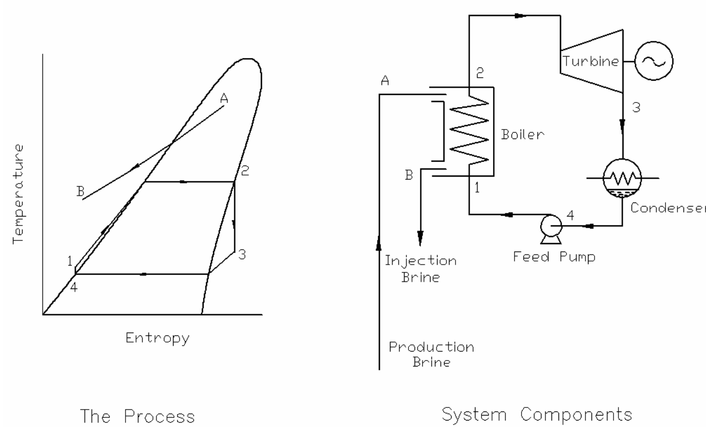


Fig 4: ORC System with Turbine as Expander

However, as already mentioned, 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, 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 appear in any of the forms shown in Fig 5, depending on the dryness fraction of the working fluid at the expander inlet.

As can be seen, the effect of increasing the wetness of the working fluid at the expander inlet is to reduce the superheat at the expander exit and, if required, even permit the working fluid to leave the expander as wet vapor. 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, both these effects tend to increase the power output. However, in the absence 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.

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 then performed taking account of internal pressure losses and pump, fan, expander, generator and motor efficiencies. The resulting power 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.

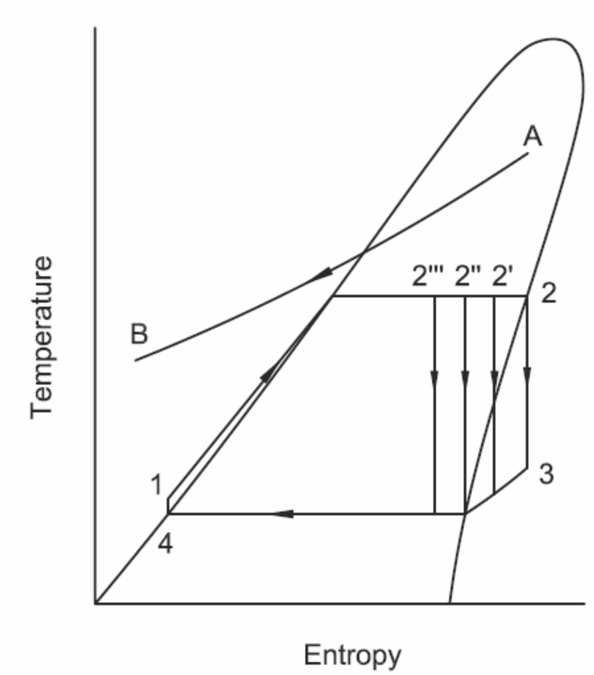


Fig 5: The effect of varying expander inlet dryness fraction

Since desuperheating of 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 increased 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 and, in a system where no fuel is consumed, the all-in cost of generation depends almost entirely on CAPEX.

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 up to approximately 5% oil by mass in the working fluid inventory (8). 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, prior to entering the boiler, can be distributed to the bearings where frictional heating will evaporate it off to leave sufficient oil to lubricate them. One possible arrangement for this is shown in Fig 6.

This arrangement 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.

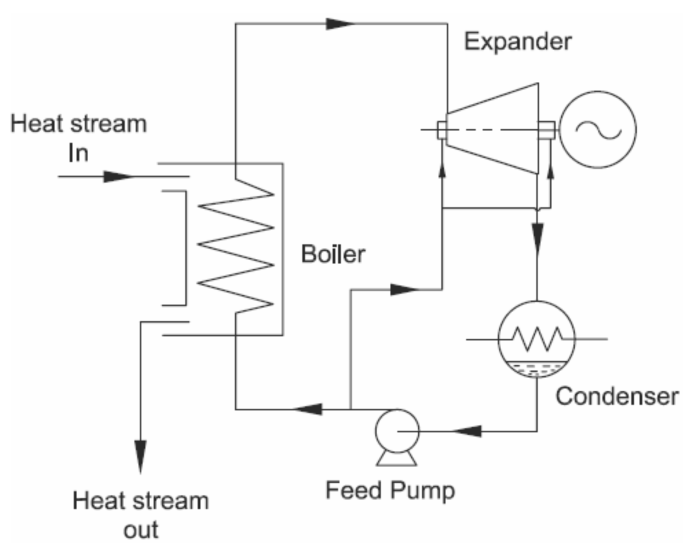


Fig 6: ORC with process lubricated bearings

This simplification of the lubrication system reduces the total cost of the expander by roughly 80% and thereby results in an even more significant improvement in the CAPEX than that achieved from the gain in power output attributed to wet vapor admission.

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 Fig 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.

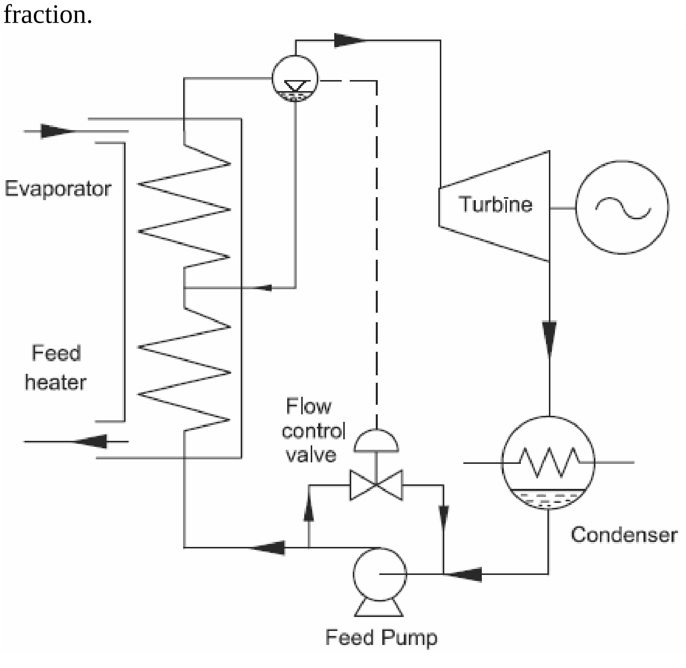


Fig 7: ORC flow control system

A number of arrangements for this are possible [9], one of which is shown in Fig 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 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.

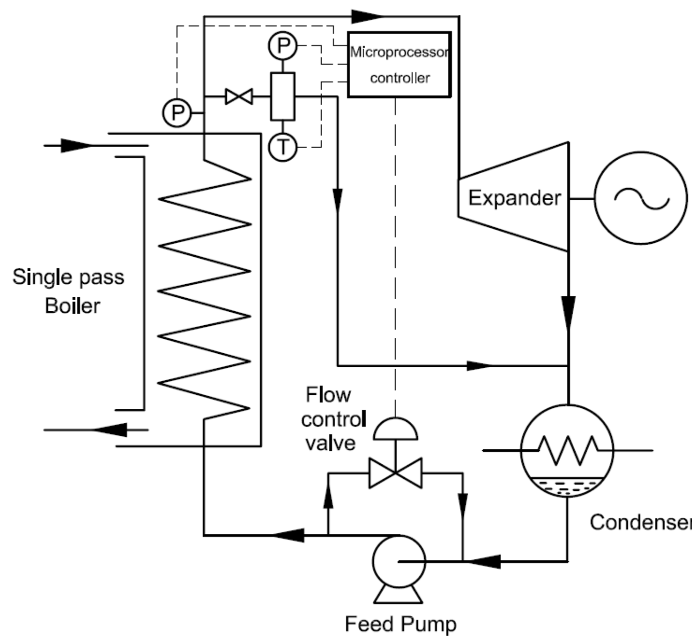


Fig 8: Microprocessor Control System

DESIGN STUDY

To determine the suitability of such a system for small outputs, a typical case of waste heat recovery from a 250kW stationary gas engine generator set was considered. In this case, the coolant, taken as a water/glycol mixture, was assumed to flow in a closed circuit through the oil cooler, the engine jacket and an exhaust gas heat exchanger in series. The recovered heat is then transferred to the ORC boiler. Using a typical engine manufacturer's data [10], it was determined that in cooling the exhaust gases to a minimum of 180°C, a total of 250 kW of heat was recoverable, with the overall coolant temperature changing from 103.6°C (218.5°F) to 90.9°C (195.6°F). The boiler pinch point temperature difference was taken as 5°C. It was assumed that a recirculating water cooling system was used with cooling water entering the condenser at 18.3°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. It was assumed in this case that the expander adiabatic efficiency would be 70%.

The results of the study showed that the ORC system would operate with an optimum output when the fluid enters the expander 88% dry at a temperature of 90.4^{o}C and leaves it as dry saturated vapour at 31.3^{o}C. The screw expander would then develop a gross shaft output of 24kW.

Using a well developed screw expander simulation program [11,12] it was estimated that a standard air compressor, with a male rotor diameter of 102 mm would meet this requirement, when rotating at 4,600 rpm at which condition, the estimated efficiency was 71%. Assuming that the expander is coupled to the main engine generator by a simple belt drive, it was estimated that the net increase in electrical power output from the entire system would be 21kWe. This would thus boost the total power output by approximately, 8.5%.

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. A demonstration unit was therefore designed, which incorporated all of them. The application was to produce 30kWe net output from a flow of hot water at only 90^{o}C with a cooling water supply at 15^{o}C, using R124 as the working fluid. For this purpose, an available oil flooded air compressor, with “N” profile rotors of which the male rotor diameter was 128 mm was adapted to operate in the expander mode. The choice, in this case, was limited by the need to use a standard size machine and couple it directly to a generator rotating at 1800 rpm. A photograph of the demonstration unit is shown in Fig 9. Preliminary test results have shown that the machine runs smoothly and quietly with process fluid lubrication of the bearings and rotors, as described, and that expander efficiencies have been obtained that are in fair agreement with those predicted. Further tests are proceeding and will be reported in a later paper.

Fig 9: ORC Demonstration Unit

Based on the results obtained so far from this machine, it is estimated that units of this size can be built and installed for a total cost in the range of $1,500 to $2,000/kWe, depending on size.

CONCLUSIONS

Contrary to previously held views, it has been shown that by taking full advantage of the potential of screw machines as expanders, it is possible to produce ORC units for power recovery from low temperature heat sources with outputs of as little as 20 kW, at an economically viable cost.

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