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Optimisation of two-stage screw expanders for waste heat recovery applications

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Abstract. It has previously been shown that the use of two-phase screw expanders in power generation cycles can achieve an increase in the utilisation of available energy from a low temperature heat source when compared with more conventional single-phase turbines. However, screw expander efficiencies are more sensitive to expansion volume ratio than turbines, and this increases as the expander inlet vapour dryness fraction decreases. For single-stage screw machines with low inlet dryness, this can lead to under expansion of the working fluid and low isentropic efficiency for the expansion process. The performance of the cycle can potentially be improved by using a two-stage expander, consisting of a low pressure machine and a smaller high pressure machine connected in series. By expanding the working fluid over two stages, the built-in volume ratios of the two machines can be selected to provide a better match with the overall expansion process, thereby increasing efficiency for particular inlet and discharge conditions. The mass flow rate though both stages must however be matched, and the compromise between increasing efficiency and maximising power output must also be considered. This research uses a rigorous thermodynamic screw machine model to compare the performance of single and two-stage expanders over a range of operating conditions. The model allows optimisation of the required intermediate pressure in the two-stage expander, along with the rotational speed and built-in volume ratio of both screw machine stages. The results allow the two-stage machine to be fully specified in order to achieve maximum efficiency for a required power output.

1. Introduction

The Organic Rankine Cycle (ORC) provides a means of recovering useful energy from low temperature heat sources. In comparison with conventional high temperature steam Rankine cycles, the low temperature of these heat sources means that the attainable cycle efficiency is much lower, while the required surface area of the heat exchangers per unit power output is much higher. The lower latent heat of evaporation of organic fluids relative to steam also means that the feed pump work required in ORCs is a significantly higher proportion of the gross power output.

For low source temperatures, the power generation cycle normally considered is that where the working fluid enters the expander as dry saturated vapour, as shown in Figure 1. However, in most cases, this leads to the working fluid leaving the expander with some superheat, which must be removed before condensation begins.

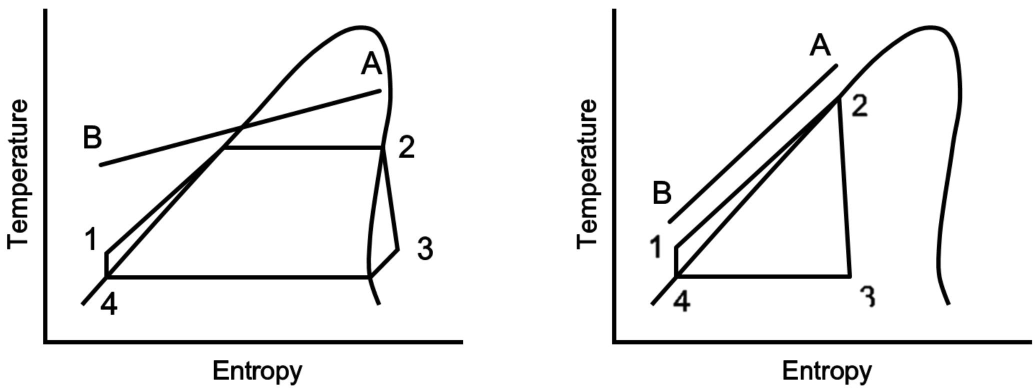

Figure 1: Illustrative T-s diagrams showing conventional ORC with dry saturated vapour at the expander inlet, and TFC with wet saturated vapour at the expander inlet

Maximising net power output from the cycle is a compromise between increasing the mean temperature of heat addition (which, in accordance with Carnot's principle, can increase cycle efficiency) and increasing the amount of heat extracted from the source, which requires a lower evaporation temperature.

This can be achieved in a Trilateral Flash Cycle (TFC) which expands the working fluid from a saturated liquid state as shown in Figure 1. Although this system has been considered for many years, 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.

By the use of a screw expander, instead of the more conventional turbine, it is possible to admit the working fluid to the expander as wet vapour and thereby eliminate both the need to desuperheat the vapour after expansion and simultaneously to raise the evaporation temperature, thus improving the cycle efficiency. The potential cost and performance benefits of using screw expanders in ORC systems have been extensively studied for geothermal applications [1-3].

In the fields of geothermal and waste heat recovery systems, there is growing interest in generating power from heat sources with initial temperatures in the 170-200°C range. At these temperatures, simple ORC systems are less attractive, as the method recommended for their use under these circumstances is to operate two such systems in a cascade arrangement. 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 thirty 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 is pure n-pentane. With such a working fluid, the expansion process involves a volume ratio of expansion of the order of 160:1, which, using twin-screw machines, requires a two stage expander to achieve efficient expansion. The problem, therefore, lies in the design of the first and second stage expanders, to admit saturated liquid and wet vapour respectively. While previous work has studied the performance of combined twin-screw and turbine systems [3], recent progress in developing and validating a computational twin-screw model has allowed the performance of single and two-stage systems to be investigated in greater detail. This model allows the optimisation of the expander parameters for a particular application, and can be incorporated with other detailed component models to allow multi-variable optimisation of low temperature heat recovery systems.

2. Twin-screw expander model

A full thermodynamic model of the expander has been created for investigating the performance of two-stage expanders. This is based on the quasi one dimensional analysis of twin-screw machines as described by Stosic and Hanjalic [4,5], which has been extensively validated for compressors for a wide range of working fluids and operating conditions. For expanders, the model has been validated for expansion of low dryness fluid (including saturated liquid) using the refrigerant R113 [6], and more recently for the expansion of high dryness wet steam [7]. Using this procedure, machine geometry and rotor profiles have been optimised for a particular set of operating conditions representative of those considered in this paper, and have been fixed for the purposes of the current study. The City University 'N' rotor profile described in Figure 2 has been used in the current analysis, as this geometry is known to have benefits including greater throughput and a stiffer gate rotor than is possible using alternative profiles with similar blow-hole area and sealing line lengths [8].

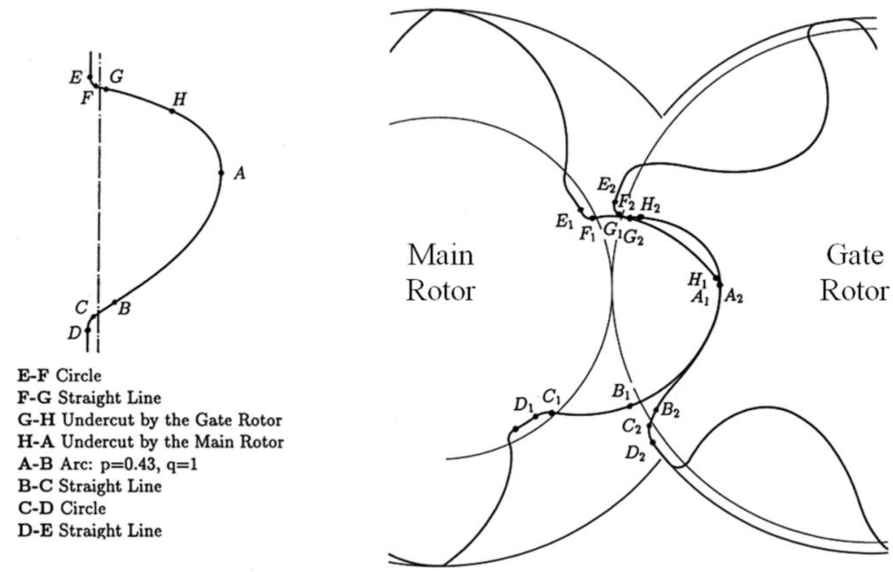

Figure 2: Description of the City University 'N' rotor profile for positive displacement screw machines

In principle, the screw machine geometry optimisation can be integrated with the expander analysis described here to ensure the best profile is used for the required operating conditions, but this would be very computationally intensive and is not expected to significantly affect results. Furthermore, from a manufacturing perspective it would be prohibitively expensive to produce an optimised machine for every different application. For a specified geometry, the characteristics of the twin-screw machine such as the curve of working chamber volume against angular position (illustrated in Figure 3), sealing line lengths, blowhole area and axial/radial clearances between the rotors and the casing are defined as fixed inputs for the expander model. An important machine parameter in the built-in volume ratio, BIVR, defined as the ratio of working chamber volumes at discharge port opening and suction port closing. Figure 3 illustrates how increasing the BIVR for a particular machine increases the volume of working fluid admitted through the suction port per revolution. For a particular rotational speed of the machine, the volumetric and mass flow rates can be determined.

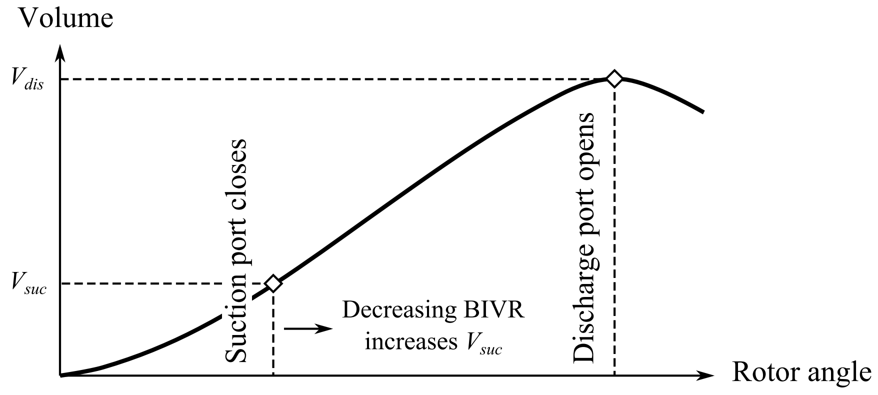

Figure 3: Schematic diagram of working chamber volume as a function of rotor angle

For single stage screw machines, the inlet dryness fraction and the pressures at the inlet and discharge are defined by the requirements of the cycle. The variable input parameters required for the expander model are then limited to the expander size, main rotor speed and BIVR. Two approaches can be taken to match the machine operation to the required cycle conditions:

i. The BIVR is specified and iterations are performed to find the rotor speed required to match the mass flow rate of expander to that of the working fluid required in the cycle – no limits are imposed on rotor speed, which in some cases can become impractically high.
ii. The rotor speed is fixed and iterations are performed to find the value of the BIVR required to match the mass flow rate – if the BIVR is greater than the limit for the chosen screw machine geometry then the expander cannot meet the requirements of the cycle conditions.

For two-stage machines, the intermediate pressure between the two stages is an additional input parameter. There is also an additional constraint, as the mass flow rate through both the high pressure (HP) and low pressure (LP) stages must be the same. While the mass flow rate of the HP stage is largely dependent on the inlet conditions and the size and BIVR of the HP machine, it is also dependent on the intermediate pressure, as this affects leakage flows in the machine. To characterise the performance of a two-stage expander for particular conditions, the following iterative approach is therefore required:

i. Specify the size, speed and BIVR of both stages.
ii. Estimate the intermediate pressure, and calculate the mass flow rates of the HP and LP stages.
iii. While the difference between the HP and LP mass flow rates is greater than an allowable error, repeat step ii.
iv. While the difference between the converged and required mass flow rates is greater than an allowable error, repeat steps i-iii, fixing either the speed or BIVR of the two stages as required.

The single or two-stage expander efficiency calculated using these approaches can be used in a thermodynamic cycle model to calculate overall cycle performance for specific operating conditions. It is possible to then apply an iterative numerical procedure to identify the optimum operating conditions for the cycle. However, the focus of this paper is to illustrate how the optimum expander parameters can be selected for specified cycle conditions.

3. Modelling of single and two-stage expanders for TFC application

The analysis presented in this paper has been performed for a simple heat recovery application from a single phase source fluid, defined as follows;

Assumptions:

Heat Source 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	5°C

For the proposed TFC system, in view of the fact that expansion must begin from the saturated liquid condition, a suitable working fluid for this case is n-pentane, which has a critical temperature of 196.6°C. The pressure of the working fluid in the condenser was constrained to be greater than or equal to atmospheric, so as to prevent air leaking into system. An initial cycle analysis program was used to identify suitable operating conditions for the expander;

Expander Inlet dryness fraction	0 (i.e. saturated liquid)
Expander Inlet Temperature	175°C
Condensing Temperature	36°C

Rather than specify the heat input to the cycle, and thereby determine a required mass flow rate for the working fluid, it is useful to characterise the performance of a range of single and two-stage expanders as a function of mass flow rate at these conditions. Standard twin-screw machine sizes, with main rotor diameters ranging from 145-408mm, have been analysed in order to illustrate what is achievable with practical single and two-stage expanders. To identify the maximum mass flow rates possible with these machines, and to ensure high efficiency, the performance has been considered at maximum allowable rotational speeds corresponding to a main rotor tip speed of 60m/s.

4. Results of TFC expander modelling

For the application specified above, a two-stage machine requires a relatively small HP machine in comparison with the size of the LP machine, due to the much higher density of the fluid at the HP inlet. A combination of a 145mm HP machine with a 408mm LP machine has been found to achieve good overall performance with well-matched expansion in both stages. The mass flow rate, overall adiabatic efficiency, required intermediate pressure and total shaft power are all dependent on the BIVRs of the HP and LP machines. The overall performance of the two-stage machine has therefore been calculated over a range of BIVR values, and contour maps of the key results are shown in Figures 4 and 5.

The results in Figure 4 show that mass flow rate is, as expected, very strongly dependent on the BIVR of the HP stage. Figure 5 shows that the maximum overall adiabatic efficiency occurs at BIVR values of 3.4 and 3.6 for the HP and LP stages respectively. This corresponds to an intermediate pressure of 6.9 bar(abs), mass flow rate of 9.9 kg/s and total shaft power of 520 kW. For a fixed HP BIVR, it can be seen that the required intermediate pressure increases as the LP BIVR increases. The maximum efficiency point corresponds to the case when the BIVRs of both stages are well matched to the expansion, but at lower values of LP BIVR, the intermediate pressure falls, leading to under expansion for the HP stage and over expansion for the LP stage. Conversely, at higher values of LP BIVR the rise in intermediate pressure leads to over-expansion for the HP stage and under-expansion for the LP stage. In conclusion, the circular efficiency contours are a result of over and/or under expansion in one or both of the expander stages.

Power output can be increased by moving away from the maximum efficiency point, but it is important to choose the BIVR values so as to ensure that efficiency is maximised for a particular power output. Figure 5 shows that maximum power for any HP BIVR occurs at a constant LP BIVR value of around 3.6, and that this closely corresponds to the maximum efficiency possible for a particular value of HP BIVR. It is therefore possible to plot curves showing the maximum values of shaft power and adiabatic efficiency as functions of the mass flow rate. These are shown in Figure 6 along with the corresponding performance of single stage expander for the same application. In all cases, the curves show the full range of performance achievable within the practical range of BIVR values.

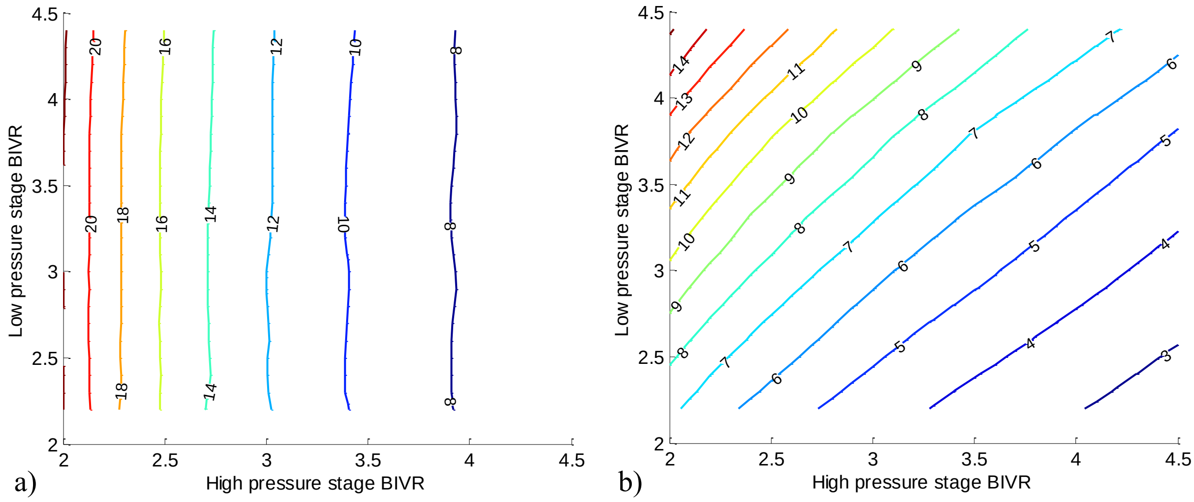

Figure 4: Contour maps showing a) mass flow rate (kg/s) and b) intermediate pressure (bar) of two-stage expander

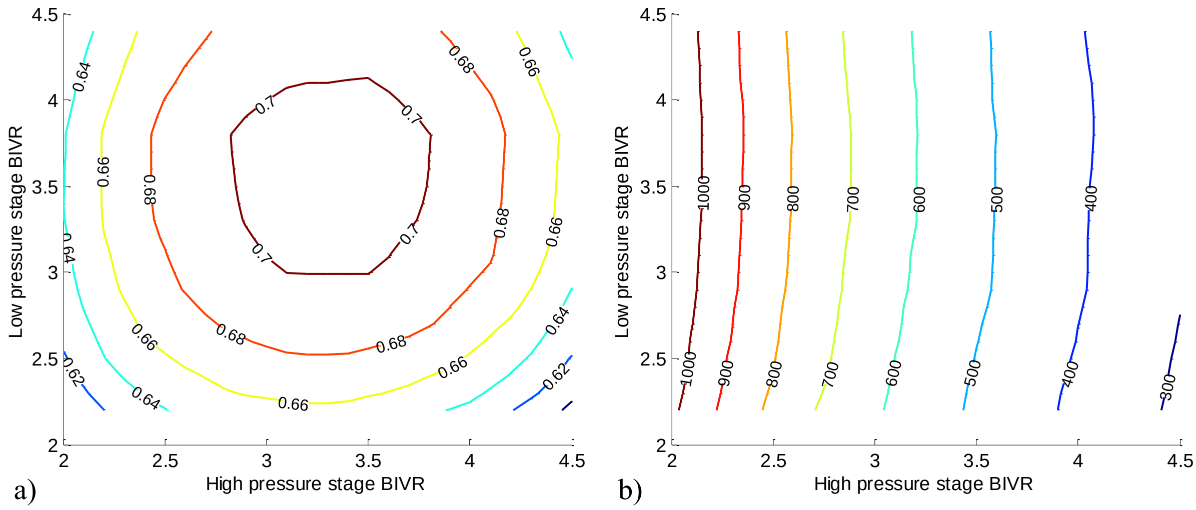

Figure 5: Contour maps showing a) adiabatic efficiency and b) total shaft power (kW) of two-stage expander

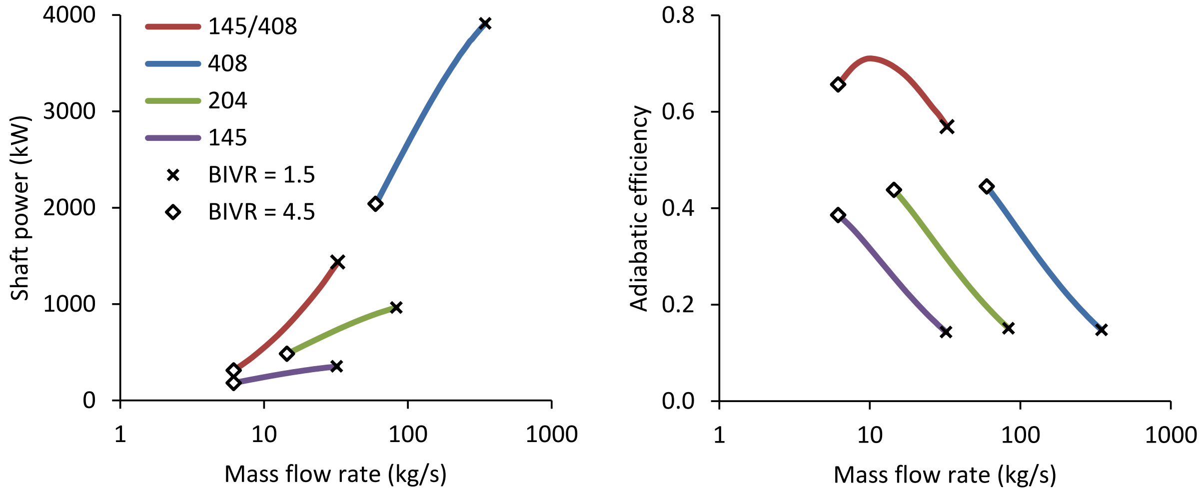

Figure 6: Performance comparison of single and two-stage expanders

The results in Figure 6 show that for single stage expanders, increasing the BIVR increases the efficiency, while reducing the mass flow rate through the expander. For all of the single stage expanders considered, the maximum efficiency is well below 50% due to the large degree of under expansion. This is a result of the limited BIVR being much lower than the actual specific volume ratio of the working fluid over the expansion process.

The two-stage machine achieves a much greater combined BIVR, and is therefore able to better match the overall expansion. The peak of the efficiency curve in Figure 6 shows the point where the expansion in the two-stage machine is best matched to the operating conditions; at higher mass flow rates the efficiency falls due to over expansion of the working fluid in the HP stage, while at lower mass flow rates it falls due to over expansion in the LP stage. Interestingly, the results in Figure 6 suggest that a two-stage machine may be viewed as equivalent to the LP machine operating as a single stage but with a BIVR higher than the practical limit; this is illustrated by the fact that the shaft power and efficiency curves for the two-stage 145/408 machine are essentially extensions of the performance curves for the single 408 machine, covering a lower range of mass flow rates. It is also worth noting that, as the mass flow rate of the two-stage machine is largely a function of the BIVR of the HP stage, this range of achievable mass flow rate is very close to that of the single 145 machine. In summary, compared to the LP stage operating alone, the addition of the HP stage can be seen to increase efficiency, but only by reducing mass flow rate and hence power output.

5. Effect of expander selection on TFC performance

The efficiency and mass flow rate of the expander affect the required power input and net power output of the TFC system. Two important measures of the overall system performance are the conversion efficiency, defined as the net power output divided by the *available* heat input, and the cycle efficiency, defined as net output power divided by the *actual* heat input. The following component efficiencies have been used in order to estimate the performance of both the TFC and a conventional saturated vapour ORC (see Figure 1).

Assumed component efficiency:

Feed pump	0.7
Motor	0.9
Generator	0.95
Turbine	0.82

For the TFC system, net power output is dependent on the mass flow rate, as this affects both the expander efficiency and work done in the feed pumps. Using the above component efficiencies, the net electrical power output of the system is shown in Figure 7, and the resulting cycle and conversion efficiencies are shown in Figure 8.

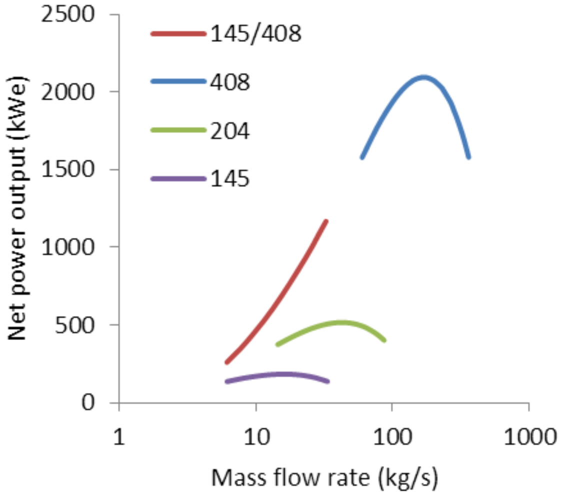

Figure 7: Net power output from TFC system using single and two-stage expanders

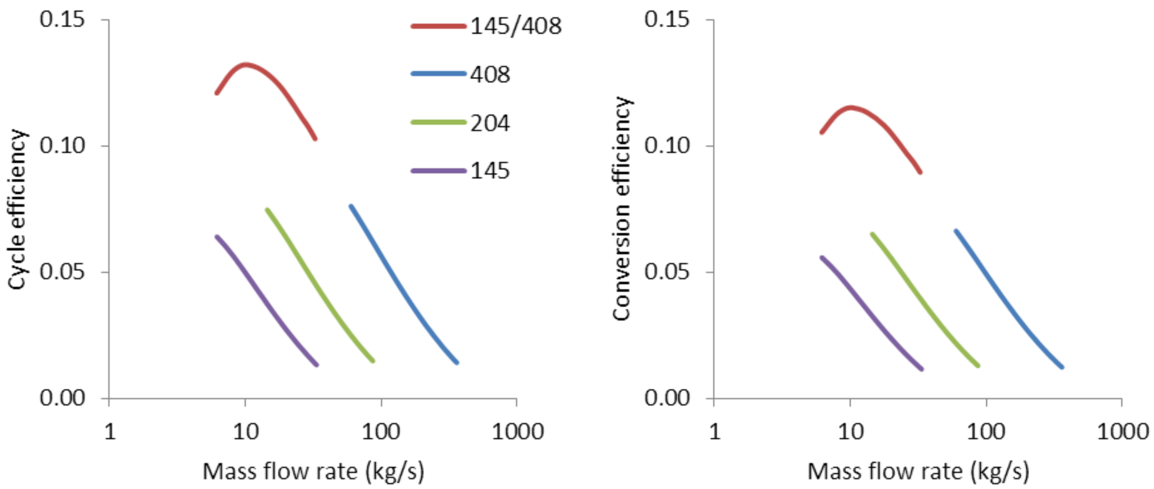

Figure 8: Achievable TFC cycle and conversion efficiency using single and two-stage expanders

Using the same operating conditions and component efficiencies in a thermodynamic cycle model, a simple saturated vapour ORC without recuperative feed heating, operating under the same conditions, was optimized in order to achieve maximum net power output. This was found to occur with cycle and conversion efficiencies of 12.5% and 8.8% respectively. The results from Figure 8 suggest that in the range of system sizes covered by the TFC with two-stage expander (250-1200kWe), the conversion efficiency is always greater than that which can be achieved in a simple saturated vapour ORC, largely due to the greater recovery of available heat from the source fluid.

6. Conclusions

The study presented in this paper shows that two-stage screw expanders can match the required volume ratio for the expansion of saturated liquid in waste heat recovery applications, and achieve high overall adiabatic efficiency. The design parameters for the two-stage machine can be optimised in order to maximize shaft power output for a given mass flow rate, and the possible range of operation of the two-stage machine has been mapped out. This allows a direct comparison of the performance of different single and two-stage machines operating under the same conditions. In the application discussed in this paper, the TFC using a two-stage expander is predicted to achieve a higher overall conversion efficiency than a conventional saturated vapour ORC.

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