Steam as the working fluid for power recovery from exhaust gases by means of screw expanders

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ABSTRACT

Rankine cycle systems, using steam as a working fluid, are not well suited to the recovery of power from heat sources in the 300-450°C temperature range, such as IC engine exhaust gases, due, mainly to the relatively large enthalpy of vaporisation of water. Admitting the steam to the expander as vapour approximately 50% dry, would be preferable but turbines cannot be used to expand vapours from this state. However, screw expanders can operate well in this mode. It is shown that, apart from being environmentally benign and free from flammability risks, a screw driven wet steam cycle system can recover power from engine exhaust gases, with comparable efficiencies to turbine driven systems using organic fluids at a significantly lower cost per unit output.

1 INTRODUCTION

Mechanical power is most commonly recovered from external heat sources, such as combustion products, in a Rankine Cycle system, using steam as the working fluid. However, in recent years, as interest has grown in using heat sources at lower temperatures for power recovery, there has been a growing trend to look for alternative working fluids and for heat sources at temperatures of up to approximately 200°C. In most cases, it has been shown that organic fluids such as light hydrocarbons or common refrigerants, have distinct advantages over steam. These fluids have unique properties and much of the art of getting the best system, for power recovery from a given heat source, is based on the choice of the most suitable fluid [1,5].

Those fluids most commonly used, or considered are either common refrigerants, such as R134a (Tetrafluoroethane) or R245fa (1,1,1,3,3-Pentafluoropropane), or light hydrocarbons such as isoButane, n-Butane, isoPentane, n-Pentane, n-Hexane and cyclo-Hexane. For some systems highly stable thermal fluids, such as the Dowtherms and Therminols have been used but the very high critical temperatures of these fluids create a number of problems in system design which lead to high cost solutions.

There are, however, numerous sources of heat, mainly in the form of combustion products, already used for other processes, such as the exhaust gases of internal combustion (IC) engines, where the temperatures are rather higher, typically having initial values in the 300°-450°C, where organic working fluids are associated with thermal stability problems and their thermodynamic properties are less advantageous. Unfortunately, at these temperatures, conventional steam cycles also have serious deficiencies.

2 RANKINE CYCLE SYSTEMS

A basic Rankine cycle system, using steam, is shown in Fig 1. It comprises only four main elements, namely, a feed pump, a boiler to heat and vaporise the water, an expander, for generating mechanical power, and a condenser to reject the waste heat and return the water to the feed pump inlet. Normally, the steam is expanded in a turbine, when it is preferable to superheat it before expansion begins in order to avoid condensation of vapour during the expansion process. This is important because steam velocities within the turbine are very high and any water droplets, so formed, impinge on the turbine blades and erode them and also reduce the expander efficiency.

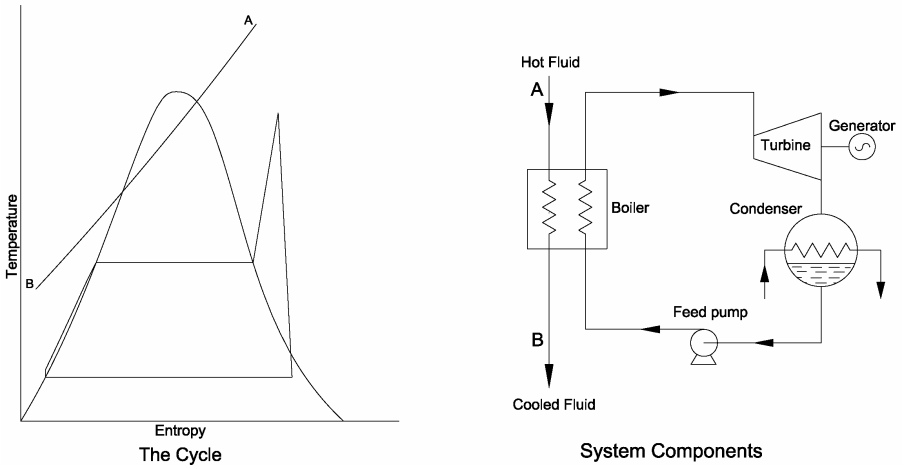

Fig 1 Superheated steam Rankine cycle

By using special materials on the turbine blade leading edges it is possible to reduce the blade erosion problem and thereby steam can enter the turbine in the dry saturated vapour condition, as is done in some geothermal systems. Such a cycle is shown in Fig 2, and this allows for increasing wetness in the latter stages of expansion at the sacrifice of some efficiency. However, no turbine has yet been constructed that can safely accept substantially wet fluid at its inlet.

A problem then exists with admitting steam to the turbine inlet, which becomes more pronounced as the initial temperature of the heat source is reduced. This is the matching of the temperatures of the heat source and the working fluid in the boiler. This is best understood by reference to Fig 3, which shows how the temperature of the working fluid and the heating source change within a boiler, when hot gases are cooled from an initial temperature of 450°C to 150°C to heat pressurised water and evaporate it.

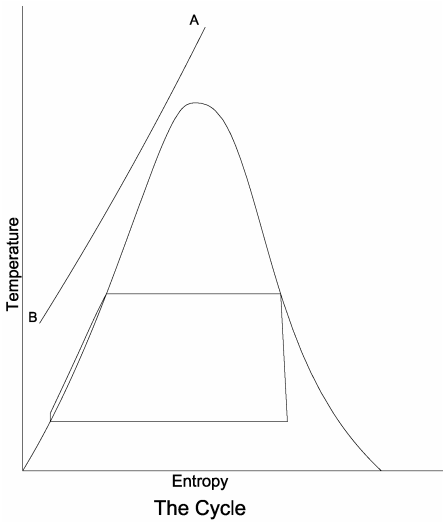

Fig 2 Saturated steam Rankine cycle

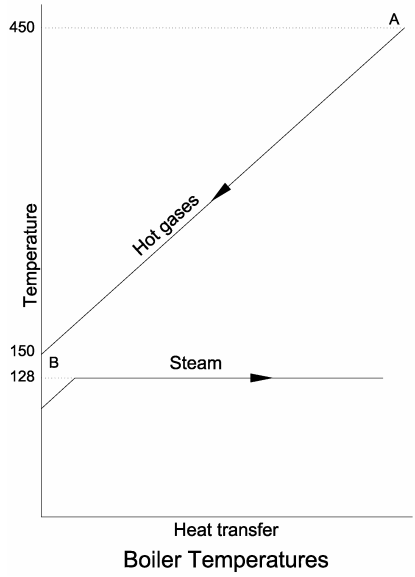

Fig 3 Boiler temperatures

As can be seen, because water has the largest latent heat of any known fluid, the greatest part of the heat received by the steam is required to evaporate it and this occurs at constant temperature. However, the gas stream temperature continuously decreases as it transfers heat to the steam. Accordingly, the evaporating temperature of the steam must be very much lower than that of the initial gas stream temperature and in this case, despite the relatively high initial temperature of the gas stream, the steam cannot evaporate at temperatures much above 120°C.

This great degradation of temperature needed to evaporate the steam, results in a poor power plant cycle efficiency, because high cycle efficiencies are only achieved by increasing the evaporation temperature..

In contrast to this, organic fluids have a much lower ratio of evaporative heating to feed heating and hence can easily attain much higher temperatures, therefore giving better cycle efficiencies. An example of this is shown in Fig 4, where, using the same heat source, it is possible to evaporate pentane at 180°C. This is generally considered to be a safe upper limit for pentane in order to avoid thermal stability problems associated with chemical decomposition of the fluid.

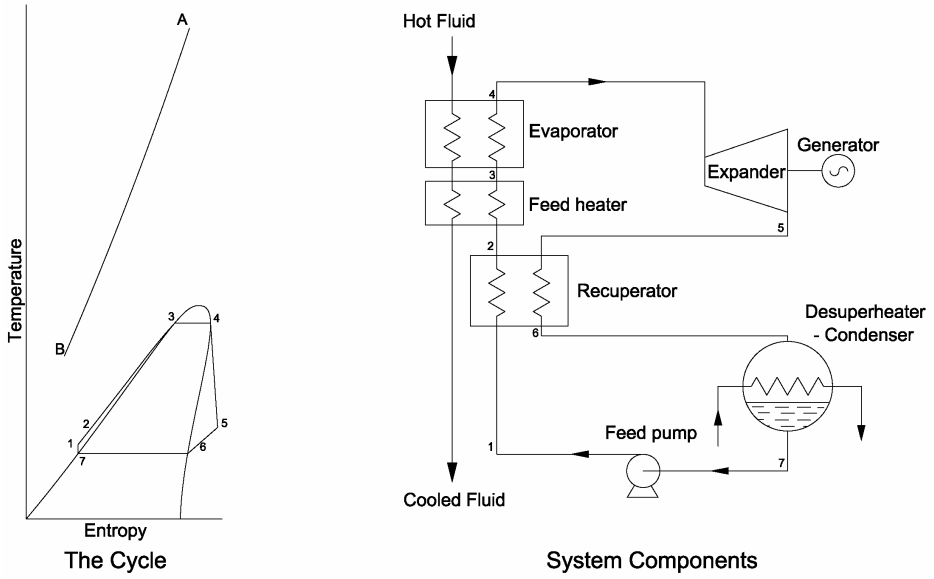

Fig 4 Recuperative ORC system

It can be seen in this case, that unlike steam, starting from saturated vapour, the working fluid becomes superheated as it expands. There are therefore no blade erosion problems associated with its use. In order to improve the cycle efficiency, at the end of expansion, the low pressure superheated vapour can be passed through a counterflow heat exchanger, or recuperator, to recover the heat that would otherwise be rejected in the condenser and use it to preheat the pressurised liquid leaving the feed pump before it enters the boiler. Thus, using pentane, higher cycle efficiencies are attainable.

A major problem with the use of organic fluids is that thermal stability problems are not limited to the bulk temperature of the working fluid. Thus, in the case of pentane, much higher temperatures are attainable by superheating, but when the temperature of the boiler surface in contact with the pentane is much higher than that of the working fluid, as is the case with heat recovery from exhaust gases, there is the risk of fire or explosion in the event of any rupture occurring in the heat exchanger wall separating the working fluid from the heating source. This risk is entirely absent with steam.

A further problem associated with steam is that it has very low vapour pressures at normal condensing conditions required in vapour power plant rejecting heat either

to a cooling water stream or the atmosphere. Thus, at a condensing temperature of 40°C, the vapour pressure of steam is only 0.074 bar. This means that the density of the expanded steam is very low and huge and expensive turbines are required, while there are problems with maintaining a vacuum in the condenser. In contrast to this, pentane at 40°C has a vapour pressure of 1.15 bar. It is therefore far more dense and consequently, the expander required for it will be much smaller and cheaper.

3 WET STEAM CYCLES WITH SCREW EXPANDERS

For units of relatively small power output, in the range of 20 kW to 1 MW, it is possible to consider the use of positive displacement machines such as screw expanders, as an alternative to turbines. Unlike the mode of power transmission in turbomachinery, power is transferred between the fluid and the rotor shafts by pressure on the rotors, which changes with the fluid volume. Consequently the fluid velocities within them are approximately one order of magnitude less than in turbines. Thus, only a relatively small portion of the power recovered is due to dynamic effects associated with fluid motion. Fluid erosion effects are therefore eliminated and the presence of liquid in these machines, together with the vapour or gas being compressed or expanded, has little effect on their mode of operation or efficiency.

On this basis, steam can be used in a modified Rankine cycle in which it enters as very wet fluid, typically with a dryness fraction of the order of only 0.5. as shown in Fig 5. This value can then be adjusted to give the best match between the heat source and the working fluid. Under these operating conditions, it is easy to attain wet steam temperatures of up to 230°C. Temperatures much above this value are limited by rapidly increasing pressures and thermal distortion of the casing and the rotors.

A positive feature of steam is that at these higher temperatures, the pressure is not too high, being only 25 bar at 224°C while its specific energy is much higher than that of organic fluid. Accordingly the feed pump work required for pressurising the working fluid is relatively much less in a steam cycle than in an organic fluid cycle.

The main problem remaining with utilising wet steam with screw expanders therefore lies only with the large size of machine needed to expand to low condensing temperatures. However, this can be largely overcome by raising the condensing temperature to approximately 100°C or more. At this value, this vapour pressure of steam is just over 1 bar and though less than that of the most commonly used refrigerants and hydrocarbon working fluids at the same temperature, is of comparable value.

The disadvantage of condensing at a relatively high temperature is partially overcome by the fact that wet steam can safely attain higher maximum temperatures than the usually employed organic fluids and, because the cycle is inherently better and does not need any recuperative heat exchanger within it, as shown in Fig 4. Accordingly, it will still be better than an ORC when either the heat rejected from the power plant is required for heating purposes, as is the case with Combined Heat and Power (CHP) plant, or even when the ORC operates with a lower condensing temperature when there is no cooling water supply available and an air cooled condenser is required that is not to be excessively large or expensive.

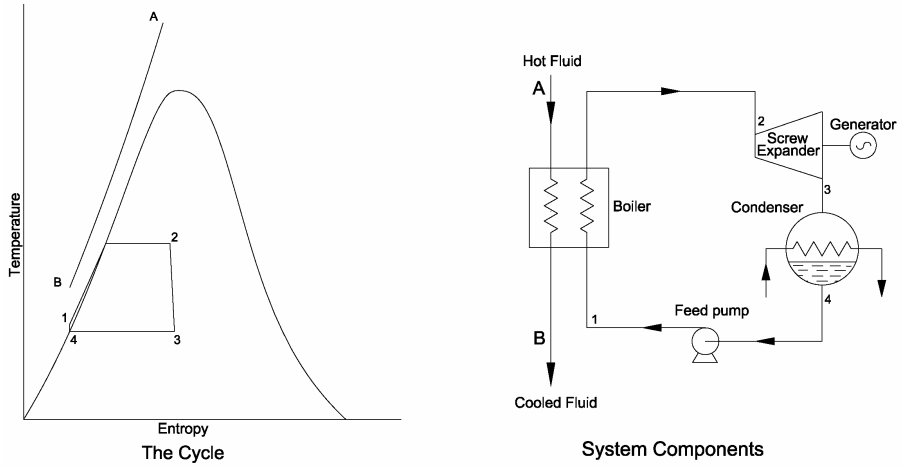

Fig 5 Wet steam cycle system

However, when a supply of cooling water is available, that enables significantly lower condensing temperatures to be attained, the heat rejected from the wet steam cycle condenser can be supplied to a low temperature ORC system in order to recover further power, without incurring the problems of large machine sizes required to expand steam to low temperatures, as shown in Fig 6.

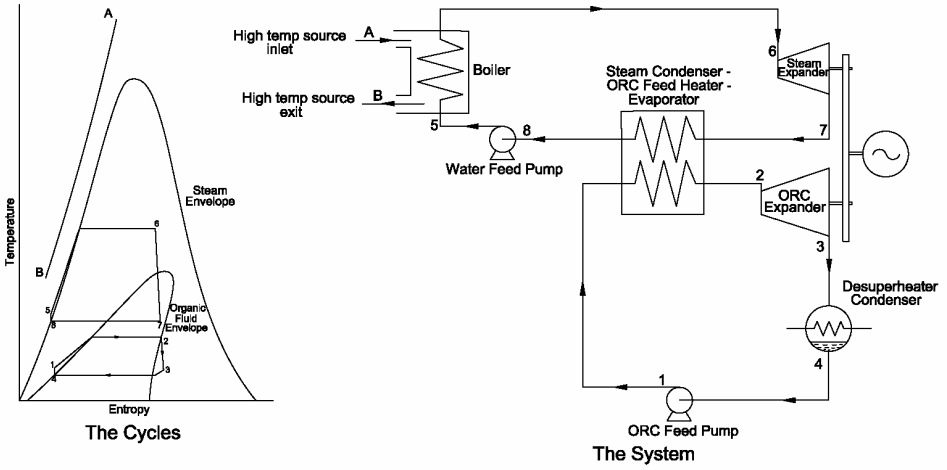

Fig 6 Power recovery when cooling water is available

In fact, such an ORC bottoming cycle has also been proposed for higher temperature ORC systems, when a different working fluid is used for the two cycles.

When the heat source is the exhaust gas of an IC engine, a further possibility is to use a wet steam system to recover the exhaust gas heat that condenses at 100°C

and supplies the rejected heat to a lower temperature ORC system that also receives the engine jacket heat, as shown in fig 7. This will actually reduce the overall efficiency of conversion of heat to power and will therefore increase the cost per unit power output of the power recovery system but will increase the total power recovered.

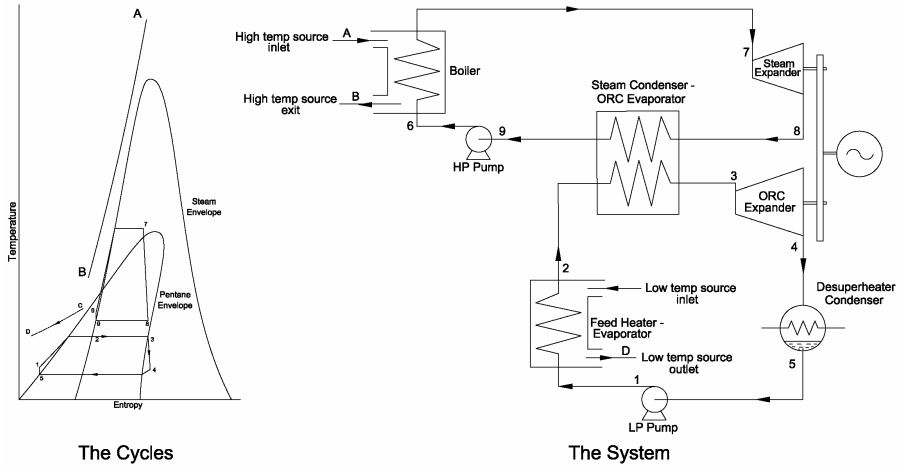

Fig 7 Combined wet steam ORC System receiving heat from two sources

Further details of the possibilities for a wet steam system are given in refs [6, 7].

4 COMPARATIVE STUDY OF WET STEAM AND ORC SYSTEMS

To illustrate the relative merits of wet steam and ORC systems a comparative study was carried out on the recoverable power from a typical IC engine used for electrical power generation. A Jenbacher JGC320GS-LL generator set was selected. When operating with landfill gas as the fuel, this has a rated power output of 1063 kWe with an exhaust gas exit temperature of 455°C. Assuming that the exhaust gases can be cooled to 180°C, the safe limit with landfill gas as the fuel, then 498kW of heat is recoverable from them.

Three cases were considered, namely:

i) A CHP system, where power is recovered from the exhaust gases to either a wet steam or ORC system, from which the heat from the condenser is transferred to a hot water heating system.

ii) Power recovery only from the exhaust gases to either a Wet Steam or an ORC system fitted with an air cooled condenser.

iii) Power recovery only in the case where a supply of cooling water is available. In this case, the Wet steam condenser acts as the feed heater-evaporator of an ORC System that uses R245fa as the working fluid.

Assumed Coolant Conditions:
Case i): Hot water entering the condenser at 70°C and leaving it at 90°C.
Case ii): Air at 30°C

Case iii): Cold water entering the condenser at 20°C and leaving it at 30°C.
Assumed Power Plant Details:

	Screw Driven Wet Steam System	Turbine Driven ORC System
Expander Efficiency	75%	80%
Generator Efficiency	95%	95%
Feed Pump Efficiency	65%	65%
Auxiliary Fan/Pump Efficiency	75%	75%
Auxiliary Drive Motor Efficiency	90%	90%
Maximum Fluid Temperature	224°C	200°C
Working Fluid	Water (+R245fa Bottoming Cycle)	n-Hexane

The results of the study are given in the following table:

	Combined Heat and Power (CHP) System		Air Cooled System No Heat Recovery		Water Cooled System No Heat Recovery	
	Wet Steam	Hexane	Wet Steam	Hexane	Wet Steam + Bottoming ORC	Hexane
Condensing Temp. (°C)	100	100	100	80	Steam =100 R245fa = 35	35
Gross Power Output (kWe)	75.6	69.0	75.6	86.0	Wet Steam = 75.6 R245fa = 41.3 Total =116.9	113.6
Feed Pump Power (kWe)	0.8	7.1	0.8	6.6	Water= 0.7 R245fa = 2.6 Total = 3.3	5.4
Net Power Output (kWe)	74.1	61.1	73.2	72.6	112.3	106.8
Cycle Efficiency (%)	14.9	12.3	14.7	14.6	22.6	21.5
Total Heat Delivered (kW)	419	431	0	0	0	0
Relative Total Heat Transfer Surface Area	28.5	45.2	15.9	30.8	53.3	57.2
Expander Exit Volume Flow Rate (m^3/s)	0.31	0.195	0.31	0.31	Steam = 0.31 R245fa = 0.174	1.19

In this case, n-Hexane was chosen as the ORC working fluid since it has a higher critical temperature than n-Pentane and therefore can be heated to 200°C, which is

above the critical temperature of Pentane. Also, it is known to be used as the working fluid by one known developer of higher temperature ORC systems. Other fluids of higher critical temperatures could have been chosen such as Toluene, which indicate a higher power output but with significantly larger heat exchangers for the ORC system. However, since the author is unaware of any system having been operated with Toluene, its overall suitability is unknown. The comparison was therefore limited to working fluids that are known to have been used.

As can be seen, the Wet Steam system produces more power than the ORC in every case.

The analysis also made some assumptions more favourable to the ORC. Thus, it was assumed that for these relatively small power outputs the ORC turbine would have an adiabatic efficiency of 80%, while that of the screw expanders, assumed both for the Wet Steam system and, where appropriate, its ORC bottoming cycle, would be only 75%. Also, in comparing the relative heat exchanger sizes, it was assumed, as a first approximation, that the overall heat transfer coefficients for the ORC heat exchangers would be equal to their equivalent Wet Steam unit. The main disadvantage of the ORC system is the need for a large recuperative heat exchanger to desuperheat the working fluid before entering the condenser. Without this, the ORC cycle efficiencies would be unacceptably low. In fact, even when an ORC bottoming cycle system is linked to the Wet Steam condenser, the high heat transfer coefficients, associated with two-phase heat transfer on both surfaces, result in a smaller total heat exchanger surface area.

In most cases, the volume flow rate through the steam expander is no greater than that through the turbine. It should be noted that in the case of the water cooled system, not only is the Hexane turbine very large, but the condensing temperature is only 0.3 bar, which means that the ORC system would have to be completely hermetically sealed in order to prevent problems with maintaining a vacuum in the condenser. This is normally achieved by the turbine being directly linked to a high speed generator with an invertor control to produce output at 50/60Hz. Such units are expensive. In contrast, the screw expander could be coupled directly to a 2-pole generator or through a low cost belt drive and its cost would be only a fraction of that of the equivalent turbine.

To summarise the comparison, the Wet Steam Cycle system has the following potential advantages over ORC equivalents that are known to have been used.

i) Water is a natural environmentally friendly working fluid.
ii) Steam has no thermal degradation problems. Hence, higher working fluid temperatures are possible within the cycle.
iii) Water is a very low cost of working fluid.
iv) Water requires far smaller feed pump work than organic working fluids. Hence, even if the feed pump efficiency is very low, its effect on the cycle is small. Water pumps are also far more freely available.
v) By condensing at 100°C or higher, the problems of large expanders are overcome while cycle efficiencies are still higher than attainable with currently used organic fluids condensing at lower temperatures.
vi) By condensing at 100°C, air cooled units become economic even under the most unfavourable climatic conditions, while at this rejection temperature additional power can be recovered even in CHP systems.
vii) Heat exchanger costs are far lower than with organic fluids because:
    a) No recuperator is required.
    b) Less heat is rejected from the condenser due to the higher cycle efficiency.

c) Water-steam has far higher thermal conductivity and two-phase heat transfer coefficients than any other working fluid.
d) The heat exchangers will be smaller because they can work with larger temperature differentials across the surface.

viii) Where low ambient temperatures or cooling water supplies are available, the heat rejected from the steam condenser can be supplied to an ORC system to maximise power recovery without the need for huge expanders. In that case the overall cost per unit output of the combined system will be less than that of a single ORC system because the additional cost of the intermediate heat exchanger, the additional feed pump and screw expanders is less than that of the large recuperator and turbine that the single ORC system will require.

5 CONCLUSIONS

By taking advantage of the fact that screw expanders can admit wet fluids, which turbines cannot, wet steam can be used as the working fluid for power recovery from IC engine exhaust gases and equivalent heat sources in the 250°-500°C temperature range. The resulting systems are likely to have comparable efficiencies while being significantly more cost effective than ORC systems operating over the same temperature range.

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