TRILATEKAL FLASH CYCLE SYSTEM
A HIGH EFFICIENCY POWER PLANT FOR LIQUID RESOURCES

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ABSTRACT

The Trilateral Flash Cycle (TFC) system is basically a binary power plant in which expansion starts from the saturated liquid rather than the saturated, superheated or supercritical vapour phase. By this means the transfer of heat from a liquid heat source to the working fluid is achieved with almost perfect temperature matching. Irreversibilities are thereby minimised and hence, provided that the two-phase expansion process is efficient, its potential power recovery is 14 - 85% more than from ORC or flash steam systems. The preferred working fluids for it are light hydrocarbons.

Extensive analytical and experimental investigations have shown that, contrary to the total flow experience, by correct design and working iluid choice, twin screw machines are most suitable for two-phase expansion. Thus adiabatic efficiencies of over 70% were obtained from a 25 kW experimental machine and values in excess of 80% are predicted for multi megawatt units if built with two stages. At higher temperatures, expansion to dry vapour is possible. The expander may then contain a final turbine stage.

A detailed engineering design study of a TFC plant for power recovery from a Hot Dry Rock source has been carried out for the UK Department of Energy. This showed the fully installed cost for a 9 MWe TFC plant to be $1700 per KW nett and $1300 per kW gross.

THE SYSTEM

For power recovery from single phase heat sources, such as hot liquids, the ideal thermodynamic cycle appears on Temperature-entropy coordinates as a trilateral figure, as shown in Fie 1.

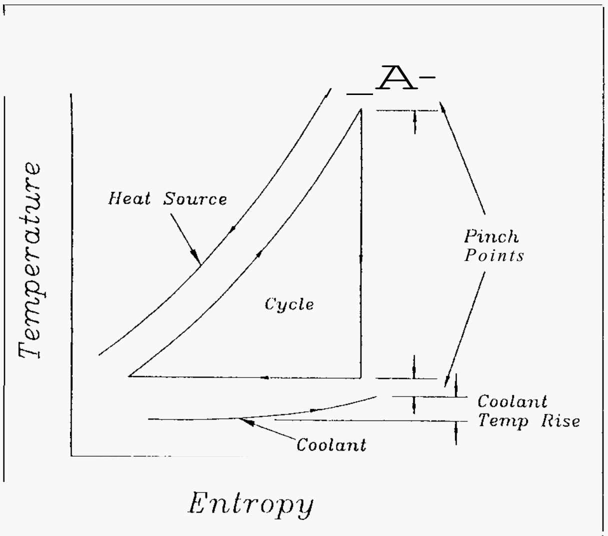

Figure 1 The Ideal Trilateral Cycle

One means of approximating fairly closely to it is to heat a working fluid to its boiling point only and then expand it as a two-phase vapour. The resulting cycle and the system components needed to follow it are shown in Fig 2. The layout of parts is identical to that of Rankine cycle systems but it results in much higher output since simultaneously more of the available heat is recovered while higher fluid temperatures are attained. Such a system has been described in more detail by Smith (1993) and named as a Trilateral Flash Cycle (TFC) system. It was conceived, primarily as a means of recovering power from hot liquid streams in the 100" - 250°C temperature range.

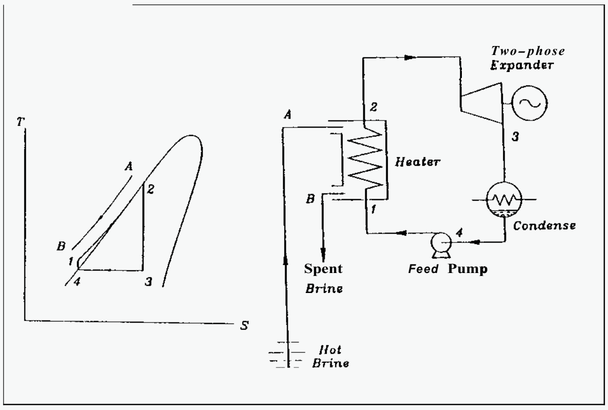

Figure 2 The Trilateral Flash Cycle System

An assessment of the relative performance advantage of the TFC system over alternative simple Rankine cycle systems is shown in Fig 3. It is' assumed that the power plant employs water cooled condensers with recirculation of the coolant through forced draught cooling towers. The adiabatic efficiencies assumed for the expansion are 80% for the Rankine systems and 77% for the TFC system. The output shown is nett, after subtraction of feed and cooling water pump and cooling tower fan power. A variety of working fluids are shown, including halocarbons no longer permitted, to demonstrate different attributes. Thus, it may be seen from the R12 curve that the TFC is more efficient than the super and transcritical Rankine cycles as well as the normal subcritical cycle. Although water gives the highest theoretical efficiency, it is not a practical working fluid due to the huge volume ratios and volume flow rates involved as shown in Figs 4 and 5. As explained by Smith (1993), light hydrocarbons are the most suitable fluids from nearly all aspects.

The primary heat exchanger of such a system must be larger than that of a Rankine cycle boiler due to the greater heat transfer and increased surface area resulting from close temperature matching between the working fluid and the heating source. In addition, the condenser will be larger because more heat is rejected. The higher cost of the bigger heat exchangers must therefore be offset by the increased power output. This will only be achieved if the adiabatic efficiency of a two-phase expander approaches that of a dry vapour turbine

TWO-PHASE EXPANSION

Two-phase expanders were studied extensively during the nineteen seventies, primarily to develop total flow system. Sprankle's (1973) proposal to use a Lysholm twin screw machine was followed by intensive investigation by Steidel et al (1982), (1983) on small scale units but they obtained adiabatic efficiencies only of the order of 50%. Further work on a 1 MW unit by LaSala et al (1985) was equally disappointing. Elliott (1982) carried out an extensive study of two-phase turbines but ultimately was able to predict maximum efficiencies of only 65 - 70% when using organic fluids in a closed cycle and rather lower values with water as the working fluid. More favourable predictions for two-pirase Lysholm expanders were made by Taniguchi et al (1988), who carried out small scale tests accompanied by extensive computer simulations in an effort to develop a throttle valve replacement in large heat pumps.

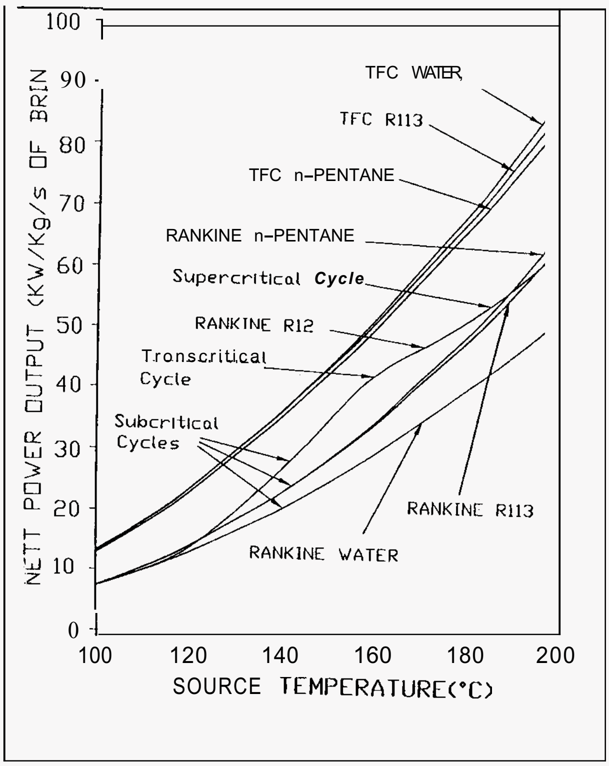

Figure 3 Relative System Performance

The authors, working independently, concluded that the flow rates and volume ratios required for water expansion made it an unsuitable working fluid. Much better chances of success appeared to be likely with light hydrocarbons as working fluids where the

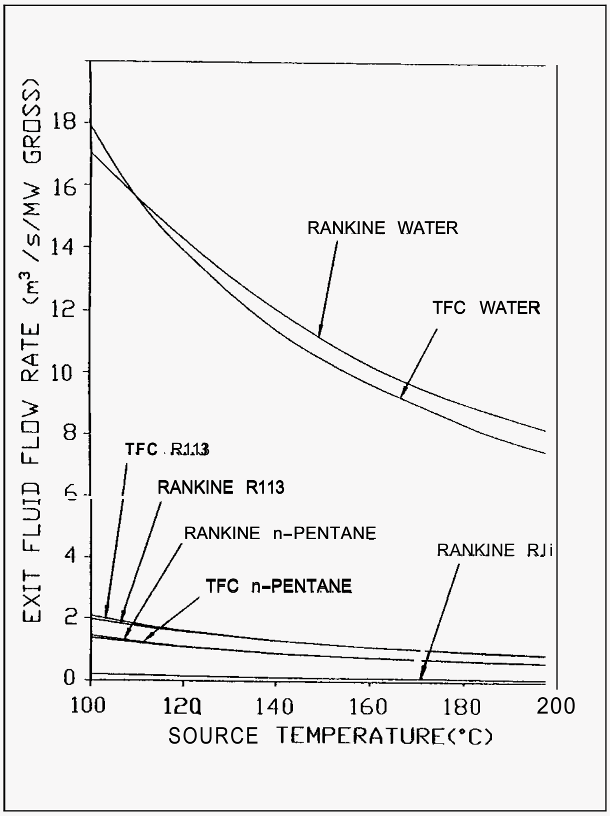

Figure 4 Expander Exit Fluid Flow Rate

volume flow rates could be reduced by a factor of 10 or more and volume ratios by factors of several hundred. On this basis Lysholm twin screw expanders, possibly in one stage but more probably in two stages, seemed to offer the best prospects.

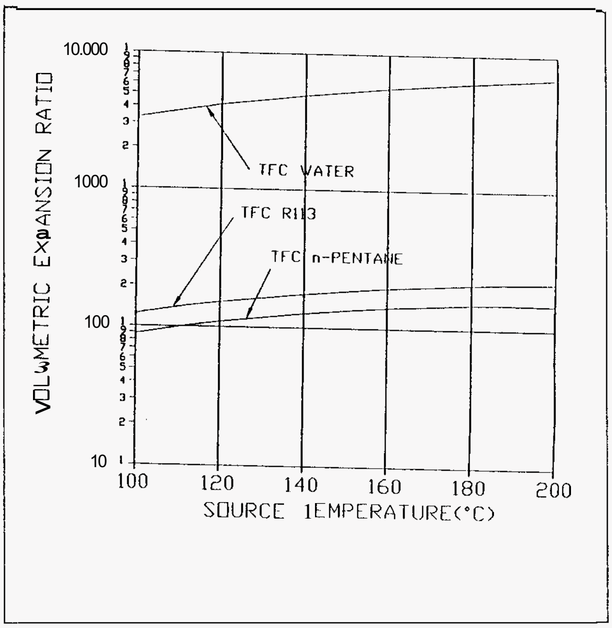

Figure 5 Volumetric Expansion Ratio

A test rig was constructed at City University, London, in which initial tests were carried out on a converted compressor of 204 mm rotor diameter using R113 as the working fluid. Some of the first results are shown in Fig 6. As can be seen, the maximum overall adiabatic efficiency achieved was just over 72% at a power output of approximately 25 kW.

Such an encouraging start led to a series of modifications including both higher and lower built in volume ratios and latterly the

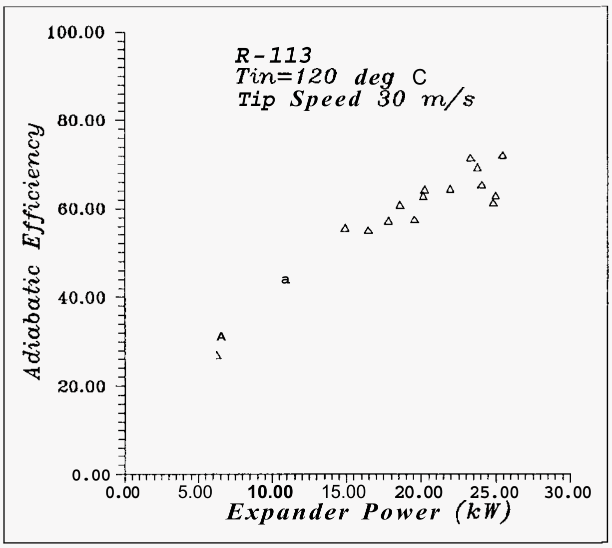

Figure 6 Small Scale Expander Test Results with R-113

construction and testing of a new purpose built expander of 163 mm rotor diameter of increased Length:Diameter ratio. However, in many cases the modifications altered the performance in a manner contrary to that anticipated, especially at higher speeds when attempts to increase the mass flow led to a decrease in the power and efficiency.

Accordingly, in order to gain a better understanding of the power production process within a screw expander, a method of analysis, broadly similar to that used for screw compressors, was developed to predict the performance. This was to write a computer program to solve the set of simultaneous non-linear differential equations which describe the bulk flow, leakage, friction and heat transfer effects in the volume trapped between the rotor lobes as they rotate. A pressure-volume diagram of the expansion process was thereby produced and integrated to estimate the work output and hence the efficiency. The analytical studies were then checked by a further experimental programme carried out on a revised version of the 163 mm expander which was designed and built by the authors. In this machine, pressure-volume diagrams of the expansion process were obtained by direct measurements.

A typical result showing both predicted and measured prcssure changes during the expansion process is shown in Fig 7. Note, that due to the high density of the working fluid at admission, the filling process is accompanied by a relatively large pressure drop which contributes significantly to the overall expansion achieved within the machine. The close agreement between the experimental and estimated values confirms the overall validity of the analytical model

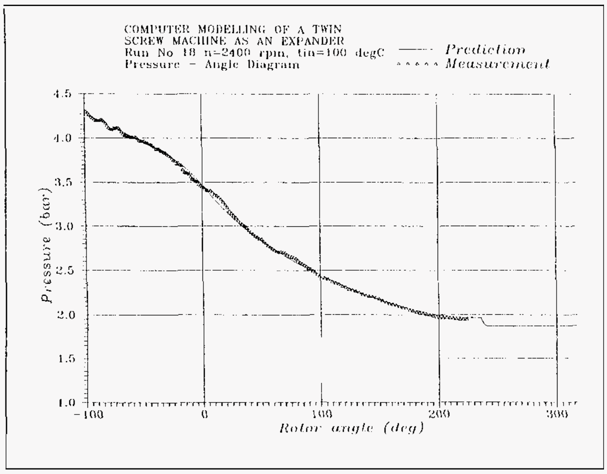

Figure 7 Measured and Estimated Pressure-Rotation Angle Diagram

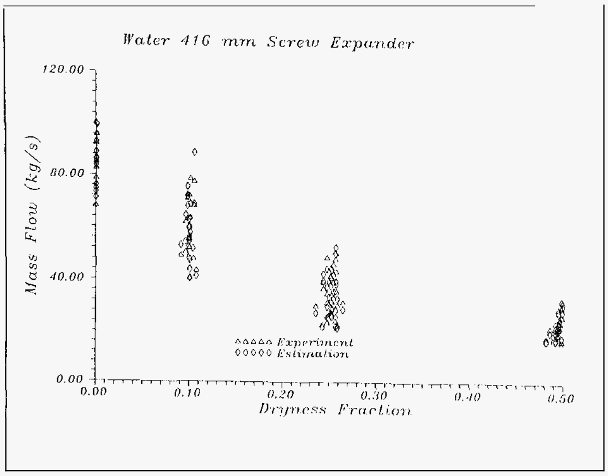

Figure 8 Total-Flow Expander Estimated and Measured Mass Flow Rate

Initially, the program was used to analyze the authors' test results. When satisfactory agreement between theory and experiment was obtained its use was extended to examine the published test results of both the small and large scale total flow expanders as produced by Steidel et al (1982) and LaSala et al (1985) as well as the small scale experiments carried out by Taniguchi et al (1988) using R12 as the workmg fluid. In all, 636 test points were studied from 8 builds of machine. This included three different working fluids, power outputs of less than 10 to over 800 kW, rotor sizes of 81 - **416** mm diameter, built in volume ratios of between 3:1 and 9.1:1, tip speeds of 7 - 55 m/s and inlet temperatures of 180 - 80°C.
Statistical analysis showed that the closest agreement between theory and experiment was obtained with the total flow data. A comparison between the predicted and measured values of mass flow, power output and adiabatic efficiency, given only the speed,

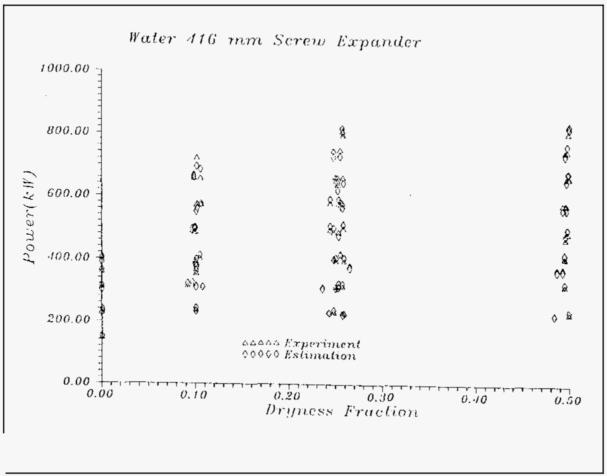

Figure 9 Total-Flow Expander Estimated and Measured Power

the fluid entry condition and the expander exit pressure as input data for the analysis, is given in Figs 8 - 10 for the 1 MW unit described by LaSala. **A** comparative set of the authors' data, which includes the results shown in Fig 7, is given in Figs 11 - 13. Both sets of results cover a wide range of speeds and inlet temperatures. The slightly better agreement of the total flow results is attributed to the expander working over a larger temperature and pressure difference than the R113 and R12 machines. Errors due to the relatively small difference between large quantities were thereby minimised. Note the much higher maximum efficiencies obtained with the far smaller R113 expander.
From a study of the results, it became possible to understand the processes involved and thereby explain both the poor performance of the total flow systems and the apparently anomalous behaviour of the authors' test results.

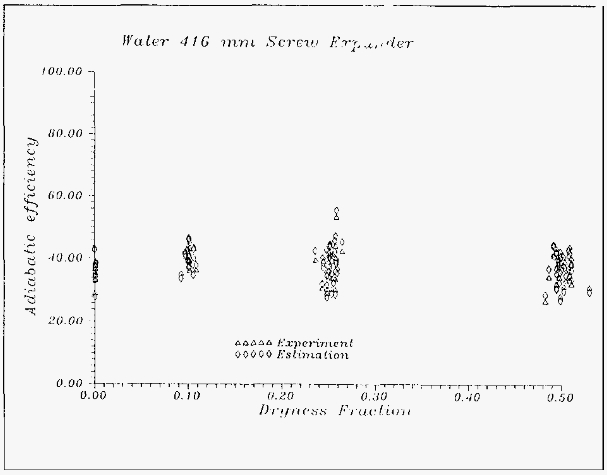

Figure 10 Total-Flow Expander Estimated and Measured Adiabatic Efficiencies

In the case of the total flow tests the main difficulty was that due to the very high critical temperature of water, its vapour pressure is very low at normal condensing temperatures. Accordingly its vapour density is very low and hence enormous volume ratios are required for two-phase expansion as already shown in Fig 4. In the small scale tests the volume ratios really required were of the order of 20 or 30:1 for full expansion to take place within the rotors. However, the built in volume ratio was only in the range of 3 - 7:1 and hence expansion was completed after the water-steam mixture left the rotors. This led to low efficiencies. Paradoxically, reducing the back pressure and thereby the increasing the external expansion, led to improved efficiencies. The reason for this is that in small machines the friction effects are relatively high. Hence reducing the back pressure increases the area of the p-v diagram and thereby the work output. Friction

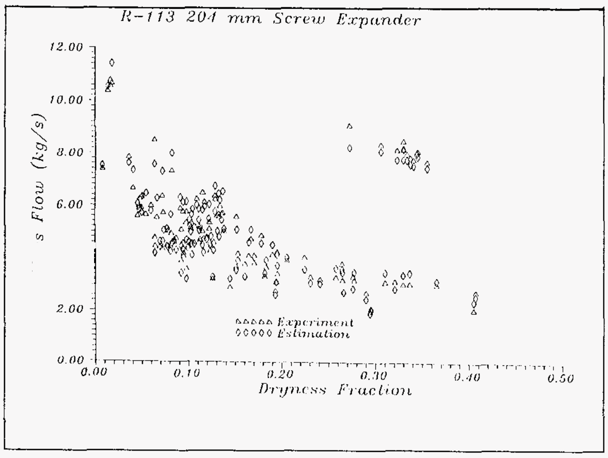

Figure 11 Small Scale Expander Estimated and Measured Mass Flow Rate

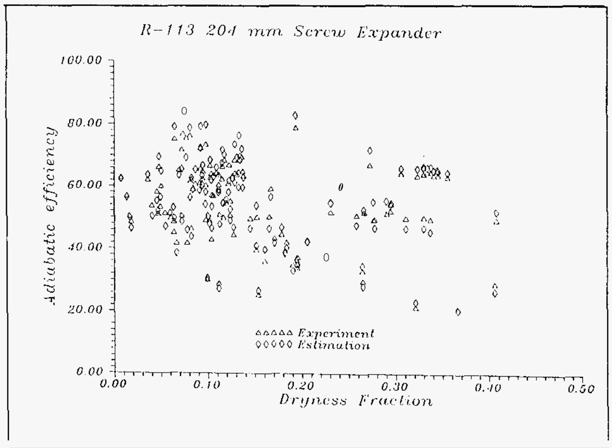

Figure 13 Small Scale Expander Estimated and Measured Adiabatic Efficiencies

losses thus become a smaller percentage of the total. Accordingly, despite an increase in expansion of the vapour beyond the rotors, the adiabatic efficiency increased.
In the larger scale total flow tests, higher volume ratios were employed and therefore better results were anticipated. Here, though, another difficulty occurred. As the built-in volume ratio of a given size machine is increased, the mass of fluid induced in the filling process is decreased. Leakage losses, which are mainly dependent on overall pressure drop, then become a much larger fraction of the total induced mass. The anticipated increases in efficiency were therefore not achieved.

In contrast to water, organic fluids with far lower critical temperatures may be chosen. By this means, overall volumetric expansion ratios of the the order of 50:1, or even less at lower

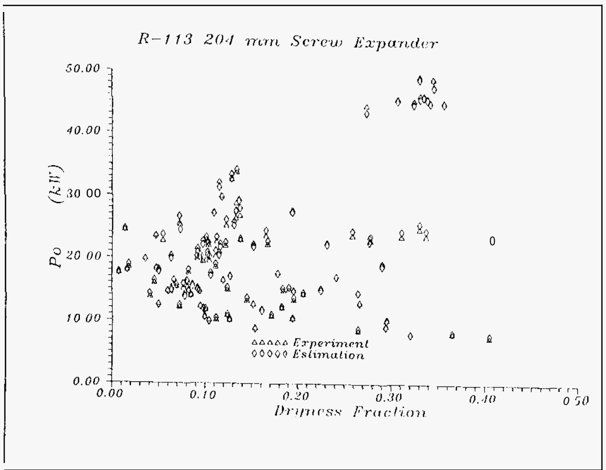

Figure 12 Small Scale Expander Estimated and Measured Power

temperatures, are required. Far higher efficiencies are therefore possible because compact expanders with low built in volume ratios may be built within which the working fluid expands fully before discharge from the rotors. This is exemplified in the authors' test results.

The unexpected fall off in performance at higher speeds when using R113 was due to the opposite effect to that experienced in the total flow experiments. Increasing the speed creates a greater pressure drop in the filling process associated with a higher fluid entry velocity. This is the non-steady flow equivalent of flow through a nozzle. As a result of the greater expansion in filling, the R113 was overexpanded in the remainder of the process so that there was a flow reversal and pressure rise in the fluid when the discharge port opened leading to negative work during discharge of the fluid. The mass flow could only be increased by making the fluid wetter, and hence more dense, at entry. This produced greater overexpansion and thus reduced the power and efficiency even more.

LARGE SCALE PERFORMANCE PREDICITIONS

The two-phase expander simulation program was used to predict the performance of a demonstration plant at a resource where a supply of hot brine cooling from 130°C to 47°C was available and condensation at 38°C was required. The working fluid selected was isobutane and an optimisation study showed that maximum output would be obtained when it was heated to 118°C.

For the expander, a two stage system was selected. The sizes chosen for the two machines corresponded to those of standard

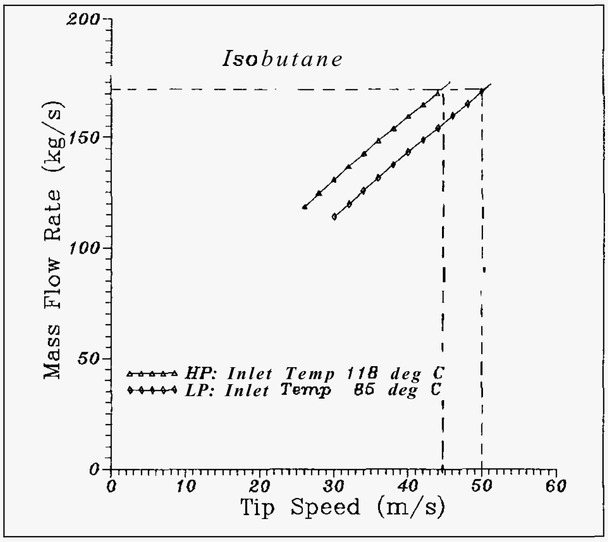

Figure 14 Large Scale Expander Estimated HP and LP Mass Flow Rate

industrial compressors. Accordingly the rotor sizes selected were 408 mm for the high pressure stage and 642 mm for the low pressure stage. The mass flows, outputs and efficiencies predicted are shown in Figs 14-16. As may be deduced, for a maximum tip speed of 50 m/s a total of 4.4 MW is possible with an overall adiabatic efficiency of 80%. Revised studies carried out since, have shown that by use of more advanced rotor profile shapes, further gains in adiabatic efficiency of at least 2 percentage points are possible with corresponding gains in power output.

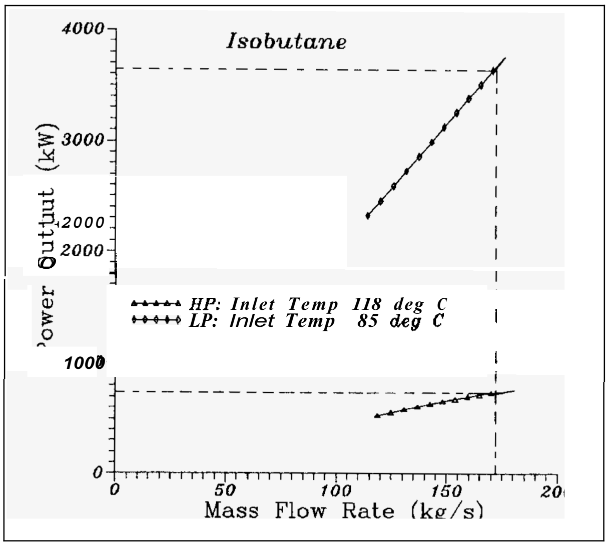

Figure 15 Large Scale Expander Estimated HP and LP Power

A FULL SCALE DESIGN STUDY

In 1987 the UK government Department of Energy commissioned a study on power recovery from Hot Dry Rock sources as part of its funded study of geothermal energy utilisation in South West England. Shock (1987), in an initial comparative study between flash steam, ORC and TFC systems showed the TFC system to be the best proposition. In this case, high output was especially desirable because the electricity cost per unit output included the drilling costs compared to which power plant costs were of only secondary consideration. A full design study of the TFC plant was therefore commissioned to determine its true cost and later published by Clarke (1988). A summary of the results is as follows:

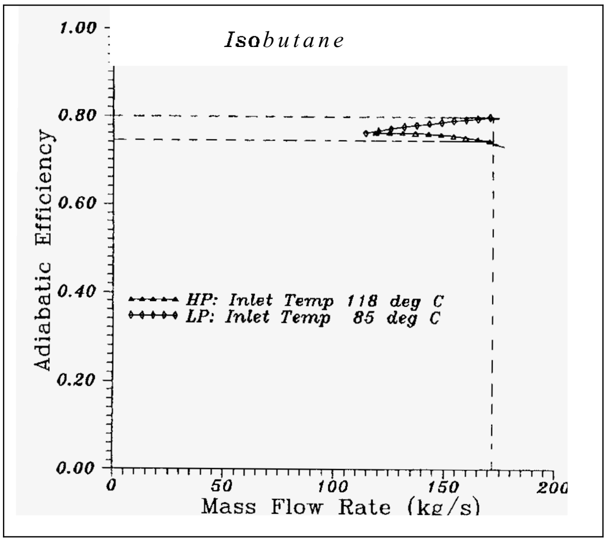

Figure 16 Large Scale Expander Estimated HP and LP Adiabatic Efficiency

Resource Details:

Brine flow rate	75 kg/s
Brine supply temperature	200 °C
Mean annual cooling water temperature	19 °C

System Specification:

Cycle and Expander System: Due to the relatively high resource temperature, using n-Pentane as the working fluid, it was possible for the two-phase expansion process to end in the dry vapour phase, as shown in Fig 17. The expander configuration then chosen was a two stage screw expander followed by a final single stage radial inflow turbine. The turbine performance is as specified by Rotoflow (1987). The screw expanders were chosen to correspond with standard sizes used for large process gas compressors. Their performance was estimated with the simulation program described in the last section.

Cooling System: Cooling water recirculated through forced draught cooling towers.

Water Temperature Rise in Condenser	6.1°C
Condenser pinch point temperature difference	5.8°C

Heater:

Maximum working fluid temperature	181.6°C
Brine reinjection temperature	35.9°C
Heater pinch point temperature difference	5.0°C

Auxiliaries: These comprise the feed pump, the cooling water and cooling tower water circulation pumps and the cooling tower fans.

Estimated Performance:

	Power Output (MWe)	Adiabatic Effic (%)
HP Screw (408 mm Rotors)	890	78.5
LP Screw (642 mm Rotors)	2460	82.1
Radial Inflow Turbine	5630	85.0
Total	**8980**	**84.7**
Auxiliary Power:	1735	
Nett total output:	**7245**	
Gross ORC output: (manufacturers quote)	**5500 Approx**	

System Cost

This includes power plant, cooling system, all civil engineering costs for installation, buildings, access roads and site fencing but excluding the drilling costs and connection to the grid. Prices are those quoted in 1987.

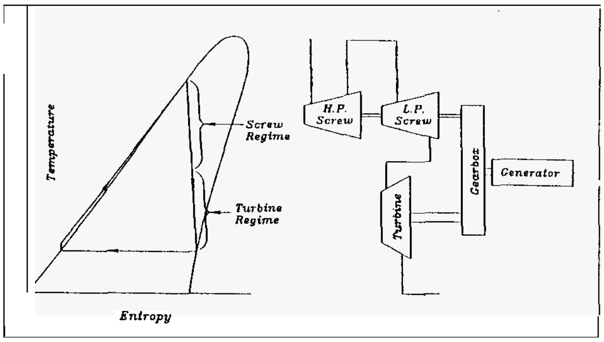

Figure 17 Expansion to Dry Vapour, the Concept and the System

	cost $ US
Equipment	4,448,620
Equipment installation	44,980
Piping	632,000
Piping Instal, fabricate and test	1,240,000
Electrical including installation	606,400
Instruments	620,800
Instrument installation	136,000
Steelwork	300,270
Steelwork erection	267,730
Civils including labour costs	1,528,000
Insulation, fireproofing, etc	72,000
Design, Engineering and Licence	1,865,600
Plant installed cost	11,762,400

This is equivalent to an installed cost of $1700/kWe Nett ($1300/kWe Gross) which is of the same order as the cost of conventional alternatives.

CONCLUSION

Experimental and theoretical studies have shown conclusively that the TFC system is capable of significantly higher outputs than ORC or flash steam systems. As a result of a sustained R and D programme, it is now possible to design and construct twin screw expanders for such applications with predicted adiabatic efficiencies of the order of 80% or more with rotors no larger than those used in large scale process gas compressors. Detailed design studies have shown that the cost of such systems compares favourably with existing Flash Steam and ORC power plant.

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