An Improved System for Power Recovery from Higher Enthalpy Liquid Dominated Fields

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ABSTRACT

It is shown that the Trilateral Flash Cycle (TFC) system approximates fairly closely to the ideal requirements for power recovery from liquid resources. A limitation, to this, which has hitherto prevented such a cycle being used in practice, is the lack of availability of suitable two-phase expanders for power outputs in excess of approximately 1 MWe.

At resource temperatures in the range of 170 – 200°C, working fluids suitable for this cycle can perform the expansion process across the entire saturation envelope so that they start as high pressure saturated liquid and end at low pressure, as dry or even superheated vapour. Existing types of turboexpanders can then be used to produce high power outputs with good efficiencies by carrying out the expansion in two stages. In the first stage, the working fluid enters a radial inflow turbine, from which approximately one third of the output is recovered by two-phase expansion, and the working fluid leaves it 80% dry. The liquid can then be removed by a simple gravity separator and used to preheat the condensate. The separated dry vapour then enters a conventional turbine, of the axial or radial inflow type for the second stage of expansion, in which approximately two thirds of the power is recovered.

Detailed numerical studies have shown that, this simple system, which requires only two heat exchangers and known components of proven efficiency, can recover approximately 60% of the ideal maximum power.

Introduction

It is well known, that when the only utilisable form of energy recoverable from a liquid resource is mechanical or electrical power, the ideal heat engine for this purpose should operate on the equivalent of a succession of infinitesimal Carnot cycles, each with a decreasing source temperature. Taken in aggregate, these form a single cycle, as shown in Figure 1. Because of its shape, this cycle has been described as an "ideal trilateral cycle".

For such a cycle, the maximum efficiency for the recovery of work from the available heat is given by the ideal conversion efficiency:

$$\eta = 1 - \frac{T_2 . \ln \dfrac{T_1}{T_2}}{T_1 - T_2} \tag{1}$$

The same equation can also be derived from consideration of exergy change, without invoking infinitesimal Carnot cycles and their integration over a given temperature range.

As can be seen, the ideal trilateral cycle recovers all the available heat by permitting the heat source to be cooled down to the temperature of the environment. At the same time the working fluid in the cycle is able to attain, at least at one point, the maximum heat source temperature and thus at every stage, heat is converted to work as efficiently as possible.

A practical approximation to this cycle, which has been described previously both by the authors in refs (1-3) and other

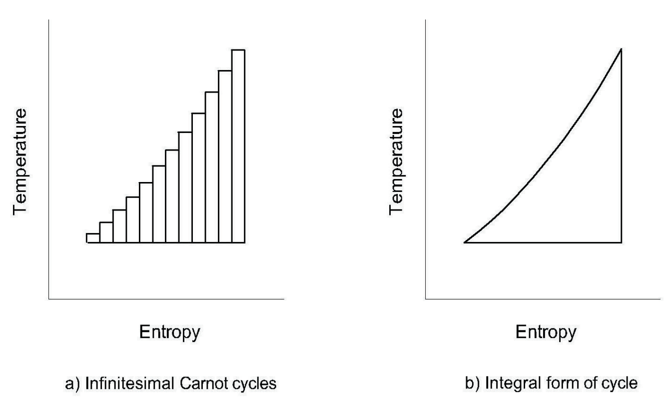
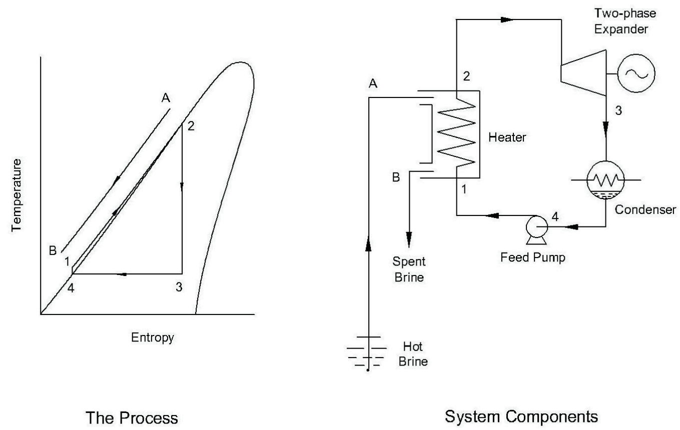


Figure 1. Ideal trilateral cycle.



investigators, is shown in Figure 2. This has been described as a Trilateral Flash Cycle or TFC system.

Figure 2. The TFC System.

Although this system has been considered for over 25 years and there is a record of its proposal as an open cycle, as long ago as 1920 (4), to date, no large scale demonstration unit of it is known to have been built. This is because of the lack of suitable two-phase expanders with adiabatic efficiencies approaching those of dry vapour turbines.

The authors have shown (5) that twin screw machines can be built cost effectively with adiabatic efficiencies approaching 80%. However, when operating as two-phase expanders, they need to rotate with relatively low rotor tip velocities. Consequently, they are large, compared to turbines and size restrictions in the manufacture of their rotors limit their unit power output to about 1MWe.

Although axial flow turbines are not suitable for two-phase expansion, progress has been made in the development of radial inflow turbines of the Francis type for this function. These are used extensively in the chemical process industries and have now been improved to the extent that they can be built to expand fluid, entering them as saturated or subcooled liquid and leaving as two-phase wet vapour, with adiabatic efficiencies of 75-80%. Such machines have a major advantage over screw expanders. This is that they are far more compact because they operate with high fluid velocities and rotor tip speeds. Also large turbine rotors are much easier to manufacture than large screw rotors. Hence far higher powers can be obtained from them than from screw expanders, within the normal limits of manufacturing practice. Nonetheless, turbines of this type have a disadvantage compared to a screw expander. This is that, while they will operate effectively when the working fluid enters either in the pure liquid phase or as dry vapour, they are unsuitable for the admission of two-phase mixtures of liquid and vapour.

The volume ratios of expansion from the liquid phase are much higher than those associated with the expansion of dry vapours but neither screw expanders nor radial inflow turbines can maintain high efficiencies when these exceed approximately 25:1. This limits their use as single stage expanders to maximum working fluid inlet temperatures of about 110°C. At higher temperatures, less volatile working fluids are required for the working fluid to enter the expander as saturated liquid and the volume ratios of expansion then may exceed 100:1.

It follows that for liquid resources with temperatures in excess of approximately 120°C, for efficient operation, a two stage expander is needed for a TFC system but, since the working fluid leaves the first stage of expansion substantially wet, it cannot be passed directly into a radial inflow turbine to complete the expansion in the second stage.

The use of liquid dominated resources and the continuing development of Hot Dry Rock studies around the world, has now led to the possibility of the realisation of relatively large liquid resources with initial temperatures in the 180°-200°C range from which binary vapour plant may be required to recover power. At these temperatures, simple ORC systems are not so efficient and the method recommended for their use under these circumstances is to build two such systems to operate in cascade. The first would operate over a higher temperature range and the condenser of this unit would act as the evaporator of the second unit with different working fluids in each closed loop. Alternatively, Kalina type systems, which require at least three heat exchangers, may be suitable. In the light of the relative complexities of these systems, the authors re-examined the possibility, first considered some twenty years ago, of using a TFC system for power recovery from higher temperature resources.

At resource temperatures in the 170°-200°C range, a suitable working fluid for a TFC system would be either n-Pentane or a mixture of it with neopentane (2). With such a working fluid, the expansion process would involve a volume ratio of expansion of the order of 160:1. This would require a two stage expander. The problem, therefore, lay in the design of a second stage expander, which could admit wet vapour leaving the first stage. To be acceptable, an alternative to a twin screw machine was needed in order to avoid the need for a large number of units in parallel.

One possible method of overcoming this is shown in Figure 3. Here, at the end of the first expansion stage, the fluid leaving the expander enters a separator. Separators for such a purpose are in common use in the chemical and process engineering industries and are normally of either the gravitational or centrifugal type. In view of the fact that exit velocities from the first stage two-phase expander are of the order of only 10m/s, the gravitational separator is preferable because the pressure

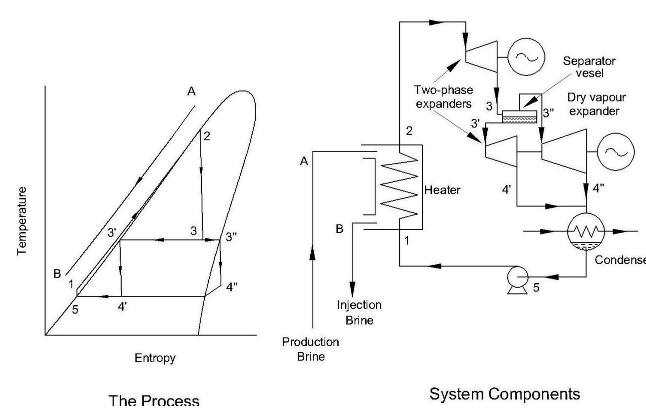


Figure 3. The TFC System With Divided Second Stage Expansion.

loss associated with the separation process is very small. The second stage of expansion could then be carried out by two radial inflow turbines in parallel, one admitting pure liquid and the other dry vapour. However, a disadvantage to this is that the volume ratio of expansion of the second stage turbine, admitting pure liquid, would still be very high and therefore that machine would have a poor adiabatic efficiency.

When considering working fluids such as pentane, it is, however, important to take into account that due to the large number of atoms in its molecule, its saturated vapour line, plotted on temperature-entropy coordinates, has a strongly positive gradient. Thus, as shown in Figure 4, at higher expander admission temperatures, it is possible to carry out an adiabatic expansion, starting with saturated, or even slightly subcooled liquid, which finishes with the working fluid leaving the expander as dry or even superheated vapour.

On this basis, it appeared possible that by the right selection of the intermediate temperature at which the fluid is separated, the bulk of the fluid leaving the first stage expander would be vapour and hence, the contribution of the second stage expander admitting the separated liquid, might not be large.

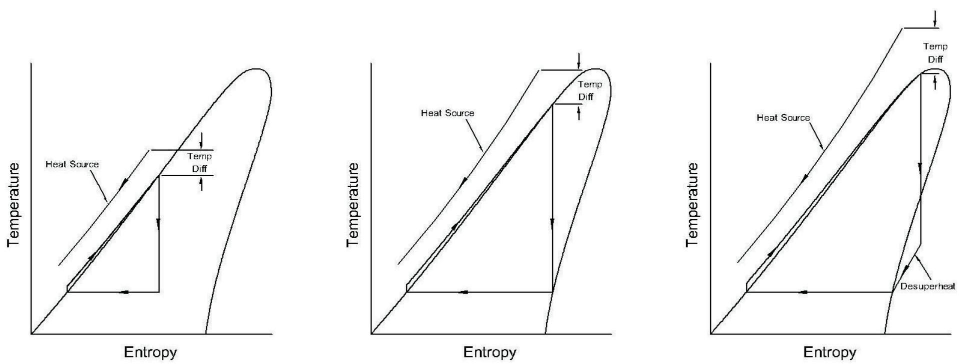


Figure 4. TFC System Expansion Changes With Increasing Resource Temperatures.

To obtain an appreciation of how this arrangement would perform, a design study was therefore carried out assuming brine temperatures and flows that might be typical of a single well in a liquid dominated or HDR resource.

A Study of a TFC System with Divided Second Stage Expansion

Assumptions:	Brine Flow Rate	75kg/s
	Brine Inlet Temperature	190°C
	Heater Pinch Point Temperature Difference	5°C
	Available Cooling Water Temperature	20°C
	Cooling Water Temperature Rise in Condenser	5°C
	Condenser Pinch Point Temperature Difference	3°C

In viewexpansion must begin from the saturated liquid condition, the most suitable working fluid for this case is n-Pentane, which has a critical temperature of 196.6°C.

A complete cycle simulation program, with multivariable minimisation routines included, to determine the optimum operating conditions for the system, was run with these input parameters, to determine the flow conditions for optimum output. These were as follows:

Brine Exit Temperature from the Heater	38°C
Pentane Flow Rate	116 kg/s
Expander Inlet Temperature	175°C
Condensing Temperature	28°C

Under these conditions, allowing for drive motor and pump efficiencies, it was estimated that

Feed Pump Power Input	600 kWe

Enquiries were then made to radial inflow turbine manufacturers and assurances were received that:

i) Up to 25:1 volume ratio, the guaranteed adiabatic shaft efficiency of the first stage liquid expander would be 75%, but it was anticipated that 80% was more likely.

ii) For the separated dry vapour, the guaranteed adiabatic efficiency would be 88%.

A detailed analysis of the expansion process was then carried out assuming that a separator was installed after the initial expansion stage. The pressure loss in the separator was ignored and the following adiabatic shaft efficiencies were assumed for the three expanders:

i) First stage expander	75%
ii) Separated dry vapour expander	88%
iii) Separated liquid expander	50%

The low value assumed for the separated liquid expander was because the volume ratios of expansion anticipated were of the order of 100:1 and hence it was unlikely to be high. This is probably pessimistic, but as will be shown, the value selected hardly affects the result. It was also assumed that the expander shafts drive generators through reduction gearboxes and that the gearbox and generator efficiencies are both 95%.

A series of calculations were then performed in which the intermediate temperature at which the fluid left the first stage of expansion and entered the separator was varied. The results of this are given in graphical form in Figures 5 to 7.

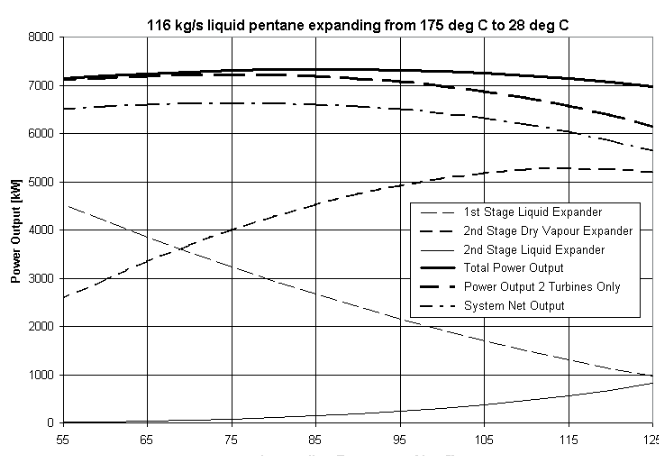


Figure 5. Power Output from Separated Flow Expander System.

As can be seen from Figure 5, the power output possible from the system, which is given in kWe, is very much dependent on the value of the intermediate temperature at which the two-phase mixture is separated. The main features to notice are that as the value of the intermediate temperature decreases, the mass flow of dry vapour increases and therefore a higher power output is obtained from the dry vapour turbine. However, as the intermediate temperature continues to decrease, the available enthalpy drop for the dry vapour is reduced and so a point is reached where the gains in vapour mass flow are insufficient to counter the loss in specific enthalpy drop. At that point, the power from the dry vapour turbine starts to decrease. On the other hand the first stage expander always increases in output as the intermediate temperature drops while the second stage separated liquid expander power decreases continuously with reduction of the intermediate temperature.

The net result of these conflicting trends is that the power output possible from the system reaches a maximum at an intermediate temperature of 85°-90°C when its gross value is approximately 7330 kWe. At these temperatures, approximately 78% of the fluid mass is dry vapour and 22% is liquid.

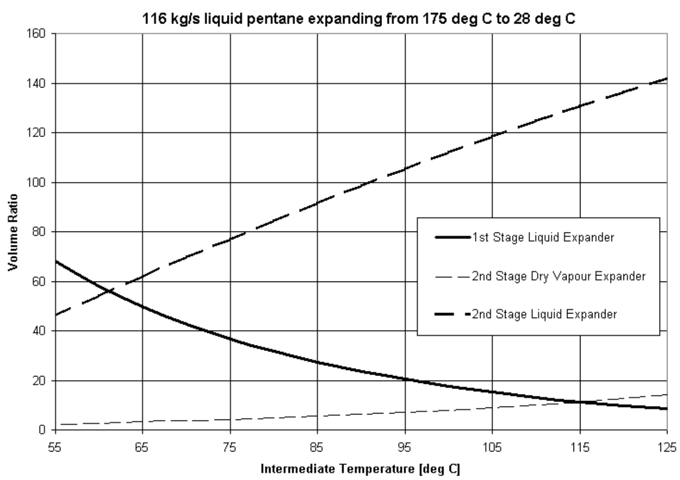


Figure 6. Expansion volume ratio of the Separated Flow Expanders.

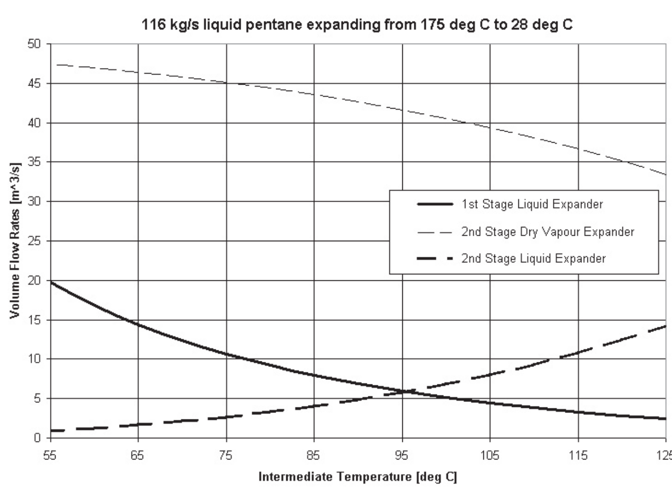


Figure 7. Exit Volume Flow rates from the Separated Flow System Expanders.

As can be seen from Figure 6, this maximum point corresponds fairly closely to the point where the first stage liquid expander volume ratio is of the order of 25:1 which is the operating condition up to which the turbine manufacturers guaranteed the assumed adiabatic shaft efficiency.

We may also note from Figure 5, however, that at this point, the contribution of the second stage, separated liquid expander to the total power is only of the order of 165 kWe, whereas the power output from the first stage expander is approximately of 2500 kWe..

As can be seen from Figure 7, at these intermediate temperatures, the exit volume flow from the second stage separated liquid expander, is of the same order of magnitude as that from the first stage. This is because the vapour density is very much lower at the condensing temperature than at the intermediate temperature. Expander costs are mainly dependent on the exit volume flow, which determines the frame and rotor size. Thus, it follows that the cost per kW of the additional power from the second stage separated liquid expander is approximately 15 times that of the first stage expander.

An Improved Cycle for Power Recovery

The results of the design study indicate clearly that the second stage separated liquid expander can be eliminated for only a small efficiency penalty and a significant cost saving. The system can then be configured in an improved form as shown in Figure 8. Some minor variations on this are given in ref (6).

This system could offer a guaranteed gross electrical output of 7150 kWe and a net electrical output of approximately 6,600 kWe, while the combined overall adiabatic shaft efficiency of the two expanders $\cong$ 80.5%. This value is much higher than is normally associated with two-phase expansion.

It should be noted in the above, that by use of a two-stage feed pump, some of the unrecovered power from the separated liquid is saved by reduced feed pump work. Since the pressure rise across the feed pump is from approximately 0.75 bar – 25 bar, a two stage pump would probably be required even if all the fluid were expanded to the condensing pressure. Thus, this requires no additional component.

The following features would also affect the cost:

i) As shown in Figure 8, the two expander stages can be linked to a common generator.

ii) Although the working fluid expander inlet condition would be unchanged, some of the heat required to reach this state is now supplied by the hot separated liquid mixing with the colder partly pressurised condensate. Thus the working fluid would now enter the heater at approximately 45°C. This would raise the brine exit temperature from the heater to approximately 50°C. Omission of the second stage liquid expander then reduces the external heat input required by about 8% for a loss in power output of 2%. The cycle efficiency of the system is therefore increased and would result in the heater and condenser being slightly smaller and cheaper.

There is still some scope for improvement on these values if:

i) The first stage expander attains an adiabatic shaft efficiency of 80%, as suggested by the turbine manufacturers.

ii) The dry vapour expander, which is relatively large, can be coupled directly to the generator without an intermediate gearbox.

In this case the gross electrical output would be increased to 7350 kWe and a net electrical output of approximately 6,750 kWe, while the overall adiabatic shaft efficiency of the two expanders combined ≅ 82%.

A further possible means of improving the system is to add a small percentage of a lighter hydrocarbon to the pentane. This potentially, could increase the percentage vapour at the intermediate temperature, increase the enthalpy drop in the dry vapour turbine by reduction of exhaust superheat and reduce the size and cost of the expanders by raising the intermediate and condensing pressures. Work related to this on a TFC system with two-phase expansion throughout, has already been performed by the authors, as described in ref (2). More detailed studies are required to evaluate what changes this could effect, but as far as performance is concerned, an initial estimate indicates that these would only be of the order of 2-3%.

Overall, the net output from this system is approximately 60% of that possible from the ideal trilateral cycle as defined by equation (1).

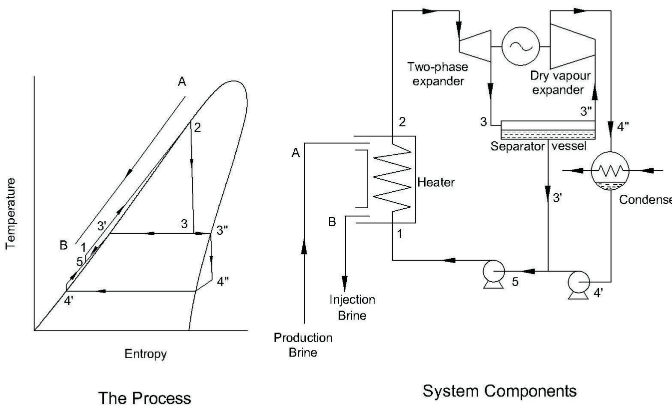


Figure 8. Improved Cycle System.

Practical Advantages of the Improved Power Plant Cycle

The main advantages of the improved cycle system are that for higher liquid brine temperatures, it can yield a high power output with a simpler arrangement than a two stage cascaded Rankine cycle system or any of the Kalina cycles, while it is comprised entirely of standard components operating under conditions wholly familiar to their manufacturers, for which performance guarantees can be given.

In addition, its working fluid in this case is n-Pentane, which, apart from its known flammability, is safe to work with and does not involve special material requirements to protect against potential adverse chemical reactions with it.

Also, when compared to the TFC system, from which it is derived, the slightly higher brine return temperature to the heater improves the cycle efficiency, thereby reducing the heat exchanger costs.

Due to the fact that the dry vapour leaves the second stage expander in the superheated phase, a high level of liquid separation is not required at the exit of the first stage expander, since any residual liquid will evaporate in the dry vapour turbine inlet nozzles and thereby slightly boost the power output.

As the resource ages, the maximum brine temperature will fall and with it, the intermediate temperature will also decrease, hence tending to keep the mass of liquid withdrawn from the separator constant. This will cause the first stage power output to increase, while that of the second stage dry vapour expander will decrease. However, the overall effect will be a fall in output. Partial compensation for this is possible by the addition of more volatile components to the working fluid, as already mentioned, since this can be arranged to further increase the mass of vapour separated out to the dry vapour turbine. The fall in output will also be minimised because the efficient operation of both the first and second stages will hardly be prejudiced with radial inflow turbines, which are fitted with variable inlet guide vanes and thereby maintain high efficiencies over a wide operating range.

Conclusions

The improved cycle system proposed is of great simplicity, which is well suited to recovering electrical power from a liquid resources in the 170°-200°C temperature range and at these temperature conditions it can recover approximately 60% of the theoretical maximum power output. Unlike, its predecessor, the TFC system, from which it is derived, its components are solely of standard type operating in their normal mode. Accordingly, reliable performance estimates can be made for it even on a first build. Also, by using simple binary mixtures of hydrocarbon working fluids, additional gains in output over those cited in this paper, may still be possible.

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