Development of the Trilateral Flash Cycle System: Part 1: Fundamental Considerations

I K Smith

Proceedings of the Institution of Mechanical Engineers, Part A: Journal of Power and Energy 1993 207: 179
DOI: 10.1243/PIME_PROC_1993_207_032_02

Development of the trilateral flash cycle system Part 1: fundamental considerations

I K Smith, BSc(Eng), DIC, PhD, CEng, FIMechE
Department of Mechanical Engineering and Aeronautics, City University, London

The world market for systems for power recovery from low-grade heat sources is of the order of £1 billion per annum. Many of these sources are hot liquids or gases from which conventional power systems convert less than 2.5 per cent of the available heat into useful power when the fluid is initially at a temperature of 100 °C rising to 8–9 per cent at an initial temperature of 200 °C. Consideration of the maximum work recoverable from such single-phase heat sources leads to the concept of an ideal trilateral cycle as the optimum means of power recovery. The trilateral flash cycle (TFC) system is one means of approaching this ideal which involves liquid heating only and two-phase expansion of vapour. Previous work related to this is reviewed and details of analytical studies are given which compare such a system with various types of simple Rankine cycle. It is shown that provided two-phase expanders can be made to attain adiabatic efficiencies of more than 75 per cent, the TFC system can produce outputs of up to 80 per cent more than simple Rankine cycle systems in the recovery of power from hot liquid streams in the 100–200 °C temperature range. The estimated cost per unit net output is approximately equal to that of Rankine cycle systems. The preferred working fluids for TFC power plants are light hydrocarbons.

NOTATION

MW	relative molecular mass
R	specific ideal gas constant
T_0	ambient temperature
T_1	initial source temperature
T_c	fluid critical temperature
η_{cycle}	cycle efficiency (net work output/heat input)
η_{recov}	heat-recovery efficiency (heat received/heat available)
η_{con}	conversion efficiency (net output/heat available)

1 INTRODUCTION

1.1 The market for low-grade heat power generation systems

Consideration of the Earth's limited fossil fuel reserves and fears of environmental pollution have led to a growing interest in the generation of power from non-fossil fuel sources other than nuclear energy. One consequence of this is an intensified study of the recovery of power from sources of heat at temperatures far lower than those previously thought to be worth while. The most abundant of such low-grade heat sources is geothermal energy, from which there was a world installed generating capacity of approximately 6000 MW by the end of 1991 with a projected capacity of nearly 9000 MW by the end of 1995 (**1**). This is equivalent to an average annual growth rate of over 8 per cent. The installed cost of geothermal power plant is approximately $2000 per kW. Hence, assuming a uniform rate of growth during the period considered, the average annual expenditure on such systems is currently of the order of £1 billion.

Hot dry rock geothermal resources are very widespread and are most easily accessible in South West England. Unfortunately, available rock drilling and fissuring techniques are too expensive to enable heat to be recovered from them at an economic cost. The bulk of geothermal power is therefore generated around the Pacific 'ring of fire' associated with the Earth's tectonic plate junctions, where drilling is cheap and natural aquifers abound. Few of the countries in these regions have the resources to design and build the power plant they employ. Thus there is a considerable export potential for such systems.

Studies carried out in the 1970s in the wake of the then oil crisis (**2**) showed that at that time there were considerable amounts of waste heat unused by industry which might be used to generate power. Unfortunately, the economic criteria of power generation, which are based on long payback times and assured sales, conflict with those of manufacturing industry, which are based on payback times of the order of one to two years. Industrial use of waste heat for power generation therefore did not then materialize. The non-fossil fuel obligation imposed by the 1989 Electricity Act and the threat of a possible CO_2 tax may induce a change in attitude of industry to power generation, while more recent surveys (**3**) have shown that there are still large amounts of available industrial waste heat.

In view of the foregoing considerations a greater interest in power recovery from low-grade heat than the British power generation industry has hitherto shown appears to be justified.

1.2 Basic considerations of power generation from low-grade heat

All low-grade heat-generating systems are inherently more expensive to construct than their fuel-fired counterparts since:

1. In the absence of internal combustion, heat exchangers are normally required.
2. More heat must be transferred per unit of power output due to the reduced cycle efficiencies resulting from the low temperatures.
3. The heat exchangers involved must have a larger surface area per unit of heat transferred to minimize

The MS was received on 27 July 1993 and was accepted for publication on 18 October 1993.

the temperature difference across them. This is essential if the overall temperature difference between heat supply and rejection within the power plant is to be large enough to obtain a worthwhile conversion efficiency.

The best low-grade heat source for the recovery of power is a stream of steam or vapour which can readily be passed through an expander. Such streams are unfortunately rare. Much more common are streams of hot gases or liquids which can only be used as sources of heat. Any transfer of heat from a single-phase heat source is associated with a change in its temperature. This has profound effects on the type of thermodynamic cycle required for optimum power recovery (**4**), particularly when the resource temperature is less than the normal containment metallurgical limit of approximately 550 °C. The trilateral flash cycle (TFC) system was conceived specifically to optimize power recovery from such sources in the 100–250 °C temperature range.

1.3 The ideal single-phase heat source cycle

Classical heat engine theory is based on the Carnot cycle concept in which both the heat source and sink are considered to be infinite. Accordingly their temperatures do not change as a result of heat transfer and the heat supply and rejection processes within the Carnot cycle engine are at constant corresponding temperatures. Consider an ideal Carnot cycle heat engine receiving heat from a finite single-phase heat source, such as a hot liquid or gas stream, of constant specific heat capacity initially at temperature T_1 and rejecting heat to the surrounding atmosphere, the temperature of which remains constant at T_0. The processes of both the cooling heat source and the heat engine are shown in Fig. 1. Since heat supply to the engine is at constant temperature T, the amount of heat that can be transferred to it is $\propto (T_1 - T)$, whereas the maximum heat transferable from the source is $\propto (T_1 - T_0)$. Increasing T raises the Carnot cycle efficiency but reduces the amount of heat transferred. It can readily be shown that the work recovery is a maximum when $T = (T_1 T_0)^{0.5}$. In that case:

$$\text{heat-recovery efficiency} = \frac{T_1 - (T_1 T_0)^{0.5}}{T_1 - T_0} = \eta_{\text{recov}}$$

and

$$\text{cycle efficiency} = \frac{(T_1 T_0)^{0.5} - T_0}{(T_1 T_0)^{0.5}} = \eta_{\text{cycle}}$$

while the overall conversion efficiency of available heat to work,

$$\eta_{\text{con}} = \eta_{\text{recov}} \eta_{\text{cycle}} = \frac{T_1 - (T_1 T_0)^{0.5}}{T_1 - T_0} \cdot \frac{(T_1 T_0)^{0.5} - T_0}{(T_1 T_0)^{0.5}} \quad (1)$$

This is only about 30 per cent of the efficiency of a Carnot cycle operating between infinite heat reservoirs at T_1 and T_0 for source and sink temperatures normal for low-grade heat utilization.

A better criterion for the maximum conversion of heat to work may be obtained if a finite heat source is considered as an infinite number of constant temperature heat sources. Each of these can supply heat to the environment through an infinitesimal Carnot cycle engine, as indicated in Fig. 2a, to produce the maximum amount of work with 100 per cent usage of available heat. In the limit this is equivalent to the supply of heat to a single finite cycle, as shown in Fig. 2b. Such a cycle has been described (**5**) as an ideal trilateral cycle. By integration of the infinitesimal Carnot cycles between T_1, the initial resource temperature, and T_0, the environmental temperature, it can be shown that, for this ideal cycle,

$$\eta_{\text{conv}} = \eta_{\text{cycle}} = 1 - \frac{T_0 \ln(T_1/T_0)}{T_1 - T_0} \quad (2)$$

This leads to ideal conversion efficiencies roughly twice the value of those obtained using equation (1).

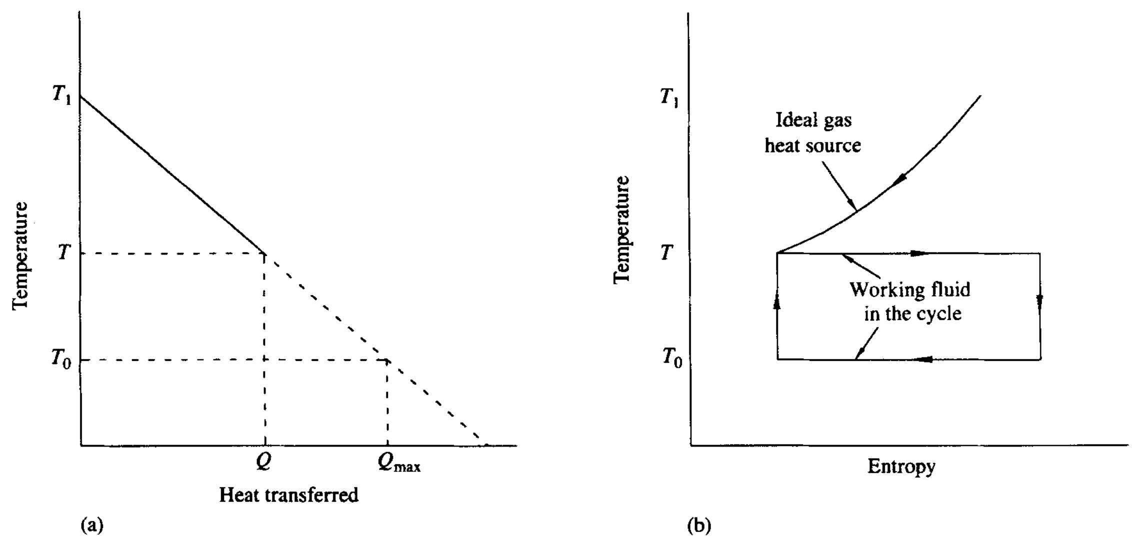


Fig. 1 An ideal Carnot cycle engine receiving heat from a constant pressure single-phase heat source. Heat supplied to the engine as a function of maximum cycle temperature is shown in (a). The combined heat source and working fluid processes are shown in (b)

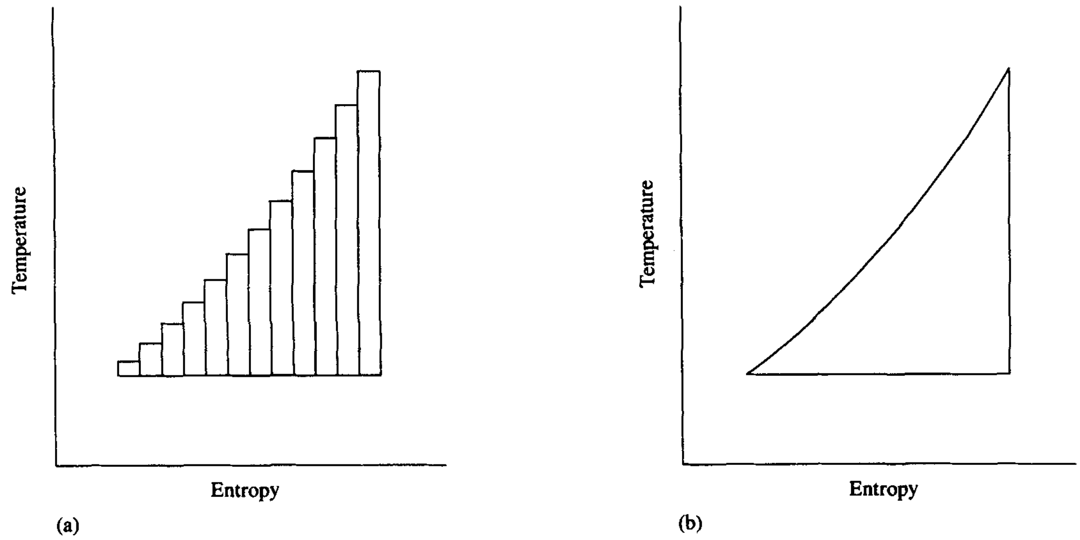


Fig. 2 Temperature–entropy diagrams of the ideal mode of power conversion from a constant pressure single-phase heat source. As shown in (a), this is a succession of infinitesimal Carnot cycles. In integral form this constitutes the ideal trilateral cycle as shown in (b)

The same expression can also be derived by consideration of the exergy change of the resource as it cools from temperatures T_1 to T_0. The disadvantage of deriving it without reference to a thermodynamic cycle is that it gives no indication of how best to design a heat engine to approximate to it in a sequence of real processes.

The ideal trilateral cycle cannot be realized in practice if the heat source itself is not the working fluid and there is to be a finite rate of heat transfer between it and the ideal heat engine to which it is supplying heat and also in the heat rejection process from the ideal heat engine to its surroundings. If these factors are considered, then a more practical form of the ideal would be as shown in Fig. 3. The additional external limitations to the trilateral ideal are then the minimum or pinch point temperature differences between the heat source and the engine, the engine and its coolant supply, and the actual temperature rise of the coolant in the process of heat rejection. It should be noted that if the latter value approaches zero, then the coolant flowrate tends to infinity and hence would require infinite power to be sustained.

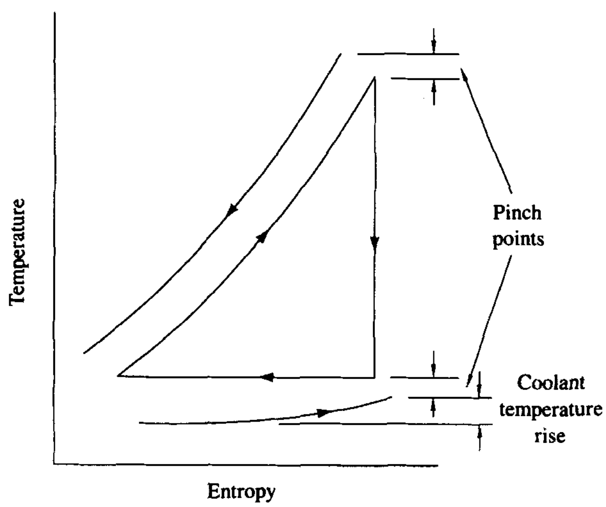


Fig. 3 The ideal trilateral cycle with a finite rate of heat exchange between a constant pressure heat source and sink. Note the necessity for a temperature rise in the heat sink

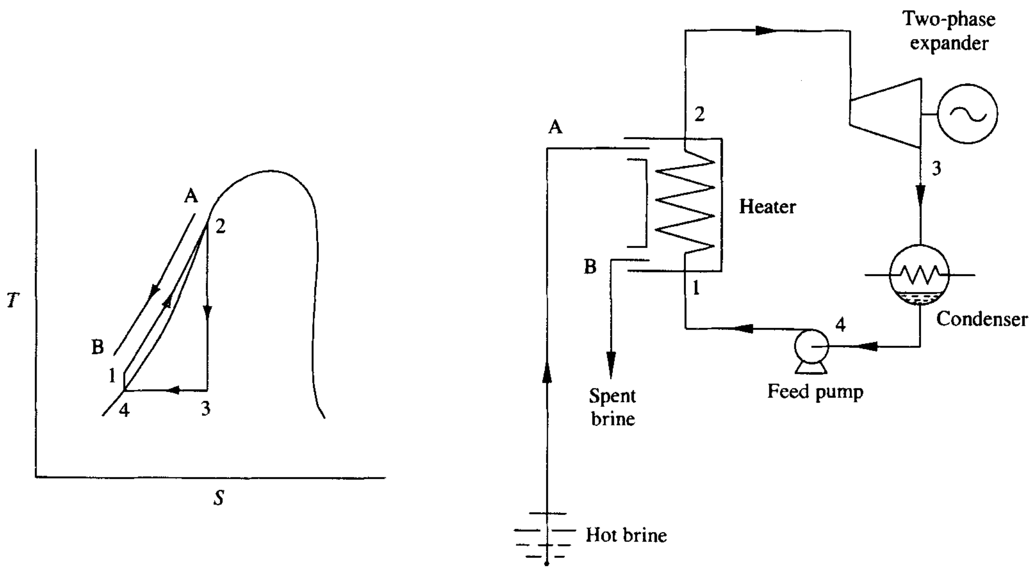
1.4 The trilateral flash cycle

A close approach to the ideal trilateral cycle can be achieved as shown in Fig. 4a. Here, a volatile fluid is pressurized adiabatically, heated at constant pressure to its boiling point, expanded adiabatically as a two-phase mixture, which flashes into vapour as the pressure drops, and condensed at constant pressure. This was first described by the author as a trilateral wet vapour cycle (**6**), but for brevity it was later called a trilateral flash cycle (TFC). The components required to perform these processes are shown in Fig. 4b. Only one of these is unconventional and that is the two-phase expander. The premise on which the development of this system was based was that such a unit could be made with efficiencies approaching those of a vapour turbine.

2 PREVIOUS WORK

2.1 Ruths' patent

The value of efficient two-phase expanders for a variety of functions has long been appreciated, although hitherto such machines have hardly been realized. In 1923 Ruths (**7**) suggested that hot water stored in a steam accumulator could be better utilized on demand by flash expansion of the bulk of it through either a reciprocating engine or a turbine and then discharged either as waste or condensed and returned as cold water to the accumulator. If returned to the accumulator it was not to mix with the residual hot water still undischarged. The whole arrangement is shown in Fig. 5. Presumably the water would be reheated by direct injection of fresh steam during the next charging process. He considered this to be preferable to drawing off only a small portion of the stored water as steam



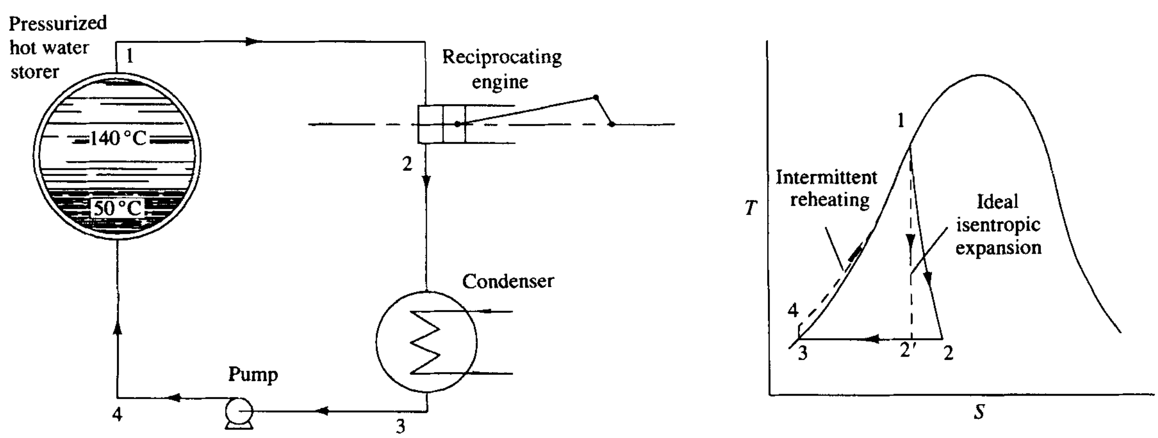
(a) The process (b) System components

Fig. 4 The trilateral flash cycle system. A temperature–entropy diagram of the thermodynamic processes is shown in (a) while the system arrangement to effect them is given in (b)

into the main steam line when the mains pressure started to fall. The whole sequence of processes therefore constituted an intermittently operating TFC system with direct contact heating by steam. Such an arrangement in fact only shows thermodynamic advantages when the first charge of water in the accumulator is considered. Subsequently, if the processes were to be repeated, huge irreversibilities would occur in heating the returned cold water by steam injection. No record has been found of any such system ever being built and there is little doubt that the expanders described would have had very low adiabatic efficiencies.

2.2 Sprankle's patents

In the early 1970s the newly emerging geothermal power industry in the United States aroused considerable interest. Dry steam from the California geysers could be expanded in conventional turbines, after cleaning out entrained and dissolved solids, to produce cheap electricity. However, the majority of geothermal reservoirs produce only hot water or very wet steam. These were and still are largely used for power generation purposes by flash expansion of the brine to intermediate pressures followed by the separation of the dry steam for power generation and reinjection of the residual hot water, as shown in Fig. 6. Both the flash expansion and hot water reinjection are thermodynamically wasteful and lead to poor resource utilization. Since exploration and drilling costs are high, electricity thereby obtained is relatively expensive. Sprankle (**8**) proposed to use such water or wet steam resources by direct two-phase expansion of the entire geothermal brine through a Lysholm twin-screw expander, as shown in Fig. 7. The expanded fluid would then be condensed and reinjected. A pure liquid resource thus uti-



(a) The system (b) The process

Fig. 5 Ruths' proposal for enhanced energy recovery from a steam accumulator. Virtually the entire hot water content could be expelled to produce work during the discharge period. This greatly increased the work capacity per unit volume of storage vessel but at the expense of less efficient usage of the stored steam

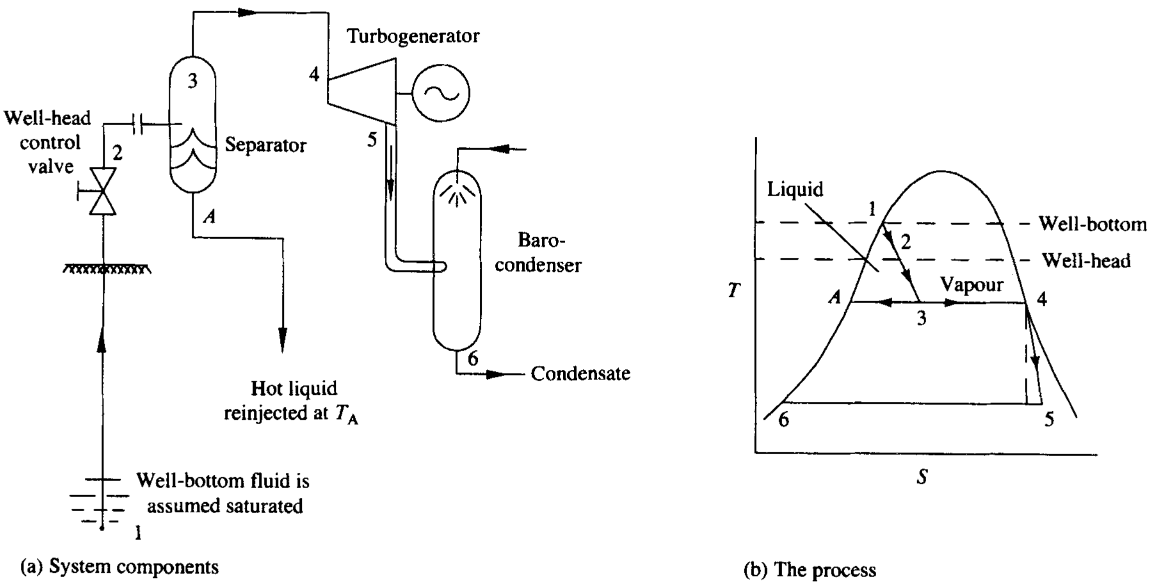


Fig. 6 A single flash steam system, showing the main components in (a) and the temperature–entropy diagram of the processes in (b). Note the loss in potential work from both the flash expansion and the reinjection of hot water

lized in effect constitutes an enormous Ruths system in which the working fluid is the geothermal brine heated by direct contact with its underground thermal reservoir. Unfortunately, although the patent was sound in concept, the Lysholm screw expander achieved maximum adiabatic efficiencies of only 53 per cent in small-scale tests and 68 per cent on a 1 MW machine. Overall, its efficiency of conversion of available energy to power therefore did not offer any improvement on a double-flash steam system while the huge sizes of screw expanders needed for large-scale power generation presented serious manufacturing problems.

2.3 Two-phase turbines

Sprankle's patents led to a wider study of 'total flow' systems of this type in an effort to make geothermal power from moderate temperature resources cost competitive with fossil-fuel-generating systems. Consequently, extensive research and development programmes were carried out on two-phase turbines at the Jet Propulsion (**9**) and Lawrence Livermore (**10**) Laboratories concurrently with the original Lysholm screw concept (**11, 12**). The turbine research led to the development of two-phase nozzles with adiabatic efficiencies of the

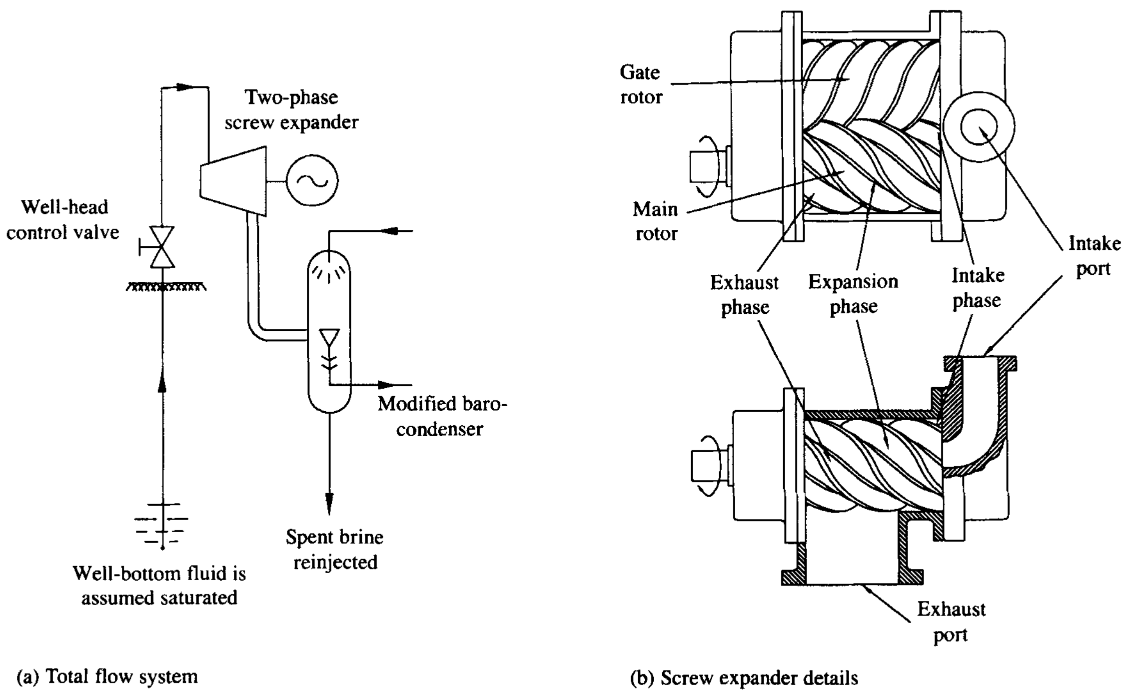


Fig. 7 Sprankle's original proposal for the 'total flow' of wet steam through a Lysholm twin screw expander. Ideally this constitutes the most efficient means possible of power recovery from such a source

order of 90 per cent. Nonetheless, at the end of the programme the predicted adiabatic efficiencies for large-scale turbines were only of the order of 65 per cent owing to the difficulties in the production of small enough water droplets in the wet vapour to respond sufficiently well to aerodynamic forces in the rotors.

2.4 The biphase turbine

An alternative hybrid system stemming from the two-phase turbine research was the biphase turbine (**13**) in which the separator of a normal flash steam plant was replaced by a high-efficiency two-phase nozzle followed by a rotary separator. There the liquid water total head was recovered to produce additional power in a pelton-type turbine rotating on the same axis as the separator, as shown in Fig. 8 The biphase rotary separator turbine was more satisfactory since its purpose was to augment the power of flash steam plant rather than to replace it, but the overall adiabatic efficiency of the separator unit was only of the order of 35 per cent. Though it has been demonstrated successfully at the Desert Peak geothermal plant (**14**) there are no further reports of its development.

2.5 Parallel screw expander and turbine

An ingenious combination of a screw expander to operate in parallel with a conventional steam turbine is reported (**15**), as shown in Fig. 9. In addition to the conventional inlet port at the high-pressure end of the rotors and exit port at the low-pressure end, additional exit ports are included at the high-pressure end face of the rotors to which the rotor spaces are exposed before rotation is sufficient for expansion to be complete. Partially expanded vapour is drawn from these additional ports. Since the motion of the rotors imparts a velocity to the fluid trapped within them, presumably reversal of the direction of flow, caused by the opening of these ports, leads to phase separation, with the water continuing in the same direction through the passages between the rotor lobes and the partially expanded steam reversing its direction to emerge from the screw expander and entering a conventional turbine. The residual water then expands further as the screws continue to rotate while the separated steam expands in parallel in the turbine. The additional steam formed in the screw expander is again separated out to enter the same turbine in a second stage, where it further expands together with the steam previously drawn off. No record has been found of such a system being built or tested, but it is doubtful if this would be better than a biphase separator turbine in series with a conventional steam turbine since there are likely to be relatively large throttling losses in the steam leaving the intermediate ports and complete separation of the liquid during this process is far from assured.

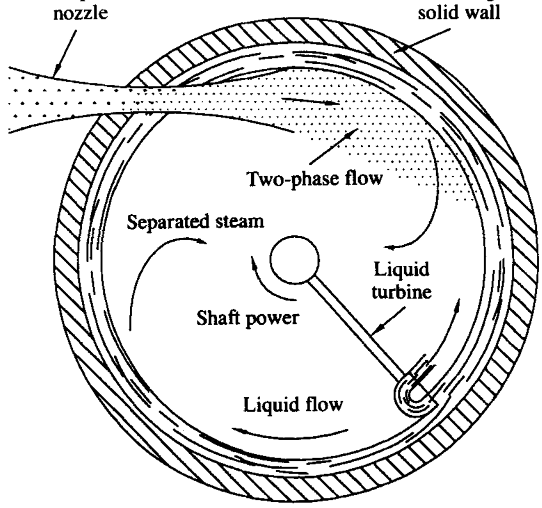


Fig. 8 The rotary separator biphase turbine. The principle of separation of water from steam on the rotary drum is shown together with the method of power recovery from the separated water. Note that the separator drum and turbine rotate coaxially

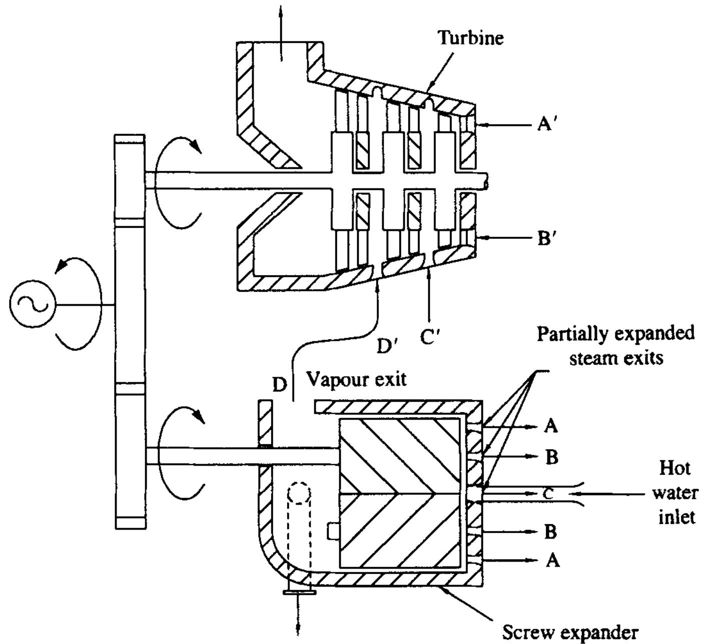


Fig. 9 A proposed improvement to the original total flow proposal. The screw expander operates in parallel with the steam turbine with partial withdrawal of steam from the screw expander at stages in the expansion process. The withdrawal of steam is required to allow for the enormous volume ratios associated with the two-phase expansion of water

2.6 Two-phase organic fluid turbines

In conjunction with the development of two-phase steam turbines, attention was given to the development of two-phase organic fluid turbines in an indirectly heated TFC system (**9**) identical to that described in Section 1.2. The results of tests on such a turbine were no more satisfactory than with steam and the programme of work on this appears to have been terminated.

Radial inflow turbines as two-phase process expanders of organic fluids in the chemical industry have been proposed by Swearingen (**16**) whose company, Rotoflow, has carried out a number of installations of such units. Adiabatic efficiencies claimed are about 67 per cent. Such units used to replace throttle valves can be cost effective since no additional heat exchangers are required. Their efficiency is too low to justify their use in a closed TFC cycle.

2.7 Screw expanders as throttle valve replacements

A variety of patents such as references (**17**) and (**18**) have been issued at intervals for the use of types of screw expander in place of a throttle valve in refrigeration systems. More recently, Taniguchi *et al.* (**19**) continued the work of researchers at Lawrence Livermore Laboratory to study the use of screw expanders as throttle valve replacements in large heat pump systems used for district heating. Experiments on small machines operating at tip speeds of up to 13 m/s were extrapolated to predict the performance of large machines at tip speeds of up to 50 m/s. From this it was concluded that adiabatic efficiencies of the order of 80 per cent were possible in machines with power outputs in excess of about 250 kW with Refrigerant 12 as the working fluid.

2.8 Work at City University

In mid 1981 the TFC system as a means of geothermal power recovery occurred to the author independently during the course of an examination of the feasibility of a liquid metal magneto hydrodynamic (LMMHD) power generation system. An initial feasibility study was carried out based on a careful examination of the properties of possible working fluids. It was concluded from this that water was not a suitable working fluid and that the greatest likelihood of success lay with the use of an organic working fluid expanded through one or two Lysholm machines in series, possibly followed by a final turbine stage if the working fluid reached a sufficient level of dryness. A literature survey then carried out surprisingly failed to reveal that this arrangement had been suggested previously. It was concluded that the poor results of screw expanders with water had led to their use with organic fluids being rejected. Accordingly a closed cycle test rig was constructed, containing Refrigerant 113 as a working fluid with a modified screw compressor used as an expander. Power outputs of up to 40 kW were obtained with peak adiabatic efficiencies of over 70 per cent at 25 kW in the first series of tests. From this it was concluded that on large-scale machines, adiabatic efficiencies of at least 77 per cent were likely. The remainder of this paper and its succeeding sections describe the basis of this approach and the subsequent development programme.

3 THE BASIS OF COMPARISON WITH ALTERNATIVE SYSTEMS

The first requirement for evaluating a new type of thermal power plant is to compare it with what is already available. In the case of low-grade heat applications the alternative is some form of Rankine cycle system. This may be built in a number of forms to include such features as steam or organic vapours as working fluids in single or multiple pressure arrangements or, more recently, in complex dual-fluid systems such as the Kalina water-ammonia configurations, which include vapour absorption and regeneration to change the concentration of the ammonia in the water in the condensation and heating stages.

A comprehensive comparison with all of these would be a very large undertaking. Moreover, there is no single basis for comparison that meets all requirements. The aim of this section is therefore not to make an absolute claim for the superiority of the TFC system but rather to illustrate its distinguishing features which may lead to criteria on how best to design it for a given application. Comparison is therefore only made with the simple Rankine cycle but with account taken of possible alternative working fluids. This will demonstrate both its most important characteristics and its potential for significant gains, even over more complex Rankine cycle systems.

3.1 Comparison based on performance

Consider a simple Rankine power plant which receives heat from a single-phase heat source. As shown in Fig. 10, there exists a choice between a high recovery of available heat but low cycle efficiency and a lower recovery of heat but higher cycle efficiency. The highest output will be obtained from the cycle with the greatest conversion efficiency.

Improvements to the conversion efficiency may be effected by either superheating the vapour or by choosing a working fluid with a sufficiently low critical temperature to enable the heating to be carried out at a supercritical pressure. These are shown in Fig. 11.

The common limitation of these forms of Rankine cycle and the TFC, as shown in Fig. 3, is the value of the minimum temperature difference or pinch point between the heating medium and the working fluid and that between the working fluid and the coolant. A comparison based on performance should therefore be based on the optimized conversion efficiencies of the TFC and Rankine cycle systems with identical minimum pinch point values.

3.2 Comparison based on cost

An alternative basis of comparison that has many advantages is that of cost per unit output. Difficulties arise here because the heat resource itself may have a variable value. Thus an industrial waste heat stream may be obtainable at little cost other than that of connecting it to the waste heat boiler. A geothermal resource, on the other hand, involves exploration and drilling costs, possible treatment costs if the brine has a high salt or silicate content and the cost of reinjection of the spent brine. In that case the cost should include both the power plant and the resource.

The optimum plant is that with the lowest cost of generation over its whole life. The factors that determine this may be complex. However, if the resource cost is ignored in the first place, then the main choice is between:

1. A system with a cheap heater and condenser obtained by means of a large pinch point and a high condensing temperature. The penalty for this would be reduced power output due to the lower maximum–minimum operating temperature difference of the working fluid within the cycle.

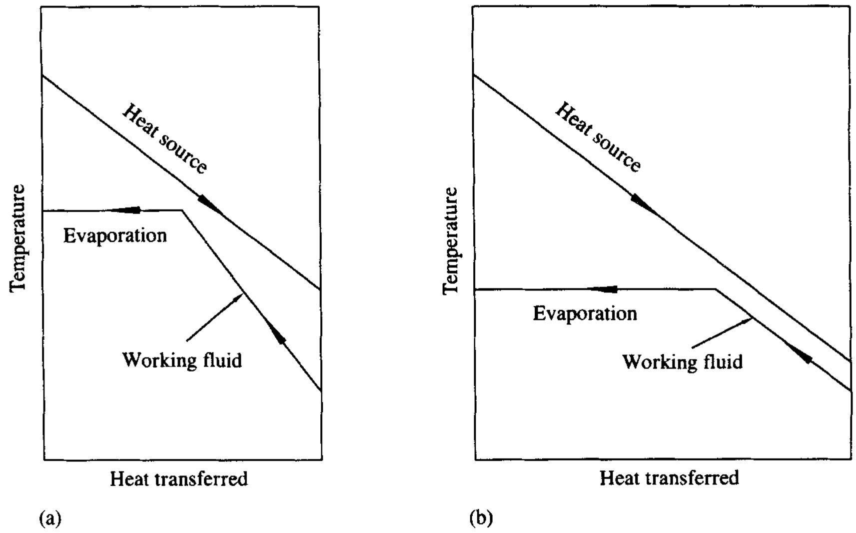


Fig. 10 Matching heat source to cycle in a simple Rankine cycle power plant without reheat. To obtain a high cycle efficiency, a high evaporative temperature is required. However, this leads to a poor recovery of heat, as shown in (a). Greater recovery of heat, as shown in (b), is only obtained by reducing the evaporation temperature, which leads to a lower cycle efficiency

2. A higher cost system with a relatively large heater and condenser with a minimal pinch point and a low condensing temperature to maximize the output. It should be noted that the power required to circulate the coolant through the condenser and cooling towers is roughly 2 per cent of the rejected heat. Thus for a typical low-grade heat system of 10 per cent cycle efficiency, 18 per cent of the system output is lost in coolant circulation. The low-cost heat-exchanger option therefore entails the additional performance losses due to greater auxiliary power requirements. Numerical studies carried out by the author showed that for all cycles considered, when account was taken of all auxiliary power requirements, given the limiting pinch point values in the heater and the condenser, the minimum cost per unit output for each system analysed differed negligibly from its value for maximum output. In that case the cost of the resource would only be significant if the system that yields the greatest output has the highest cost per unit net output.

4 ANALYSIS

From Sections 3.1 and 3.2 it may readily be seen that a proper comparison between Rankine and TFC cycle systems requires numerous cycle analyses, each involving a different combination of maximum fluid temperature and pressure, pinch point temperature difference, coolant temperature rise and, for air-cooled

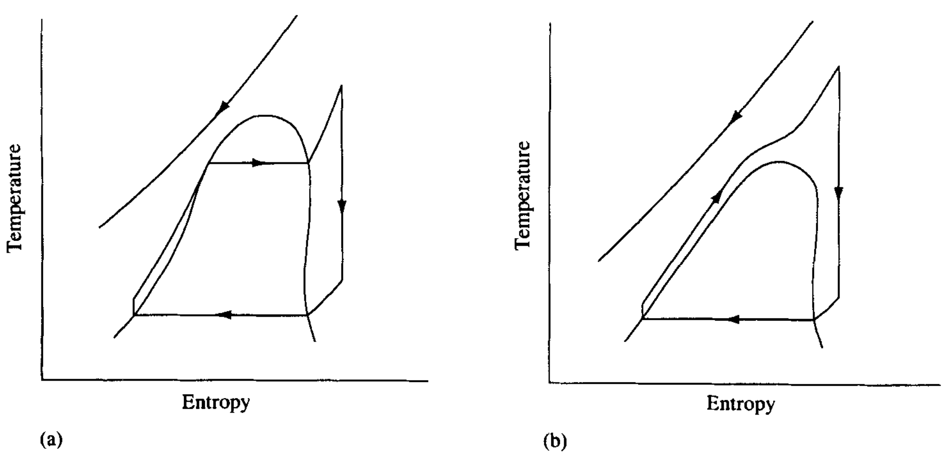


Fig. 11 Higher conversion efficiencies may be obtained from a simple Rankine cycle power plant by alternative fluid selection to obtain better matching between the heat source and cycle. The simple superheat cycle shown in (a) recovers more heat but at the expense of some higher temperature heat rejection. As shown in (b), a fluid with a low critical temperature may be used in a supercritical cycle. Desuperheating is thereby reduced but note that a constant pressure supercritical line kinks in the critical region

systems, cooling tower approach temperature. The output of each analysis must include estimates of both power output and cost and the whole procedure must be applied to each working fluid considered and each cycle. Each cycle analysis is too complex for simple mathematical modelling. Hence as a basis for both performance estimation and comparative system evaluation, there is no reliable alternative to a large computer program that performs these cycle analyses within the framework of some optimization routine.

The heart of the program must therefore be a method for estimating volatile fluid thermodynamic and transport properties at any required temperature and pressure in the liquid, vapour or two-phase states that is applicable to a wide range of fluids with maximum errors of the order of ± 2 per cent. Fortunately, this has long been the problem of chemical engineers and physical chemists and, especially over the past 30 years, a number of methods have been developed that may readily be applied to generalized cycle analysis. While a detailed knowledge of these methods is not needed, an awareness of the common characteristics of fluids on which they are based, and which govern working fluid selection, is essential.

4.1 General characteristics of working fluids

1. For most fluids, on the absolute scale of temperature, the ratio between the boiling point at atmospheric pressure and the critical temperature is roughly constant. The value of this is approximately 0.6.
2. There is a correspondence between reduced temperature, that is temperature as a fraction of its critical value, and reduced vapour pressure, which is roughly similar for most fluids. This correspondence is by no means linear since the vapour pressure increases towards its critical value much more rapidly as reduced temperatures exceed the value of about 0.95. Typically, though, there is a near-linear relationship between the logarithm of vapour pressure and the reciprocal of its corresponding absolute temperature, as shown in Fig. 12.
3. Most fluids have a critical pressure of between 30 and 50 bar. Accordingly, at atmospheric temperature, fluids with a high critical temperature, that is a high atmospheric boiling point, will have a low vapour pressure and vice versa.
4. The entropy of vaporization per mole is roughly constant for all fluids at their atmospheric boiling point (Trouton's rule). It follows from this that fluids with a low molecular weight, such as water, have a very high specific enthalpy in the vapour state. Accordingly their expansion through nozzles in the vapour phase is associated with large specific enthalpy drops and corresponding velocity changes.
5. The greater the number of atoms in a molecule, the more positive is the slope of the saturation vapour line of the fluid when plotted on temperature–entropy coordinates (**20**). This is shown approximately in Fig. 13.
6. Since the sonic velocity of ideal gas is $\propto (RT)^{0.5}$ and $R \propto 1/(\text{molecular weight})$, the sonic velocity of a vapour is roughly inversely proportional to the square root of its molecular weight.

The general similarity of all fluids in the vapour region has led to numerous attempts to derive relationships between pressure, volume and temperature and between saturated vapour pressure and temperature, which are common to all fluids. Apart from water, ammonia and other fluids with strong polar forces within their molecules, there have been great successes in developing equations of state and vapour pressure equations for this purpose. The bulk of these may be found in successive editions of reference (**21**). By means of these equations, reliable estimates of the thermodynamic properties of a wide range of working fluids may be made from little basic data beyond the critical temperature and pressure, the atmospheric boiling point, molecular weight and the constants required to express the specific heat capacity of the working fluid in its ideal gaseous state. Transport properties are not so easily generalized but little more than two experimentally derived constants are required for each property to cover a wide range of values. Where experimental data are not available, empirical equations may be used which, even when grossly inaccurate, enable heat-

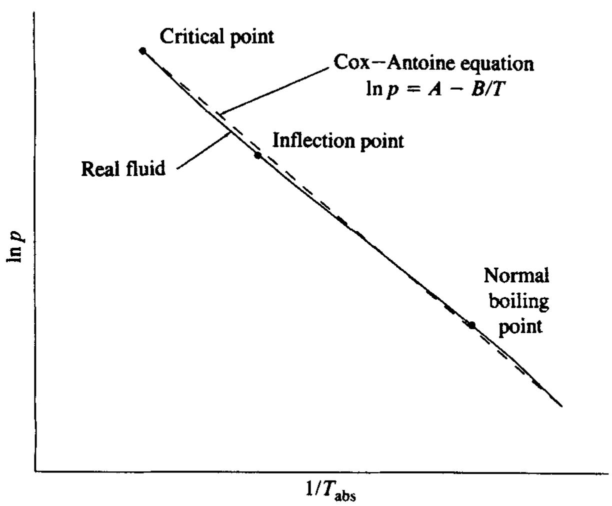


Fig. 12 The relationship between saturated vapour pressure and temperature for pure fluids

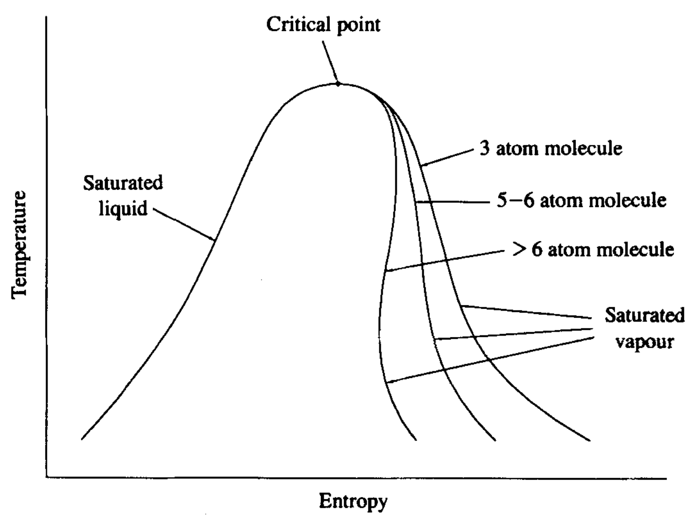


Fig. 13 The effect of molecule size on the saturated vapour pressure line slope. The greater the number of atoms in the molecule, the more positive the gradient

transfer coefficients and hence heat-exchanger sizes to be derived with relatively small error.

4.2 Method of Analysis

4.2.1 Fluid property estimation

Two sets of subroutines were prepared for the estimation of thermodynamic properties from specification of pressure, temperature and phase only. One of these, which served for the bulk of the fluids considered, was based on the Lee–Kesler equation of state and related vapour pressure equation. The other, which was used for ammonia and other polar fluids including, to a good first approximation, water, was based on the Martin–Hou equation of state together with the Cox–Antoine vapour equation. Two sets of sub-routines were also written for the estimation of transport properties. The first was based on experimentally derived values, where available, and simple curve-fitting equations. The second was used when no experimentally derived values could be found and was based on the recommended equations given in reference (**21**). In both sets, continuous transference from liquid to vapour property values in the supercritical region was obtained by means of a smoothing function developed by the author. These sets of sub-routines together with a data bank of over 120 fluids are contained in a package called THERPROP.

4.2.2 Cycle analysis

Sets of sub-routines were written to estimate the fluid properties at all key points on either the TFC or simple Rankine cycle with variable input values given only to the maximum fluid temperature and pressure, heater and condenser pinch point values and temperature rise of the coolant. Fixed values were input for the expander, feed pump and generator efficiencies together with auxiliary pump, fan and motor efficiencies, pressure loss allowances through the heat exchangers and cooling tower design parameters, where required. On completion of a cycle analysis the mass flow of working fluid and hence the main power output and auxiliary power consumption were estimated by means of an energy balance between the heating source mass flow and temperature drop and the enthalpy gain of the working fluid.

Separate sub-routines were appended, which contained empirical equations for typical heat transfer coefficients for shell and tube-type heaters and condensers based on fluid thermal properties and input values of typical fouling factors and inner to outer surface ratios, to allow for extended surface tubing where required. From these and the property values determined in the cycle analysis routines, the heat-exchanger surface areas were estimated.

Additional sub-routines were included to give first estimates of cooling tower size and power requirements which were derived from manufacturers' data and forced draught cooling tower design guidelines.

4.2.3 System costing

By means of data obtained from chemical engineers' cost estimation procedures, described in references (**22**) to (**30**), graphs were prepared on component prices. These were periodically updated by rollup factors based on government statistics on price indexing (**31**). Algorithms suitable for computer usage were then derived from them and included in a cost estimating subroutine. The entire system cost was then estimated as the sum of its component costs multiplied by an installation factor of between 1.6 and 2.0. The cost per unit output was then estimated as the total installed cost divided by the net power output after deduction of all auxiliary fan and pump requirements.

4.2.4 System optimization

The set of sub-routines described in Sections 4.2.1 to 4.2.3 may be used to produce a good estimate of the power output and cost of a Rankine cycle or TFC system, given the nature and capacity of the heating and cooling resources and the temperature and pressure of the working fluid at the heater exit and during condensation. This set was called from a multi-variable minimization software package known as MINUIT (**32**) to determine the cycle conditions that would produce the optimum output. The optimum could be the maximum power output, the minimum cost per unit output or a weighted mean of these two factors. As mentioned in Section 3.2, it was found that while the minimum cost per unit output was found to differ only marginally from its value at maximum power output, this generally insignificant gain was sometimes achieved for a very large loss of power output. The minimum cost results were therefore ignored and only those corresponding to maximum power output are presented here.

4.3 Input data

A series of analyses of both simple Rankine and TFC systems was carried out to estimate the power recoverable from a stream of hot water with initial temperatures of from 100 to 200 °C based on the following assumptions:

1. All systems were water cooled with the aid of forced draught cooling towers in which the average wet bulb temperature was 15 °C and the coolant return temperature to the condenser was 20 °C.
2. Expander adiabatic efficiencies assumed were 80 per cent for Rankine systems and 77 per cent for TFC systems.
3. The generator and all fan, pump and motor efficiencies together with internal and external pressure loss factors were taken as the same for all cases and based on published data for comparable applications.
4. All comparisons are based on maximum recoverable net power at any source temperature.

Analyses were carried out using water, R12, R113 and *n*-pentane as working fluids. Water was chosen as the benchmark with which all alternatives must be compared. R113 was taken as a typical chlorofluorocarbon with known good thermodynamic properties for organic Rankine cycle (ORC) systems and a sufficiently high critical temperature (213 °C) to enable TFC simulations within the saturation envelope to be possible over the entire range of heat source temperatures considered. R12 ($T_c = 112\,°C$) was chosen in order to make comparisons with the Rankine cycle operating under supercritical conditions. *n*-Pentane was used as a typical light hydrocarbon with somewhat similar pressure–volume relationships and critical temperature (196 °C) to that of R113.

5 RESULTS OF THE ANALYSIS

5.1 Gross power output

Maximum gross electrical power output estimates are shown in Fig. 14. Most significantly, it may be seen that the TFC output exceeds that of any simple Rankine cycle over the entire temperature range. The largest relative gains are at lower temperatures where improvements of the order of 80 per cent are possible. It is interesting to note that the output from the TFC system is almost independent of the working fluid and though water appears to yield less power, the differences are of the same order of magnitude as possible errors in the equations of state used.

The relative independence of the TFC system performance to the working fluid, its superiority to all forms of simple Rankine cycle and the significant effect of working fluid choice on Rankine cycle system performance are all due to the same factor, namely the matching of the various cycles with the heat source. Throughout the heating process of the TFC system there is an almost constant temperature difference between the heating working fluid and cooling heating source due to heat transfer from liquid to liquid. The heat-recovery efficiency is therefore almost identical for all the fluids and nearly a maximum. Hence the gross output hardly varies with the working fluid.

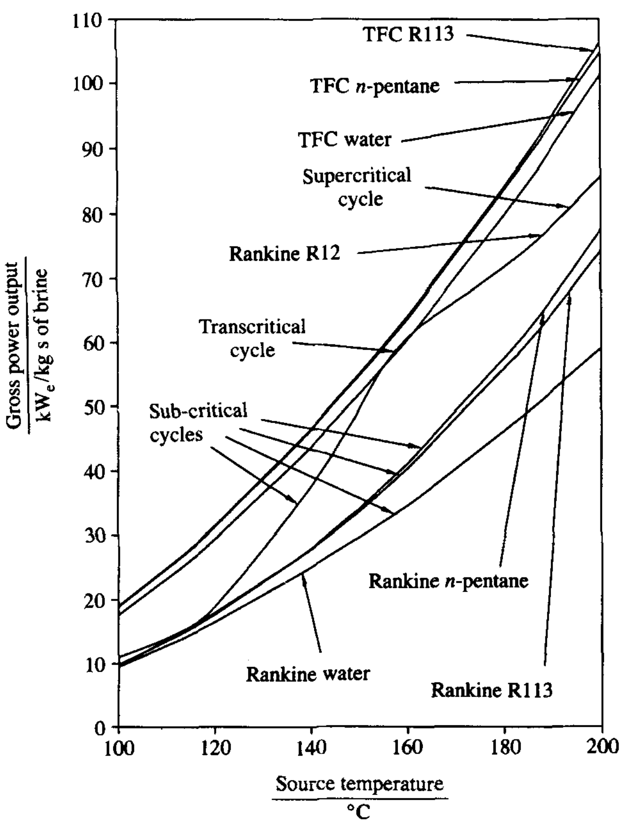


Fig. 14 The effect of working fluid choice on gross power output. Due to close temperature matching, the TFC power output is hardly altered by fluid choice but for Rankine cycles, where matching depends on the working fluid, there is a large variation

The optimum cycles for Rankine plant using water, R113 or *n*-pentane are all subcritical throughout the temperature range considered. Due to evaporation, subcritical Rankine cycles have significant heating at constant temperature. This leads to poor temperature matching in the boiler between the working fluid and the heat source and hence a lower recovery of heat or a lower cycle efficiency, as already shown in Fig. 10. The greater the ratio of heat transfer in single phase to that in evaporation, the more efficient is the heat recovery and hence the higher the power output. Hence water, with its very low molecular weight and its uniquely high enthalpy of evaporation, which is the greatest of any known fluid, yields the smallest output. Pentane (MW = 72) has a much higher enthalpy of vaporization than R113 (MW = 187) but it has a liquid specific heat more than double that of R113. Thus they have fairly similar matching characteristics with the heat source and nearly equal power outputs.

At lower resource temperatures the optimum R12 Rankine cycle is subcritical and the gross output from it is close to that from the other working fluids. As the resource temperature increases there is a gradual transformation to operation in the supercritical mode. The relative gain for R12 over alternative fluids is greatest at resource temperatures of between 150 and 160 °C, where there is an improvement of nearly 70 per cent over steam and the cycle is transcritical in operation; that is the maximum working fluid temperature and pressure are close to the critical point values and the vapour expands very close to and passes partly within the saturation vapour envelope before attaining dry saturated vapour conditions at the expander exit. The matching between the cycle and the source under these conditions then approaches that of the TFC cycle. Consequently, the gross output is of the same order bearing in mind that the assumptions of both two-phase expander and dry turbine efficiencies could be a few per cent in error.

At higher values of resource temperature, the optimum R12 Rankine cycle becomes fully supercritical. Matching between the source and sink is then inferior to that of the TFC due to the 'kink' in constant pressure lines near the critical point, as already shown in Fig. 11. Hence the TFC system then shows greater relative gains.

5.2 Auxiliary power and net output

Power losses associated with circulation of the working fluid through the heat exchangers are relatively small for all systems. Hence, in all cases the main auxiliary power requirements are for the feed pump and the cooling system. These are shown in Figs 15 and 16. For the feed pump power the essential features are:

1. The TFC feed pump power requirement is larger than that for the subcritical Rankine cycle. *This is* because, given the same fluid pressurization, more power is recoverable from expanding a dry vapour than from expansion through the two-phase region. The TFC feed pump work is therefore a greater percentage of the recoverable power in expansion.

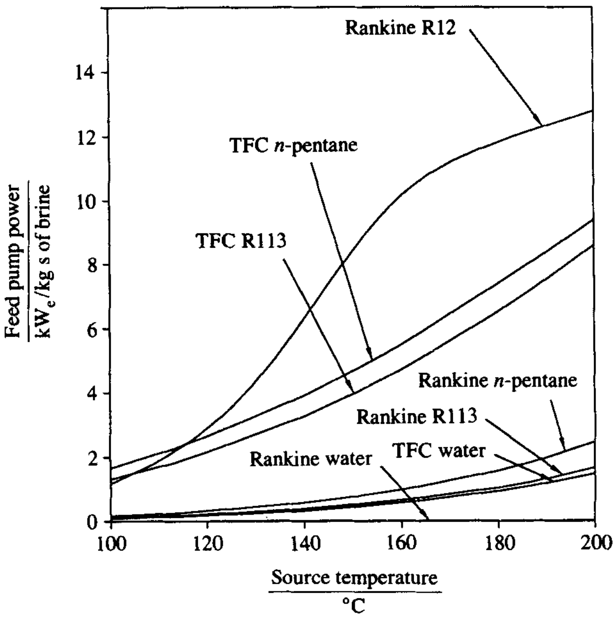


Fig. 15 Comparative feed pump power requirements. Because the specific enthalpy change in expansion is smaller for the TFC system, its feed pump power requirement is greater than that of the subcritical Rankine cycle. The supercritical cycle absorbs the highest power because the pressure rise is much larger

2. The transcritical and supercritical Rankine cycle feed pump power requirement is greater than that for the TFC. This is because of the much higher pressures required in the former systems.

Note that water, with its very high latent heat, and hence vapour specific enthalpy, requires the smallest feed pump work both in Rankine and TFC systems.

TFC system output exceeds that of the Rankine system mainly because more of the available heat is used rather than because of improved cycle efficiency. Hence it rejects more heat and accordingly its cooling system requires more power. Differences in the power requirement are more due to the cycle than the working fluid and it may be noted that the supercritical Rankine cycle, with its better matching, requires more power than the subcritical cycle.

The differences between the cycles and working fluids on a net output basis is shown in Fig. 17 and this is the most meaningful means of comparison. Here it may be seen that even allowing for uncertainties in the expander efficiency and in estimating the different fluid properties the TFC system shows significant gains over all types of simple Rankine cycle system with potential gains of over 85 per cent at the lower temperatures.

Most significantly water, due to its lower feed pump and cooling system power requirements, is better than it appears to be from gross output studies only and yields the highest output in TFC applications, although not by a very significant amount.

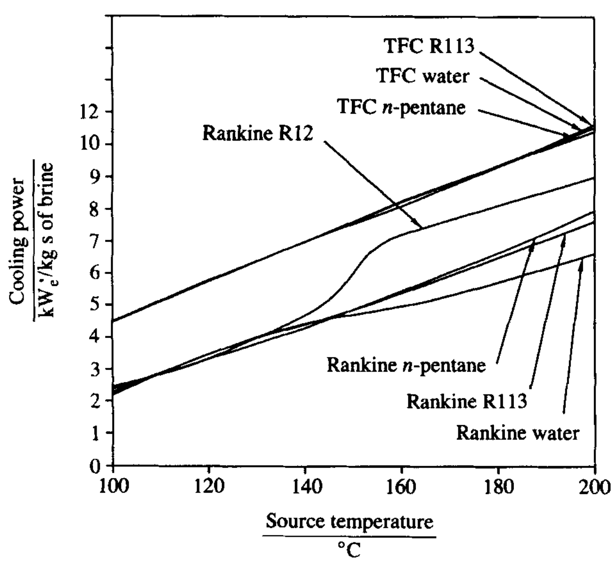


Fig. 16 Comparative auxiliary power requirements. Note that these are quite significant, especially for low resource temperatures. The TFC requires more power than the subcritical Rankine cycle mainly because it utilizes more of the available heat

5.3 Expander design considerations

Since the feasibility of an efficient two-phase expander is implicit in the TFC system proposal, the criteria affecting its design must be included in its appraisal.

5.3.1 Specific enthalpy drop

Figure 18 shows the specific enthalpy drop through the expander for all the systems considered. The main factors to be noted are the very much higher values for water-steam over the organic fluids, the lighter hydrocarbon pentane having roughly double that of the halo-

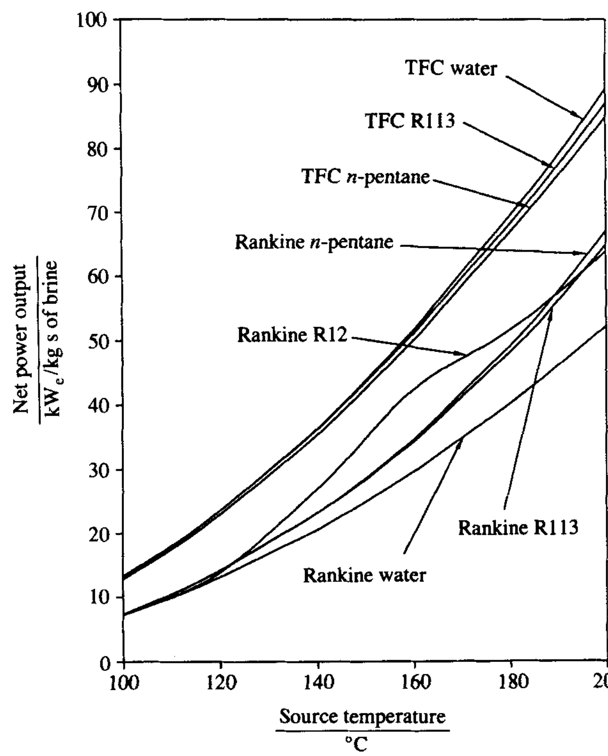


Fig. 17 Comparative net power output. This indicates the superior output of the TFC system over all types of simple Rankine cycle. Note that the TFC net output is only slightly affected by fluid choice

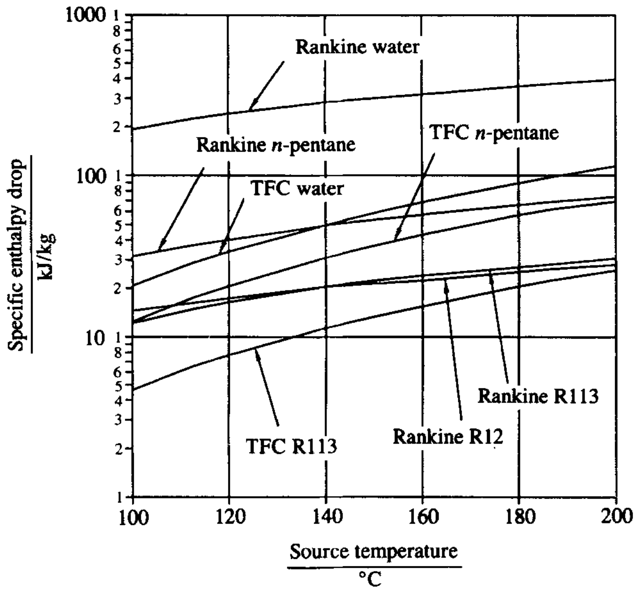


Fig. 18 Comparative specific enthalpy drop. This is dependent on relative molecular mass but for the same fluid it is much less in the TFC than in a Rankine cycle

carbons and the extremely small values associated with the halocarbon TFC expansion, which have an approximate average value of only 10 kJ/kg.

Typical steam turbine exhaust velocities are of the order of 100–150 m/s, which imply exit losses of 5–11 kJ/kg. These are far too high to permit efficient expansion in any form of organic fluid system. Inevitably organic fluid expanders must run at lower speeds. To minimize exit losses light hydrocarbons are clearly preferable to halocarbons. At the lower resource temperatures, organic TFC expanders have such a low specific enthalpy drop that there could be difficulties associated with efficient turbine design quite apart from those associated with separation of the liquid and vapour streams. Positive displacement machines of the Lysholm twin screw type therefore have inherent advantages under those conditions.

5.3.2 *Exit volume flowrate*

Figure 19 shows the expander exit volume flowrate for all the systems considered. It is interesting to note that there is no great difference between Rankine and TFC expanders for any given fluid but steam-water expanders have about ten times the volume flowrates of organic systems.

To minimize the size and hence the cost of steam turbines, higher condensing temperatures are accepted in industrial and geothermal installations, 120 °F (49 °C) being a typical value. Even with this performance restriction, the maximum velocities possible in positive displacement machines are so much less than in turbomachines that the rotor diameters required for a 10 MW total flow expander were estimated to be 28 ft (**11**). The maximum rotor diameter currently manufactured for screw compressors is of the order of 1 m. The two-phase expansion of an organic fluid in a machine of this size could yield an output of 1–5 MW when condensing at about 30 °C, the exact value depending on the fluid choice and the permitted tip speed. This is in the acceptable range of outputs for industrial and geothermal systems. Clearly, therefore, the use of organic fluids makes the design of a practical two-phase expander more feasible and here hydrocarbons have advantages over the halocarbons in minimizing both exit losses and size.

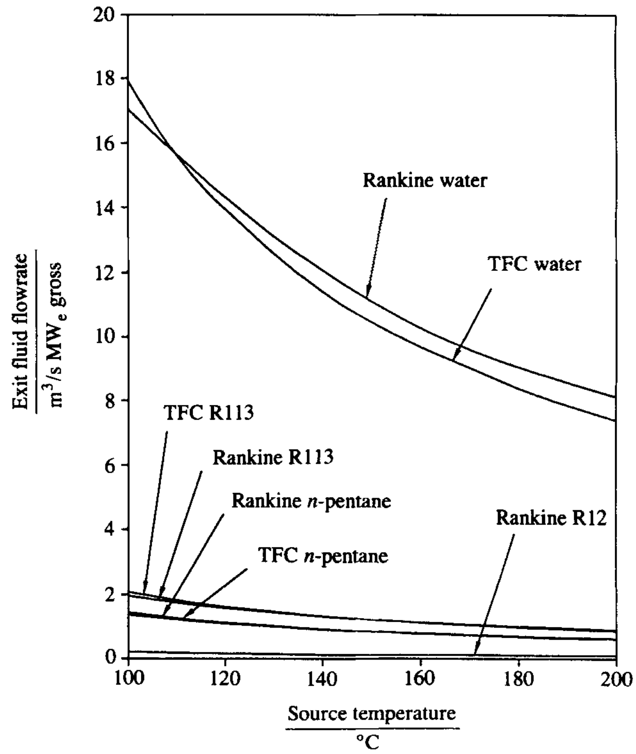


Fig. 19 Comparative expander exit volume flowrates. This shows the relatively large size of steam expanders

5.3.3 *Volumetric expansion ratio*

A feature of two-phase expander design unfamiliar to conventional power plant designers is the volumetric expansion ratio required for two-phase expansion from the saturated liquid condition. The estimated values are shown in Fig. 20 while for comparison the corresponding values for the Rankine cycle expanders are shown in Fig. 21. Water–steam expansion is roughly two orders of magnitude greater than that of saturated steam. The enormous values required, even at elevated condensing temperatures, make it difficult to visualize how such expansion could be achieved in positive displacement machines without a very large and wasteful preflashing of the fluid before entering the expander. Here the advantage of organic fluids is self-evident, with reductions of the order of 30 : 1. Again, the light hydrocarbons, with liquid densities of the order of half those of the refrigerants, appear to be the best.

5.4 Cost considerations

The relative costs of the various systems are shown in Fig. 22. Before considering these too closely, it should be noted that they are based on a set of common criteria. This includes the assumption that all the systems operate on a closed cycle and precludes the probability that where water or steam is the working fluid, it will be

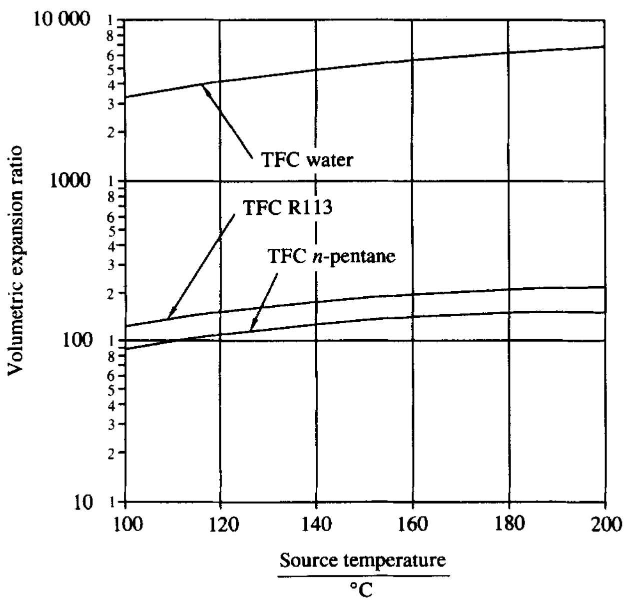


Fig. 20 Comparative volumetric expansion ratios in TFC systems. There are clearly great advantages in using light hydrocarbons

a process fluid where a boiler or heater would be unnecessary and there would be some small additional thermodynamic advantage due to the absence of a temperature difference across its surfaces. In all cases, cost is primarily ascribed to size. Although efforts have been made to adjust the derived values to likely real system costs the relationship is at best approximate and differences between the various systems of 10 per cent or less are probably not too significant. On this basis the following conclusions may be drawn:

1. A water TFC system is prohibitively expensive. This is due both to the large specific volume of low-pressure steam and the low attainable fluid velocities in positive displacement machines. Hence very large machine sizes are needed, as was already predicted for total flow systems (**10**).

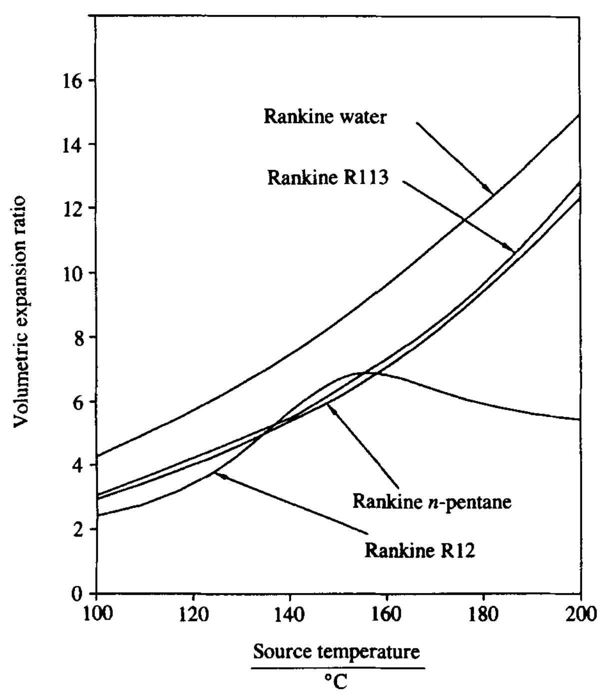


Fig. 21 Comparative volumetric expansion ratios in Rankine cycle systems

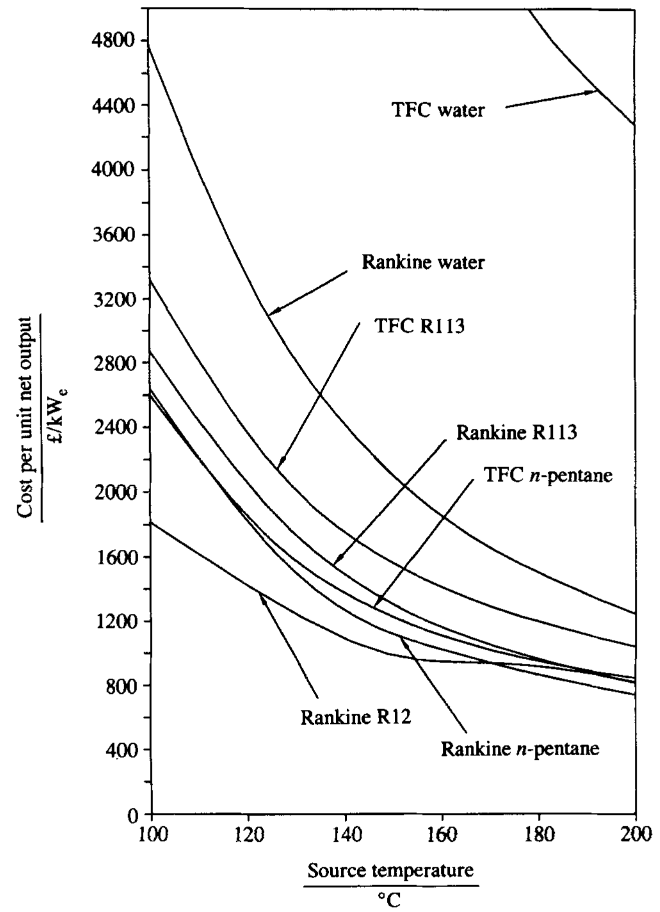


Fig. 22 Estimated comparative costs of TFC and Rankine cycle systems. A water TFC system is clearly too expensive, but the difference between ORC and TFC costs is otherwise not too significant

2. Steam systems are relatively more expensive than those using organic fluids. This is due to the cost of large steam turbines apparently exceeding that of the larger and more expensive heat exchangers required for the organic fluids. The need for higher condensing temperatures in steam plant used for industrial waste heat and geothermal applications, in order to minimize cost, is thereby confirmed.
3. TFC costs are overall about the same as those of ORCs. This is because the much larger heat-exchanger costs are more or less compensated by the greater output thereby obtained.

The cost advantage of the R12 Rankine system up to the transcritical condition is mainly due to the much lower critical temperature of this fluid resulting in higher condensing pressures and hence much smaller components. Alternative hydrocarbons such as isobutane (T_c = 135 °C) confer similar advantages on TFC systems at lower resource temperatures.

6 FURTHER FACTORS AFFECTING WORKING FLUID CHOICE

Apart from considerations of thermodynamic efficiency, cost and expander design, there are other factors to be considered in the choice of working fluid for a TFC system, and indeed any system using working fluids other than water. The most significant of these are given in the following sub-sections.

6.1 Thermal stability

Organic fluids are liable to decomposition on heating and few of the halocarbons exhibit long-term stability when heated continuously to temperatures above approximately 120 °C, especially when contaminated with lubricating oil, which is almost inevitable in power plant. In contrast, the saturated hydrocarbons are far more stable and all the alkanes likely to be suitable, including of course *n*-pentane, are fairly inert up to temperatures of at least 200 °C.

6.2 Operator and environmental hazards

A major potential hazard with working fluids other than water is the effect of external leakage on the power plant surroundings. The halocarbons were originally developed for refrigeration systems on account of both their low toxicity and flammability. In recent years, concern with the potential effects of chlorine compounds on the ozone layer has led to the planned withdrawal of R12 and R113 from production and latterly from use.

Hydrocarbons have no long-term environmental hazards and their toxicity levels are of the same order as the halocarbons. However, they do have more serious flammability and explosion hazards. Machinery containing them must therefore be not only totally sealed but also flame proofed. Nonetheless, Rankine cycle systems containing hydrocarbon working fluids have been widely deployed during the past twenty years with no known record of damage caused by leakage.

In this respect, definitions of what constitutes a hazard may be subjective. Thus, until recently, vehicle insurers refused to allow light hydrocarbons as replacement working fluids for R12 in motor vehicle air-conditioning systems. However, cars carry many gallons of petrol in unsealed tanks while the air-conditioning systems require less than 200 ml of fluid contained within totally sealed systems.

7 CONCLUSIONS

Detailed analysis of the TFC system as a means of recovering power from hot liquid streams in the 100–200 °C temperature range has shown that:

1. Provided that two-phase expanders can be built to produce outputs of the order of 1 MW with adiabatic efficiencies of greater than 75 per cent, net outputs from such a system are 10–80 per cent greater than from any simple Rankine cycle system, including the proposed transcritical ORC system, which comes nearest to it in theoretical performance.
2. Due to near ideal temperature matching between the working fluid and its heating stream in the primary heat exchanger, the net output of a TFC system is almost independent of the working fluid. However, the choice of working fluid profoundly affects the system cost and it is fairly clear that, from this consideration, water is not suitable as a working fluid.
3. Considerations of system cost, expander design, thermal stability and environmental protection all indicate that light hydrocarbons are the preferred working fluids for a TFC system. However, special precautions, including flame proofing of the whole system and possible provision of a flare arrangement, are essential when such fluids are used.
4. Preliminary comparative cost estimates indicate that TFC systems can be built for roughly the same cost per unit net output as alternative ORC systems and less than steam systems with equal condensing temperatures.
5. Positive displacement expanders of the twin screw Lysholm type are the most suitable for the TFC system on account of the high liquid content and very low specific enthalpy drops associated with two-phase organic fluid expansion. At higher source temperatures and outputs, where the working fluid may leave the expander as dry or nearly dry vapour, the final stage of expansion involving the bulk of the power output may be achieved in a radial inflow type of turbine installed downstream and in series with a higher pressure screw machine.

ACKNOWLEDGEMENTS

All who have attempted to develop unusual ideas will be familiar with the difficulties associated with innovation. Most significantly there is the task of funding. For this, special thanks are due to Peter Kalms, who first backed it and with whose continued support further help was obtained from the DTI, Southern California Edison, ETSU and the SERC. When the money ran out, the support of the City University and especially the former and present Heads of Department, George Done and David Thorley, is unforgettable. For the studies described in this section of the work, due acknowledgement must be made to the contribution of the author's former colleagues, Mohammed Abubaker and Bennie Tiefenbrunner, who developed the costing and optimization procedures described. Thanks are also due to Bryce Burnside of Heriot Watt University for the supply of sub-routines and data on the Martin–Hou equation of state.

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