Screw Expanders Increase Output and Decrease the Cost of Geothermal Binary Power Plant Systems

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1. ABSTRACT

A study has been carried out on the feasibility of using screw expanders, as an alternative to turbines, in three types of binary systems, suitable for recovering power from liquid dominated brines. The maximum power that one machine can generate is of the order of 1-3 MW per unit, depending on the type of system, and the brine and cooling conditions. They are shown to be roughly as efficient as turbines, when expanding dry vapour, and they can be coupled directly to a generator, without an intermediate gearbox. They have similar characteristics as two-phase expanders, when their efficiencies are not much reduced. This latter feature enables improvements to be made in the conversion of heat to power, of the order of 30% over conventional ORC systems, from brines in the 100^o – 140^oC temperature range. The costs of these machines are very competitive and it is shown that in a cycle designed for power recovery from brines in the 170^o- 210^oC range, the cost per unit output of the first expander stage may be less than 50% of that of a radial inflow turbine-generator set.

2. INTRODUCTION

To the geothermal community, screw expanders are associated with low efficiencies and unreliable operation. This comes from unsuccessful attempts to use them for direct expansion of liquid dominated geothermal brines, during the late nineteen seventies and early eighties [1-4]. Since then, much work has been done on a more critical appraisal of their role, and, over a period of nearly 25 years, approximately US$5 million of research has been carried out by the authors in developing these machines, both as compressors and expanders, for a wide range of uses. As a result, their mode of operation is now far better understood and the applications for which they are best suited can be defined more accurately.

Early analytical work on screw expanders was carried out by Margolis [5] and Taniguchi et al [6] but was taken much further by the authors, who reported on how they can be designed for two-phase expansion, to the GRC in 2001[7]. Although it was shown that such machines could be made to operate efficiently and were relatively cheap, the range of brine temperatures and power outputs for which they are most suited, in geothermal applications, was not defined. Originally, the authors conceived their use as a means of improving power recovery from HDR resources at brine temperatures in the region of 200^oC [8,9]. However, as will be shown, more recent studies, led the authors to reassess what their role could be in geothermal power system applications.

3. ESSENTIAL FEATURES OF SCREW EXPANDERS

3.1 Principle of Operation

As shown in Fig 1, a screw expander is 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”). As they rotate, the volume trapped 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 by pressure on the rotors, which changes with the fluid volume. Thus, unlike the mode of power transmission in turbomachinery, 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 vapour or gas being compressed or expanded, therefore, has little effect on its mode of operation or efficiency.
The main requirements for efficient operation are:

i) Maximum flow area in the space formed between the lobes and the casing.
ii) Minimum leakage through clearances in the machine.
iii) Correct choice of built in volume ratio of expansion.
iv) Correct selection of tip speed.

3.1.1 Flow Area: For a given rotor diameter, this is determined by the rotor profile. As already described [7], the authors have developed a profile, which is more efficient than any other. This profile has many other advantages, including minimum leakage path, low internal friction losses between the rotors and nearly pure rolling action between the rotors. Consequently, during the past few years, it has been adopted by many screw compressor manufacturers, including major companies in the USA.

3.1.2 Minimum Leakage: Ideally, since clearance areas increase with the square of the rotor diameter while flow rate increases with the cube of the diameter, leakage, as a percentage of the total flow, should decrease as the rotor diameter increases. This is not true in practice because minimum manufacturing tolerances are fixed. Thus for very small diameters, leakage will be unacceptably high. In the 70- 450 mm diameter range, leakage rates are favourable, using available machine tools designed specifically for rotor manufacture. For rotor diameters greater than about 450 mm, the machining operation becomes complex. The percentage of flow as leakage then may even increase, while the manufacturing costs rise rapidly.

3.1.3 Built in Volume Ratio: This describes the ratio of the maximum swept volume of fluid in the expander immediately before the discharge port is exposed, to the volume of fluid contained in the expander at the closing of the inlet port. It is directly comparable to the compression ratio of an IC engine and is determined solely by the size and shape of the high pressure port. It can be altered very easily for different operational requirements, at little cost. In the case of an expander, a low built in volume ratio is desirable because it maximises the input flow of fluid, before the high pressure port is cut off. The greater the flow rate, the less are the adverse effects of leakage. However, too low a built in volume ratio leads to under expansion of the working fluid and hence expulsion of the fluid at too high a pressure. Recoverable expansive work is thus lost and the efficiency of the expander falls. When admitting pure liquids, the volume ratio of expansion is far greater than when admitting dry vapours or gases. However, as shown by the authors [7.9], due to the high density of liquids, a substantial amount of expansion occurs in the filling process, as the fluid enters the high pressure port. This leads to the overall volume expansion ratio of the fluid being many times greater than the built in volume ratio. An appreciation of this is the key to obtaining efficient two-phase expansion in these machines. The main reason for the failure of the screw expander in total flow applications was that the volume ratio needed for efficient expansion of water is too large to be practically attainable.

3.1.4 Tip Speed: Due to the clearances needed to enable it to rotate, fluid will flow through a screw expander even when it is stationary. This flow rate is determined by the pressure difference between the inlet and exit and is almost independent of speed. The ideal induced flow rate increases linearly with speed but rotational friction losses increase with the square of the speed and their magnitude depends on the nature of the working fluid. An optimum condition is reached when reduced leakage losses, induced by raising the tip speed, are equalled by the resulting increased friction losses. The values of this optimum tip speed depend on the working fluid. Thus, if there are large quantities of oil in the working fluid, it is of the order of 30 m/s whereas when the fluid is a dry vapour or gas it can exceed 120 m/s.

3.2 Expander Size, Performance and Cost

Screw compressors are, today, used for a wide range of applications, but mainly for air and refrigerant compression, where they operate in the oil flooded mode [7]. As a result of much development, they can now attain adiabatic efficiencies of up to 90%, which is comparable to that of the best aerodynamic compressors. However, unlike turbomachines, they are manufactured in large numbers and, oil flooded machines are relatively cheap.

Their efficiencies as expanders are less well proven. However, it should be noted that, unlike turbomachines, 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. On this basis, the authors adapted an oil injected air compressor that they had designed, with their patented rotor profiles, to operate as a two-phase expander. This machine, which contained other unusual features, such as process fluid lubrication of the rotors and bearings, has already been described [7]. However, since then, its range of operation was improved when flow losses through its test circuit were reduced. More recent results, obtained in 2002, are shown in Fig 2. In this case, the working fluid was R113, which has pressure-volume characteristics close to those of n-Pentane at the same temperatures. At all test points, the fluid entered the machine as slightly subcooled liquid and left it in the two-phase condition. The graphs confirm the high efficiency predictions claimed by the authors in earlier publications [7-9]. This machine, as a prototype, cost US$6,000. However, a larger production version of it, with a 159 mm diameter male rotor, is currently sold as an air compressor for only US$1150.

In support of their experimental work, the authors have developed a number of advanced software packages, based on the numerical solution of the equations of conservation of mass, energy and momentum of fluid flow through both screw expanders and compressors, when operating both on single and two-phase fluids. These have been extensively modified by comparison with real machine test results to give reliable predictions for both modes of operation.

Enquiries over the past eighteen months on the suitability of the TFC system for lower temperature geothermal brines and whether screw expanders were viable alternatives to turbines in small ORC power plant led the authors to examine what the likely limits of use could be for machines of this type in binary power systems. A number of assumptions were made in this study, which can be summarised as follows:

The entire expansion process must be carried out in a single stage.

In a TFC system, the heat exchangers are larger and therefore more costly than in an ORC system. For use in ORC systems, the screw expander must be no more expensive than a turbine. Thus, in both applications the cost must be minimised.

Contrary to screw machines, turbines operate with full admission around their circumference and at higher tip speeds. Essentially, they should therefore be smaller and cheaper. However, they often require a gearbox to couple them to the generator and they run in journal bearings, which normally require an oil lubrication system of some complexity.

To make a screw expander cost competitive either in a TFC or ORC system, it would therefore have to rotate at either 1500 or 3000 rpm (1800 or 3600 in the USA) in order to eliminate a gearbox coupling to the generator and it would need to be process fluid lubricated to eliminate the cost of an oil lubrication system.

Performance predictions have shown that limiting tip speeds for best efficiency are in the region of 60 m/s. On this basis and that of manufacturing cost, their maximum size is therefore limited to approximately 420 mm male rotor diameter. It is also essential that such machines should be adaptable from standard oil injected air compressors in order to gain the cost advantages of bulk production.

With these considerations, in mind, the authors consulted a company that produces oil injected screw compressors with the authors' rotor profiles. Their products are currently the most efficient of that type produced. Based on discussions with them, the following are first estimates of the cost of a basic screw expander, suitable for either TFC or ORC applications, when mounted on a base plate and fitted with a coupling but minus the electrical generator. The latter has been deliberately omitted because, as will be shown, the output from such machines can vary widely depending on brine and condensing temperatures.

Rotor Diameter mm	Approx Unit Cost US$
285	20,000
355	30,000
420	45,000

The authors then made estimates of what power was recoverable from such machine sizes, when used as expanders rotating at 3000 rpm in both ORC and TFC cycles.

4. PERFORMANCE ESTIMATION

This was carried out first by determining the expander inlet pressures and temperatures corresponding to optimum power recovery, given only the initial brine temperature and the condensing temperature. The calculations were based on a set of subroutines for estimating local fluid thermodynamic and transport properties at initially assumed cycle state points. The values of these points are then changed by means of a multi variable minimisation procedure to obtain the best result. These calculations were repeated for assumed brine temperatures of 100^{o}C to 140^{o}C with condensing temperatures varying from 20^{o}C to 40^{o}C for each assumed brine temperature. To obtain what outputs could then be obtained from screw expanders under these conditions, the derived optimum expander inlet and exit conditions for each case were input to the previously described program for estimating their power output and efficiency, when operating at either 1500 or 3000 rpm. Two sizes of machine were assumed. The first was a 285 mm male rotor diameter. This was taken because it is the largest machine currently produced with the authors' profiles and hence can be readily modified for experimental validation, at low cost. The exercise was then repeated, assuming a 416 mm rotor diameter, as representative of an upper practical limit for machines of this type for use in geothermal power plant.

In addition, the authors considered the use of screw expanders of 416 mm male rotor diameter, as an alternative to a radial inflow turbine to carry out the two-phase expansion in the first stage of the higher temperature cycle, described by them at the 2004 GRC meeting 10]. For brevity, they have given this the eponymous name of a "Smith" cycle.

5. RESULTS OF ANALYSES

5.1 TFC System Results

For those unfamiliar with the concept, the TFC system layout is shown in Fig 3. Essentially, it has the same components as an ORC system but does not evaporate the working fluid and instead expands it, from the saturated or slightly subcooled liquid condition, as a two-phase mixture. Its main advantages are that by close temperature matching between the brine and the working fluid, it recovers more heat and attains a higher working fluid temperature than an ORC system. By this means, provided that the expander is efficient, it will produce more power. The authors decided, that a minimum expander adiabatic efficiency of 75% is needed to make the gain in output sufficient to compensate for the larger heat exchanger costs associated with the greater heat recovery.

The working fluid selected for this study was Isobutane for all cases. Previous studies have shown that it is the most suitable for the temperature range considered, when account is taken of both optimum cycle efficiency and practicable expander design

The outputs and efficiencies obtained with a 285 mm male rotor diameter screw expander are given in Fig 4, while those for the 416 mm machine are given in Fig 5. In both cases, the power output graph shows the locus of conditions where the expander efficiency equals 75%. This is shown as the limit for an acceptable TFC expander and therefore sets boundaries to where this system may be considered as a viable alternative to other types of power plant.

5.2 ORC System Results

This system is too well known to require a diagram. In this case, based on previous more general studies, two working fluids were considered. R134a was selected as the best working fluid for brines in the 100^{o}-$120^{o}C$ temperature range and Isobutane for brines in the 120^{o}-$140^{o}C$ temperature range. The results for a 285 mm male rotor diameter are given in Fig 6, while those for 416 mm rotor are given for R134a in Fig 7 and for Isobutane in Fig 8.

5.3 "Smith" Cycle Results

In 2004 [10], the authors presented an alternative to the simple TFC system for recovery of power from higher temperature brines, as shown in Fig 9. This involves two expander stages with two-phase expansion in the first stage and dry vapour expansion in the second stage. A radial inflow turbine was proposed for the first stage, for which established manufacturers guaranteed an adiabatic efficiency of 75%. For brevity, the authors have called it a “Smith” cycle.

The case considered in [10] was for power recovery from brine at 190°C when flowing at 75 kg/s. It was shown that the optimum output was obtained with n-Pentane as the working fluid flowing at 116 kg/s from an initial condition of saturated liquid at 175°C. Expansion in the first stage reduced the temperature to 90°C. This was followed by dry vapour expansion to the condensing conditions. The total estimated power recovery was roughly 7.1 MWe gross, of which approximately one third was recovered in the two-phase expander and two thirds in the dry vapour expander.

As an exercise, the possibility of using screw expanders in place of the radial inflow turbine was considered for the first stage. In this case, a single 416 mm male rotor machine was found to be insufficient for the flow required. However, two such machines in parallel were found to be adequate for this purpose. The following summarises the results.

Turbine-Generator Set	**Screw Expander- Generator Set**
Single Radial Inflow	Two 420 mm male rotor Units
Gross Output (Guaranteed) 2160 kWe	Gross Output (Estimated) 2,300 kWe
Manufacturer’s Budget Cost US$1,150,000	Estimated Basic Cost of the Two Sets US$190,000

6. DISCUSSION

The entire development program on screw expanders, carried out by the authors since 1982, was initiated in order to find a means of making a practical TFC system for geothermal power recovery. The discovery that to make them expand, in two-phase with high efficiency, involved a far lower built in volume ratio than was previously thought, was established in 1994. However, the prices then quoted by established manufacturers of these machines, were so high that in 1995, the authors abandoned any attempts to develop them further for this purpose and instead concentrated on their use as small process expanders for refrigeration and air conditioning systems. The development of the authors’ own “N” profile rotors, the discovery that process lubrication of these machines was possible with them and, establishing within the last twelve months, the low cost at which the author’s licensees were able to produce them, together with a recent enquiry on their suitability for use in ORC systems, combined to stimulate the wider appraisal carried out in this study. It is, therefore, perhaps worth noting, that the favourable values of the limiting outputs, efficiencies and costs thus obtained, when these factors were all combined, were not anticipated by the authors when they began it.

6.1 TFC Expander

The outputs obtained in Figs 4 and 5 are fairly easily understood in that they form a regular progression, with efficiencies decreasing as the difference between the brine and condensing temperatures is increased. This is essentially due to the need for a higher built-in volume ratio to permit fuller expansion across increasing pressure differences and, associated with this, reduced fluid throughput and higher leakage losses.

The test results recorded in Fig 2 were obtained with a much smaller machine, running at lower tip speeds and with smaller pressure differences. These generally agree with power, efficiency and mass flow predictions made with the same program used for the present study and with published total flow results, as well as with earlier test results on other machines, obtained by the authors [9]. Hence the larger scale performance estimates are believed to be approximately correct and fairly certainly, define the worthwhile limits of use reasonably well. However, there is a definite need to conduct larger scale tests to validate them and, possibly, to adjust the predictive model. For this reason the results of the 285 mm rotor machine are given because it can be readily produced at a modest price. The larger 416 mm unit would require a new cutting tool for the rotor manufacture and this would put a premium on the first units manufactured.

As can be seen, a TFC system, operating with a screw expander, will not be suitable for all conditions given. However, the possibility of producing approximately 2 MW per unit from brine at 120^{o}C at a condensing temperature of 30^{o}C at an expander cost of approximately US$20/kW of shaft power output should be a strong incentive to investigate this further. At lower brine temperatures and higher condensing temperatures, the expander efficiency clearly approaches that of a dry vapour turbine and though the maximum unit shaft output then falls to only about 1.5 MW, the production of this at about US$30/kW makes the use of multiple units in parallel, for higher outputs, a distinct possibility.

6.2 ORC Expander

The predicted ability of a single screw expander to meet the needs of ORC plant operating with brine temperatures from 140^{o}C down to 100^{o}C, over the almost the entire range of condensing temperatures considered, with adiabatic efficiencies of 80% or more is highly encouraging. The reasons for its superiority over the TFC expander are twofold. Firstly, the maximum fluid temperatures and pressures in an ORC are much lower than those in a TFC operating between the same brine and condenser temperatures. Hence, the leakage rates are lower. Secondly, the volume ratio of expansion of a vapour is far smaller than that of a two-phase fluid. Hence much smaller built in volume ratios are required in the ORC expander and this has the effect of decreasing relative leakage even more.

It should also be noted that the use of R134a at lower brine temperatures has the advantage not only of greater power output, because of its higher density, but also greater cycle efficiency than would be obtainable with a turbine. This is because, unlike a turbine, expansion can take place through the vapour dome without an efficiency penalty or risk of blade erosion. Accordingly, R134a can be heated to temperatures nearer to its critical point when using a screw expander than is possible with a turbine and by this means the temperature match between the brine and the working fluid in the boiler is improved. As can clearly be seen, though, the maximum output per unit under these conditions is in the region of only 1-1.5MWe. It clearly has advantages over a turbine for units of small output.

In this connection, it should be appreciated that the matching of the working cycle to the brine to maximise the conversion of heat to power is as important as optimising the expander efficiency. As can be seen, at a brine temperature of 120^{o}C, an R134a expander can produce more power than one using isobutane. At that condition, the system conversion efficiencies are more or less the same for both fluids. It follows that R134a is probably the better choice. However, due to the low critical temperature of R134a, at higher brine temperatures, the cycle does not match up to the heat source so well and Isobutane then gives a higher cycle efficiency. Although the maximum output from a single isobutane expander is less than that from R134a, the low additional cost per unit output of a larger number of isobutane expanders needed for higher outputs makes this fluid the preferred one for higher brine temperatures.

As for the TFC system, results are given for the 285 mm machine. These are highly encouraging for smaller power output systems and, as for the TFC system, the low cost of this size of expander should encourage site developers and system manufacturers to participate in validating these predictions.

6.3 A Comparative Study between TFC and ORC Systems

A limitation to the use of the TFC system is that the brines from which it draws heat must be freer of silicates and salts than is permissible with an ORC system because they leave the primary heat exchanger at far lower temperatures. Because of this, it is possible to use a welded plate heat exchanger for a heater, rather than the traditional but more expensive shell and tube type. In general, over the range of brine and condensing temperatures considered in this study, a TFC system can generate some 20-30% more power than an ORC system. The cost penalty for this is then mainly that of the larger heat exchangers needed for the greater heat recovery and rejection. To counter this, it should be noted that the drilling cost is the same for both systems. A higher cost per unit output for the TFC does not, therefore, necessarily imply that the overall system cost per unit output, including the drilling, will be higher for the TFC system.

To get some idea of the relative merits of the two systems, consider the case of a low temperature brine, at 110^{o}C, supplying heat to a binary system, condensing at 35^{o}C. The optimum power obtainable from a 416 mm rotor expander rotating at 3000 rpm in a TFC system operating with Isobutane was determined for this case and then matched to the cycle analysis to determine what rate of brine flow is required to that output. The same rate of flow of brine was then considered to supply heat to an ORC system with a screw expander running at 3000 rpm with R134a as the working fluid. The mass flow and inlet and outlet temperature values of the brine and the working fluid passing through the heater were then sent to a plate heat exchanger manufacturer to obtain a price quote for both cases. It was also assumed that the cost of an electrical generator would be approximately US$40/kWe for both systems. No assumptions were made for the condenser and cooling system cost since this would depend on the temperature and nature of the coolant. The following result was then obtained

	Rotor Diam mm	Exp Effic percent	Gross Output Kw	Net Output kWe	Approx Exp-Gen Cost US$	Approx Heater Cost US$	Total Cost US$
TFC System	416	83	1452	925	100,000	350,000	450,000
ORC System	325	83	879	656	63,000	75,000	138,000

The additional 270 kWe of net output therefore costs some $310,000 for the larger heater and expander. There will also be an additional cost for the larger TFC condenser but this will only be some 30% more than that of the ORC condenser, since both will operate under the same temperature differential. If the drilling cost is also included in the total, it is almost certain that the TFC will be cheaper per unit output, while yielding a larger revenue.

6.4 Two-Phase Expanders in a "Smith" Cycle

The results of the study described in section 5.3 can only be described as spectacular. In fairness to the turbine manufacturer, more details of what is included in their package system are needed, especially regarding the controls supplied. There is little doubt therefore that a final figure for an equivalent package involving all the facilities contained in the turbo-generator set, included in a screw-generator system will result in a rather higher price than the $190,000 quoted. However, major parts of the turbo-generator system appear to be the gearbox and lubrication system and neither of these is required for the screw expanders. It is therefore hard to visualise that the additional cost of any extra components would cost more than, say $100,000 for each machine. On this basis, the total system cost would still be less than half of that of the turbine system.

7. CONCLUSIONS

This study has shown, to the authors' surprise, that the potential role of screw expanders in making geothermal power more efficient and cost effective, is much more significant than they had previously thought to be possible. This is especially the case for smaller systems, with unit power outputs of up to 3 MWe, at brine temperatures in the 100^{o} – 140^{o}C range. Used in the, still to be tried, TFC system, under the right conditions they can be more attractive than ORC system, while in the latter, improvements in both cost and efficiency can be made with them. Their low cost compared to radial inflow turbines, for two-phase expansion, also gives scope for improvement, when included in the authors' own Smith"cycle system, designed for power recovery from brines in the 170^{o}-210^{o}C temperature range.

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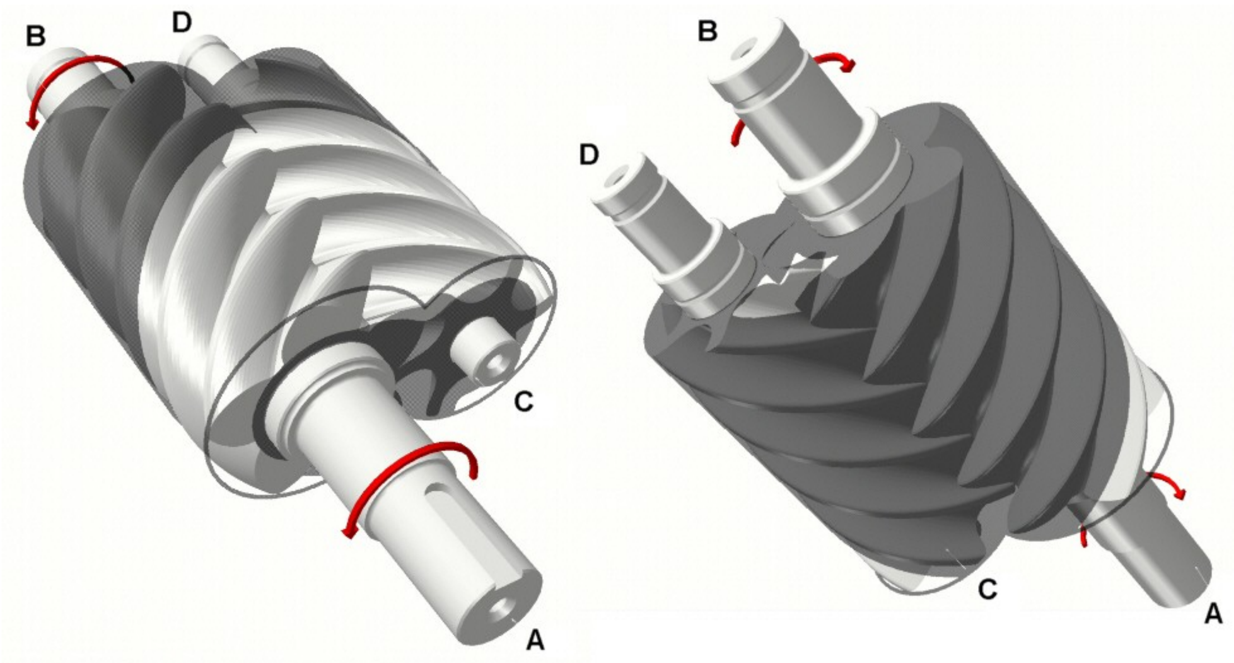
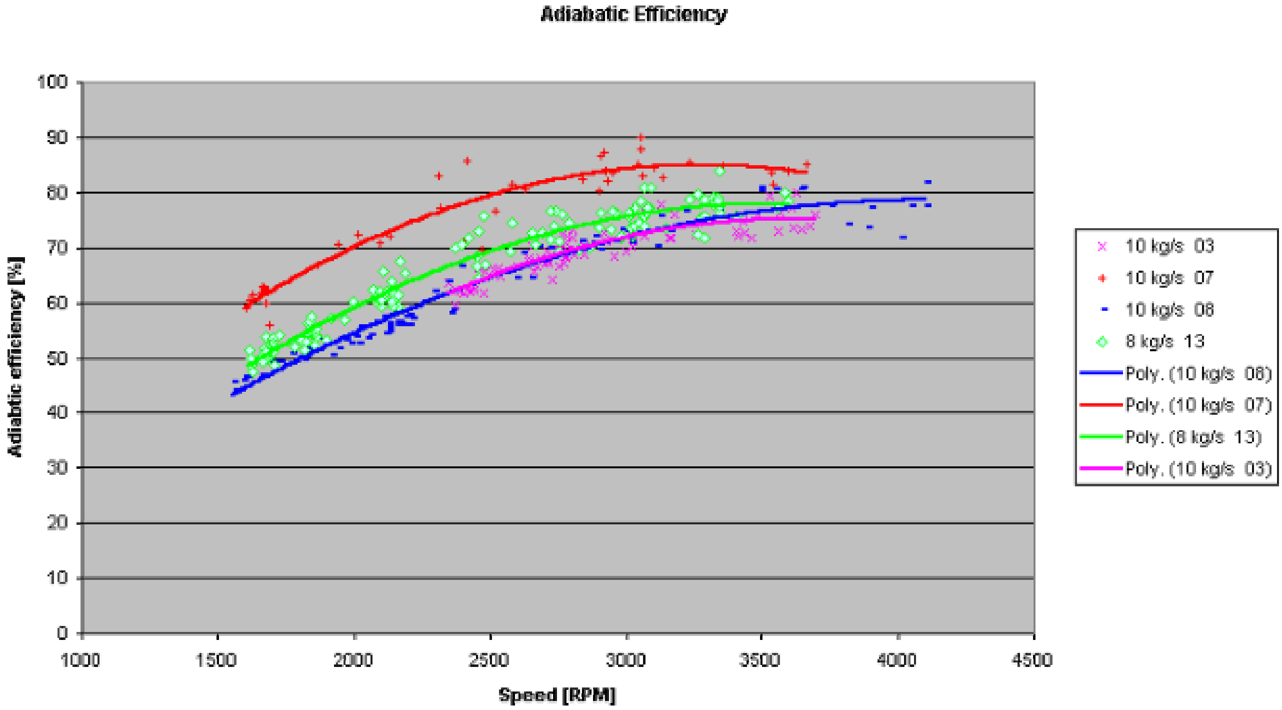


1a) View from Rear and Top 1b) View from Front and Bottom

Fig 1: Screw Expander Main Components



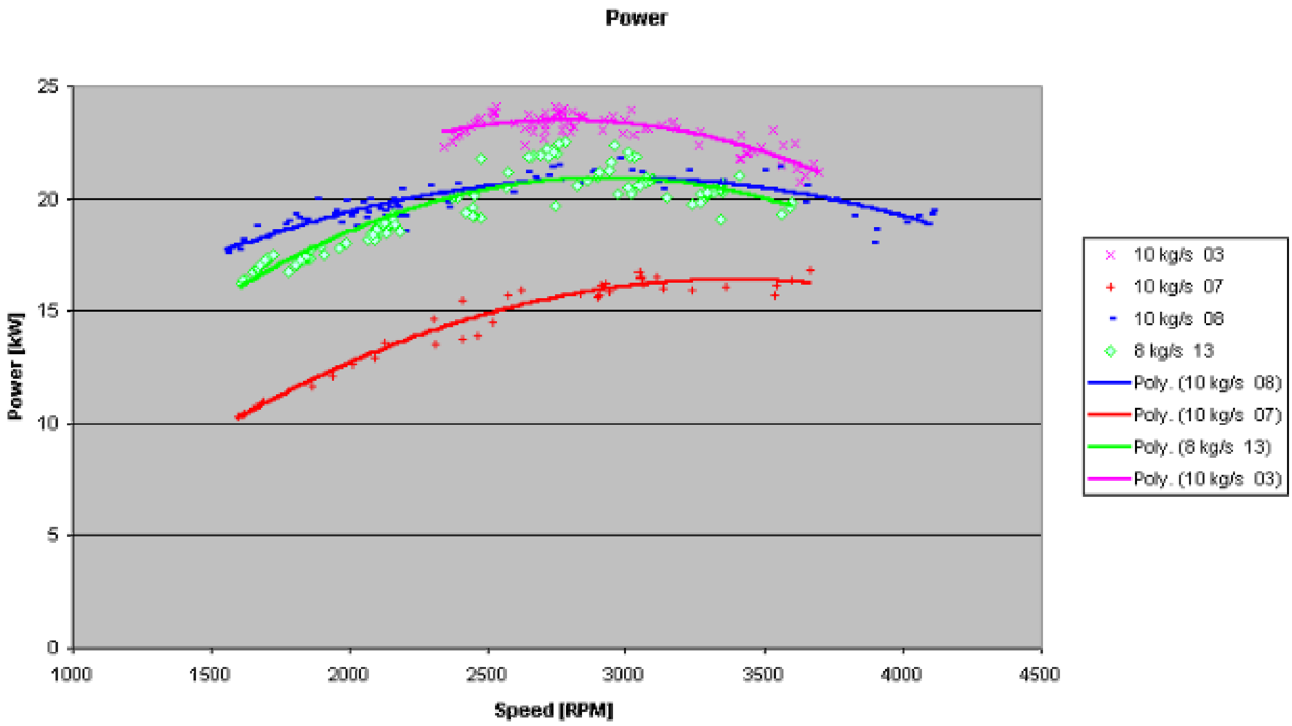
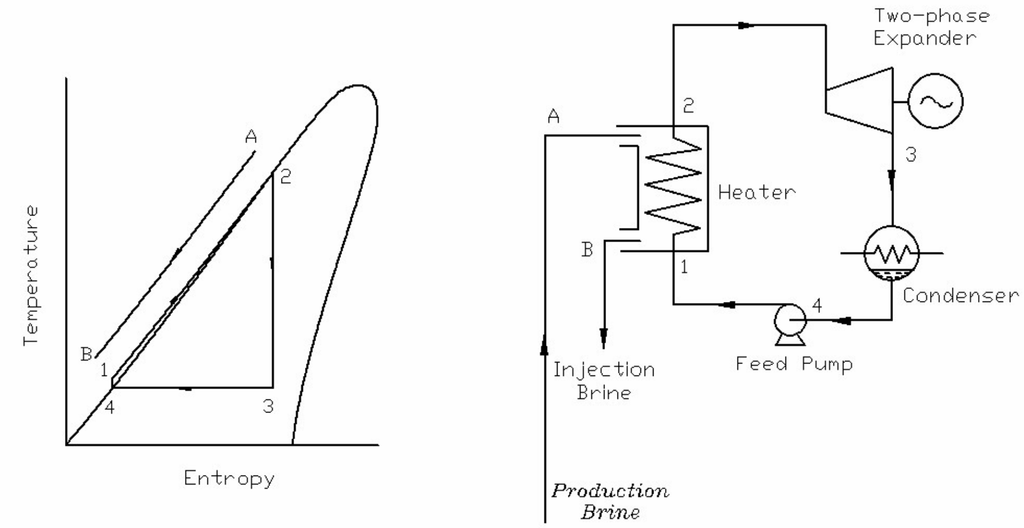


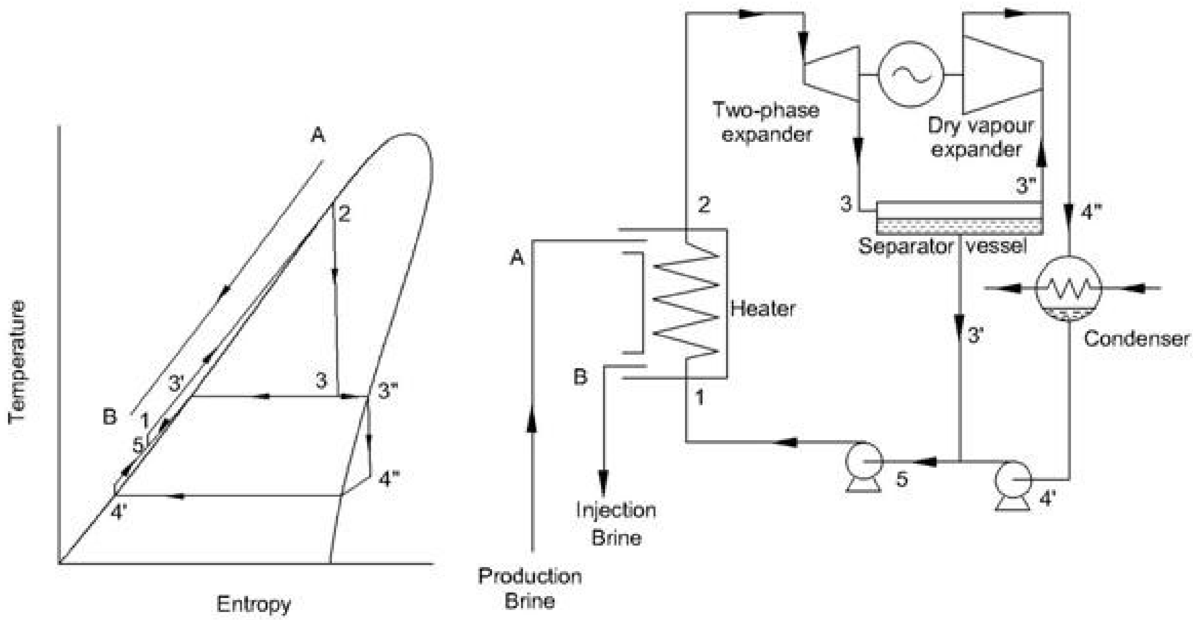
Figure 2: Power and Efficiency of 127 mm Two-Phase Screw Expander
Working Fluid: R113



The Process

System Components

Fig 3: TFC System



The Process

System Components

Fig 9 “Smith” Cycle

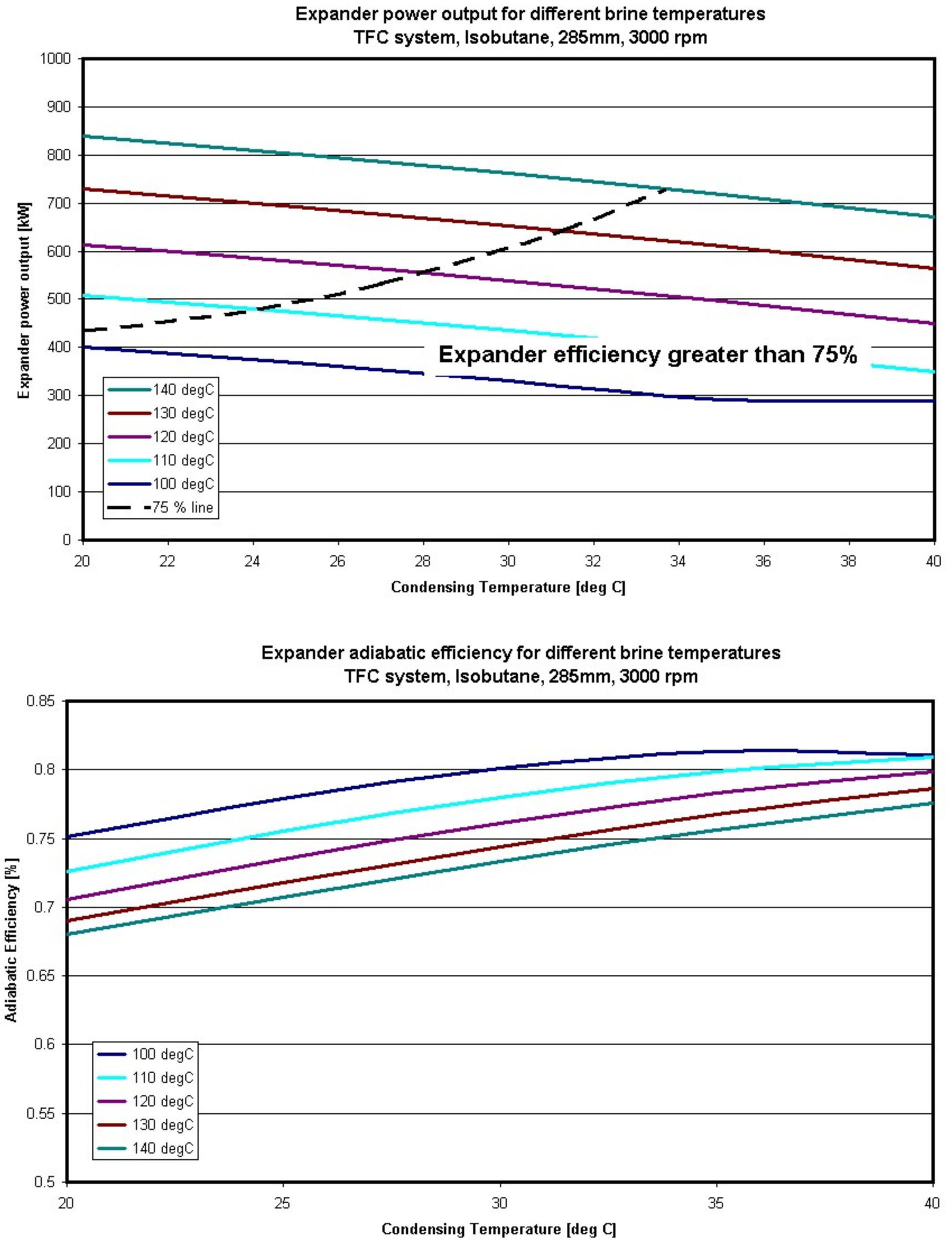


Fig 4: Power and Efficiency of 285 mm Screw Expander
TFC System Working Fluid Isobutane

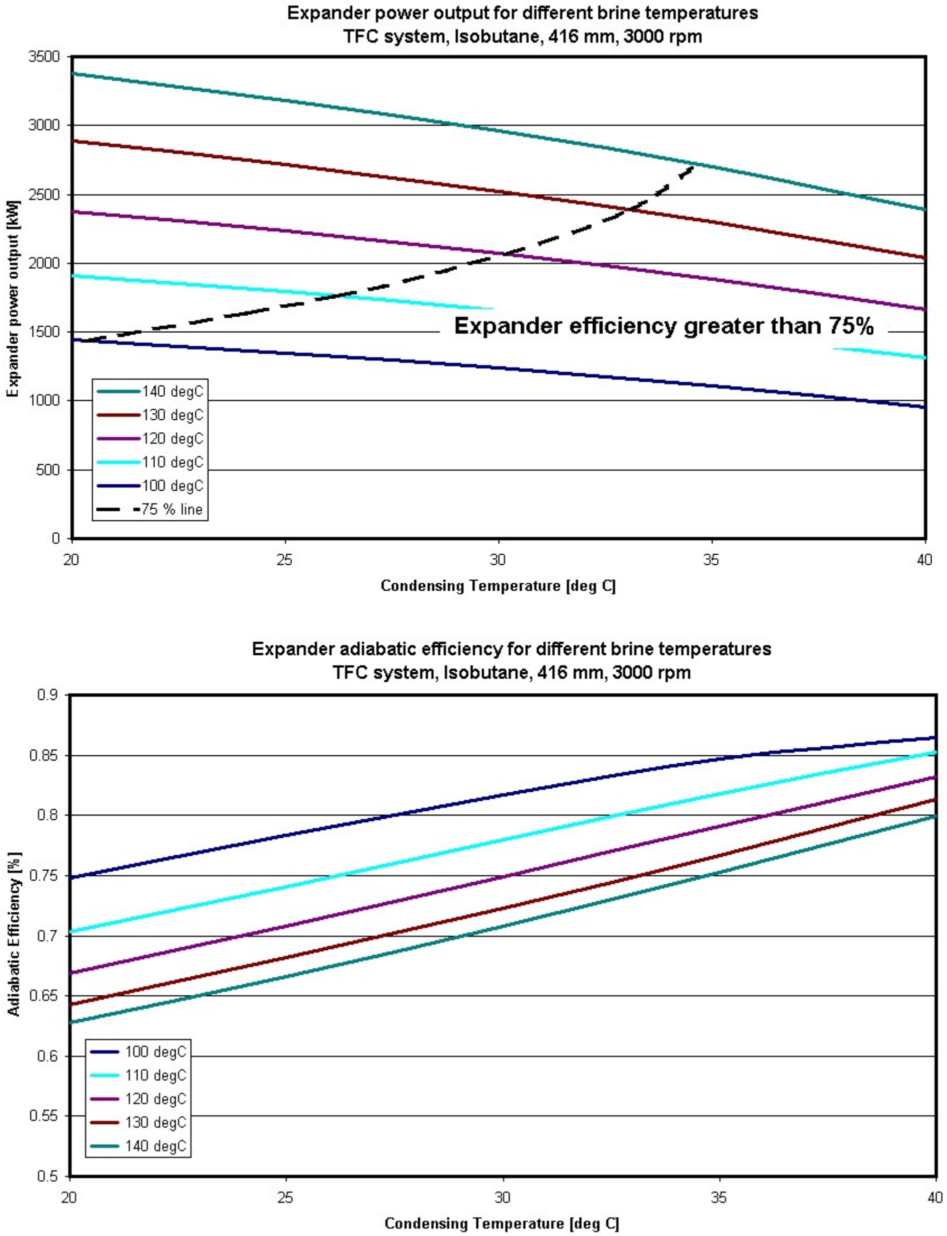


Fig 5: Power and Efficiency of 416 mm Screw Expander
TFC System Working Fluid Isobutane

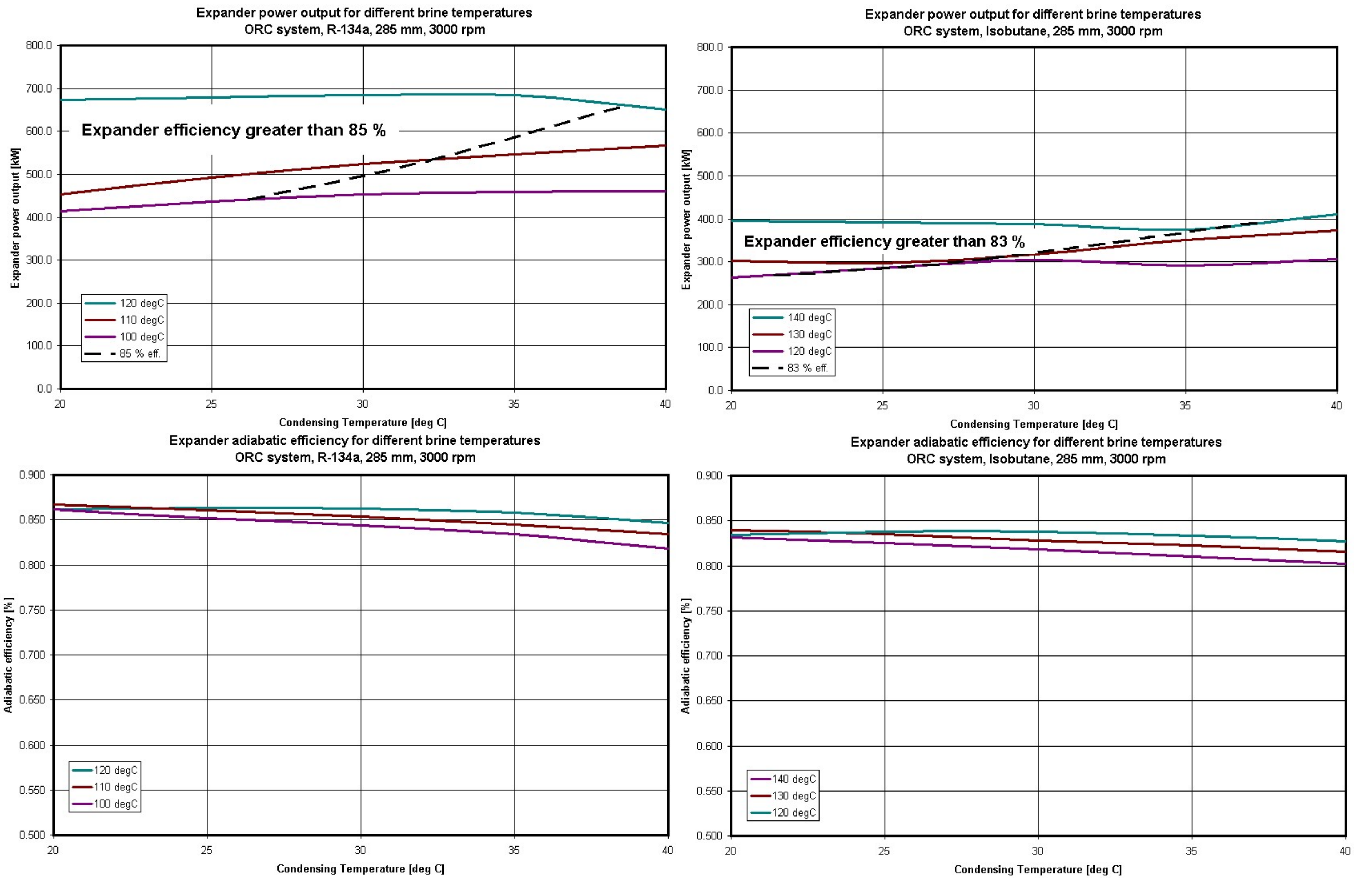


Fig 6 Power and Efficiency of 285 mm Screw Expander ORC System
Working Fluids R134a and Isobutane

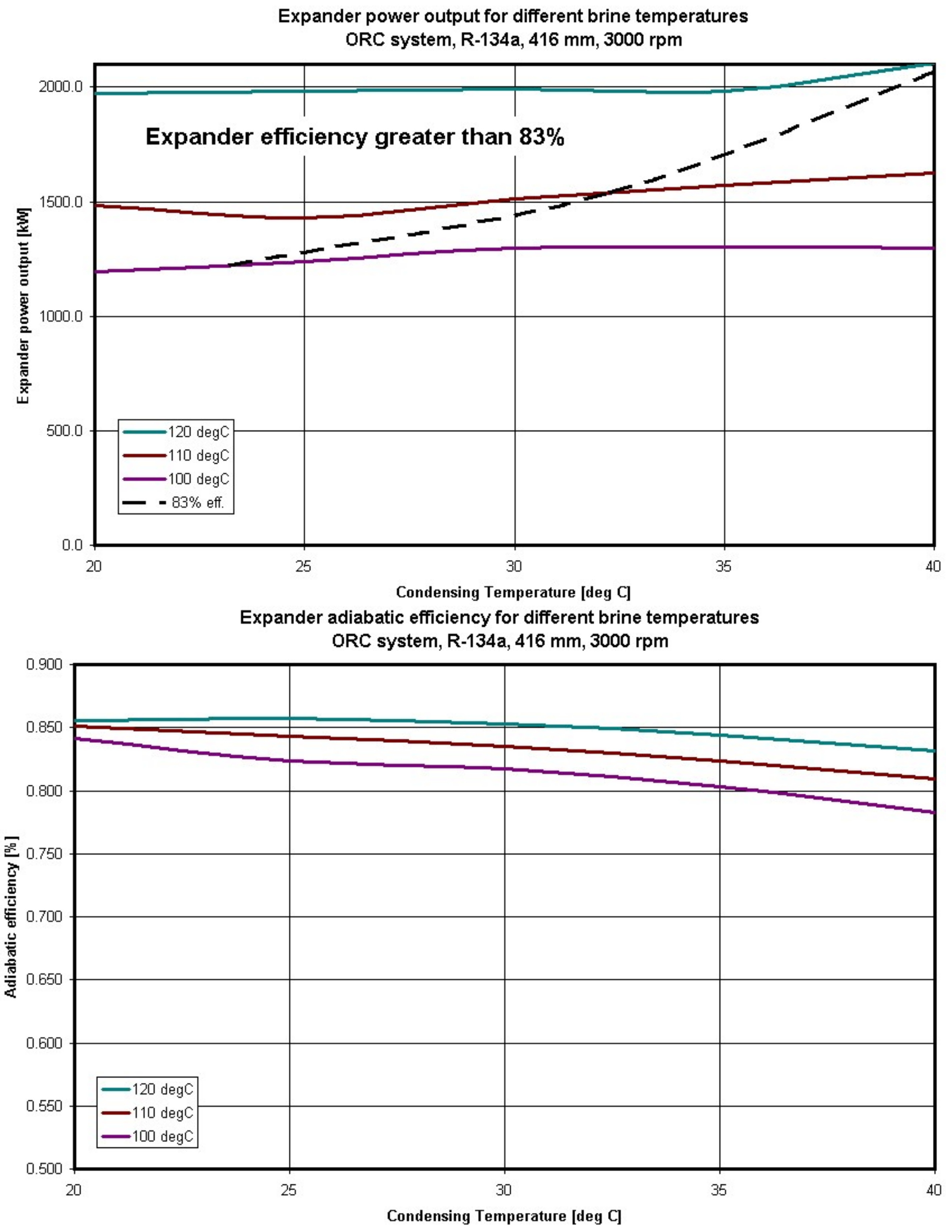


Fig 7: Power and Efficiency of 416 mm Screw Expander ORC System Working Fluid R134a

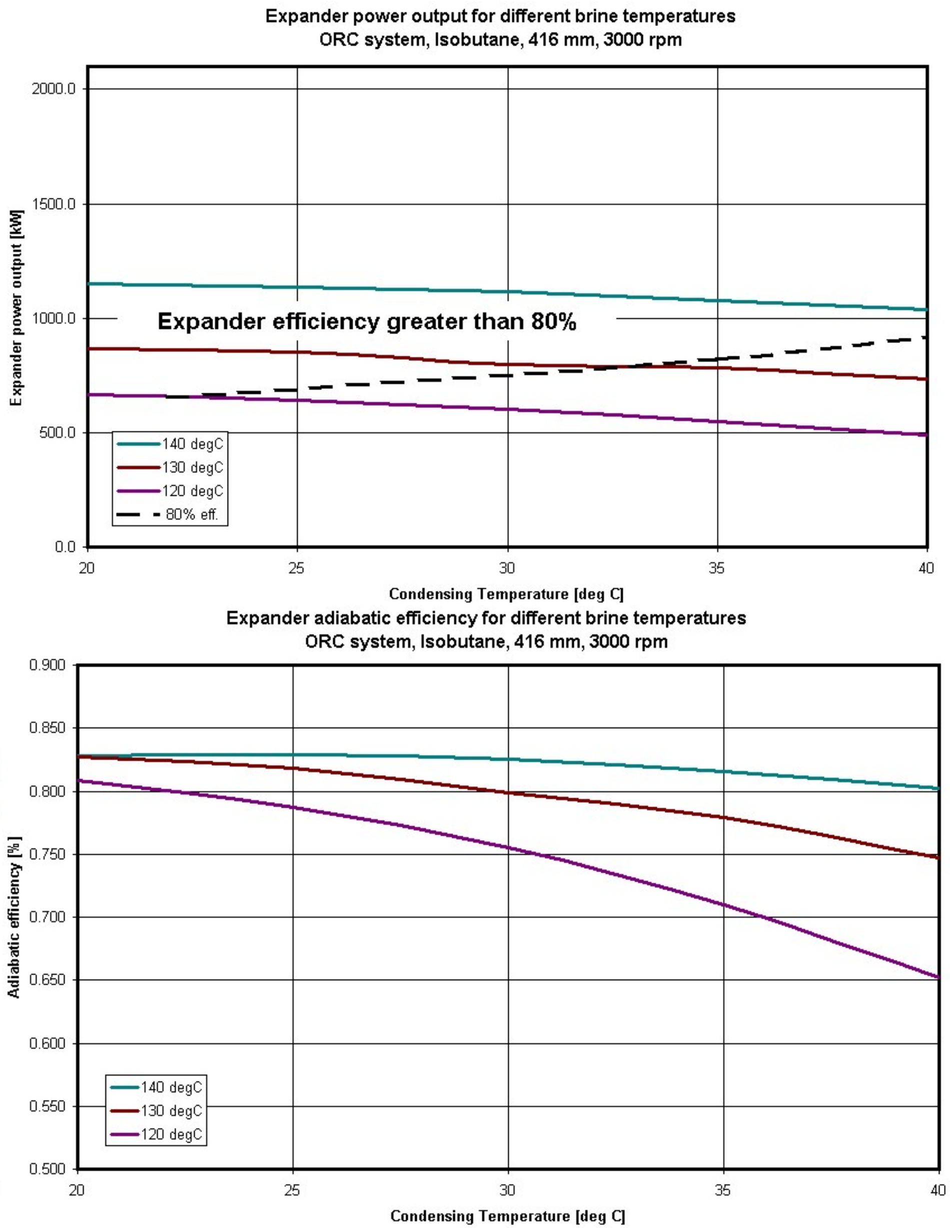


Fig 8: Power and Efficiency of 416 mm Screw Expander ORC System Working Fluid Isobuta