Investigation of a Thermo-Fluidic Exchange Pump in Trilateral Flash and Organic Rankine Cycles

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Abstract. It is well known that large amounts of energy loss occurs at low temperature states in a wide range of industrial processes. The recovery and reuse of this energy is at the forefront of increasing the overall efficiencies of industrial systems. The aim of this paper is to investigate the effectiveness of using a Thermo-Fluidic Exchange (TFE) pump at low temperature conditions in both a Saturated-Vapour Organic Rankine Cycle (SORC) and a Trilateral Flash Cycle (TFC).

For some low temperature applications, TFCs have been shown to achieve higher net power output than conventional SORCs, due to their ability to extract more heat from the source fluid. This is the subject of current research as a result of advancements made in the design of positive displacement machines for operation as two-phase expanders. Conventional turbines cannot be used for TFCs as they must operate in the vapour phase.

One drawback of the TFC is the higher working fluid mass flow rate required. Depending on the scale of the system, this can potentially cause difficulties with pump selection. A TFE pump uses heat input to the system to increase the pressure and temperature of the working fluid, rather than the work input in a standard mechanical pump. This paper compares the net power output achievable using both mechanical and TFE pumps with SORC and TFC systems. The results suggest that the TFE pump could be a viable option for TFC systems.

1. Introduction
With climate change and global energy use increasing, there is increasing awareness of the need to ensure the efficient use of resources. Waste heat recovery is a huge market as low-grade waste heat (below 373K) makes up 63% of all primary energy consumption losses [1]. The need to harness more of this energy and to make already installed systems more efficient is necessary to limit both emissions and the effects of climate change.

Trilateral Flash Cycle (TFC) cycles have received renewed attention in recent years due to the advancements made in two-phase expanders. TFC cycles have been reported to increase the cycle efficiency by 14-85% over a Saturated Organic Rankine Cycle (SORC) [2]. One characteristic of the TFC is the higher working fluid mass flow rate required compared to conventional SORCs, due to the smaller change in the specific enthalpy of the working fluid during heat addition. Depending on the scale of the system, this can potentially cause difficulties with sourcing and selecting a feed pump with appropriate capacity. This paper will focus on the TFC cycle with reference to the SORC as a basis for comparison of results.
A Thermo-Fluidic Exchange (TFE) pump uses heat input to the system to increase the pressure and temperature of the fluid, rather than work input in a standard mechanical pump. Previous results have shown that certain TFE pump configurations have a higher overall efficiency, when compared to a standard mechanical pump [3]. This is mainly due to the exclusion of input work. They also generally have a lower boiler efficiency, due to the inclusion of an internal evaporator as discussed in the next section. This application of heat to achieve pumping can be traced back to the 1700’s when Thomas Savery first described a model of ‘An Engine to Raise Water by Fire’ [4]. He used a double furnace to heat up one of two different sized boilers each with different amounts of water, and with a manual valve and filling pipes connecting them. The increasing pressure of the expanding fluid/vapour mixture was used to displace colder fluid and increase its pressure.

More recently, research has shown the inclusion of a TFE pump in a dual-loop Rankine cycle. The results show an improvement in net power of 7.6% in a low-temperature loop, compared to a mechanical pump [5]. A corresponding decrease in mass flow rate of 8.6% was also predicted, which will reduce the size of the system and thus the cost.

2. Analysis of Thermodynamic Cycles with thermo-fluidic exchange pumps
The aims of the analysis presented in this paper are to investigate the operation of cycles incorporating TFE pumps compared to mechanical pumps. This will be achieved by considering applications defined by a range of source inlet temperatures and optimising the cycle efficiency for each to show the influence on the overall efficiency of the cycle.

The analysis will use SORC and TFC both with differing mechanical and TFE pump configurations. The focus will be on the TFC with a TFE pump, with an aim of finding out how these differing cycles affect the overall efficiency and thus the net work output. The TFC is being considered as it requires a larger mass flow rate of the working fluid for the same amount of input heat, compared to a SORC.

3. Details of Cycle Modelling
The two types of cycles considered are discussed in more detail. The pumps are either assumed to be constant efficiency mechanical pumps, or TFE pumps. The TFE pump model is described in detail in Section 3.2 with reference to the multiple stages of operation.

3.1. Methodology for cycle analysis
The trilateral flash cycle is characterised by expansion of the saturated liquid as shown in Figure 1(b). The terminology used to define the various points in the cycle is as follows:
- Point 1 - Saturated liquid at P1, T1
- Point 2 - Subcooled liquid, calculated though Work Input needed and efficiency of pump
- Point 3 - Saturated liquid at P3, T3
- Point 4 - Two-phase mixture, calculated thorough Work Output achieved and efficiency of turbine.

The SORC cycle differs by adding heat at constant pressure until the working fluid is at a saturated vapour state, with work recovered via the expansion process in the superheated vapour region.
The following assumptions are made for the cycle analysis. It is important to note that the starting temperature of the working fluid is fixed, and the mass flow rate of the source fluid is set to 1 kg/s so that results are shown as per unit source mass flow rate:

- Isobaric condenser and evaporator pressure at $P_c$ and $P_{ev}$ respectively.
- $T_{1, wf}$ (pre-pump, post-condenser) is at 45°C.
- $\eta_t$ is 70% 
- $\eta_p$ is 50%
- $c_{p ex}$ is 1050 (J/kg K) for the exhaust fluid.
- $pp$ is 10°C for the evaporator and condenser, this is a dynamic pinch point.
- $m_s$ is taken as 1 kg/s.
- Pentane is the working fluid used.

Taking these main cycle assumptions, a TFE pump model can be created to suit the aims.
3.2. Analysis of Thermo Fluidic Exchange Pump

The TFE pump analysis considered here has been modelled previously by Richardson, and this paper builds on that previous model. A model of the TFE pump can be seen in figure 2 with an accompanying state and process flow of the fluid in table 1. The two main pressures are denoted by $P_c$ as the condenser pressure and $P_{ev}$ as the evaporator pressure. Density is used to help show the mass differences per unit volume through different stages and processes.

Where fluid with density:

- $\rho_{v,sat}$ is saturated vapour at a pressure of $P_{ev}$,
- $\rho_{l,sat}$ (red) is saturated liquid at pressure $P_{ev}$,
- $\rho_{l,sat}$ (blue) is saturated liquid at pressure $P_c$, and
- $\rho_{l,sat}(r)$ is the refill saturated liquid at $P_c$. The process follows the following steps and corresponding states and is shown for the movement of 1 packet of mass.
Table 1. States and processes of a TFE pump cycle

| State 1: | Condenser at pressure $P_c$, with mass fraction 0.5 saturated vapour and 0.5 saturated liquid. Mass exchange at $P_{ev}$ with 100% saturated vapour. |
| Process 1-2: | CV2 is closed and CV1 is opened which allows vapour at pressure $P_{ev}$ to flow isentropically through from the Mass Exchange. |
| State 2: | The isenthalpic flow of vapour from the mass exchange increases the pressure in the condenser, creating a volume in the Mass Exchange which is filled by saturated liquid from the condenser. |
| Process 2-3: | All the vapour at $P_{ev}$ from the Mass Exchange is transferred to the condenser and displaces all the saturated liquid at $P_c$ from the condenser to the Mass Exchange through NRV1. CV1 is closed. |
| State 3: | Condenser with 50/50 saturated vapour at $P_c$ and $P_{ev}$. Mass Exchange with 100% saturated liquid at $P_c$. |
| Process 3-4: | CV2 is opened which allows vapour at pressure $P_{ev}$ to flow isentropically through from the Evaporator. |
| State 4: | The flow of vapour from the evaporator increases the pressure in the mass exchange chamber, leaving a void in the evaporator. |
| Process 4-1: | All the vapour at $P_{ev}$ from the evaporator is transferred to the mass exchange chamber and displaces all the saturated liquid at $P_c$ from the mass exchange chamber to the evaporator through NRV2. |

During this process there is heat transfer happening in both the condenser and expander. A more detailed view of the condenser can be seen in figure 3, it shows the input of saturated liquid at $P_c$ (refill) and throttled vapour that was previously a saturated vapour at $P_{ev}$. Heat is removed to change the throttled vapour to a saturated liquid at $P_c$. The resulting output is saturated liquid at $P_c$. The change of state and volume of the fluid enables the refill fluid to enter.

The assumptions for TFE model can be summarised as follows:

- Isentropic compression through CV2.
- Isenthalpic expansion through CV1.
- Pump takes $P_{l, sat,c}$ and delivers $P_{l, sat,ev}$ to the evaporator.
- Evaporator takes $P_{l, sat,ev}$ and delivers $P_{v, sat,ev}$ to the expander.
- Condenser takes superheated vapour at $P_c$ and delivers $P_{l, sat,c}$ to the pump.
- Liquid entering the mass exchange chamber does not transfer any heat.

Once the model for the TFE is defined, it can be incorporated into the model for the TFC and SORC thermodynamic cycles, as described in the following section.

3.3. Analysis of combined cycle and TFE

Previous calculations were deduced for the TFE pump through one pump cycle during a steady state operation. An internal energy modelling is used to link the change in density and enthalpy to the heat in/output, as seen in equation 1 and 2.

$$q_{in, pump} = \frac{h_{l, ev}(\rho_3 - \rho_1) + h_{v, ev}(\rho_4 - \rho_3) + h_{v, ev}P_{ev} - h_{v, ev}P_{ev}(\rho_3 - \rho_1)}{\rho_3 - \rho_1}$$  

$$q_{out, pump} = q_{in, pump} - (h_{l, ev} - h_{l, c})$$

As this paper concentrates on the TFC, there is an introduction of a recirculation mass flow rate ($\dot{m}_r$). During an ORC which Richardson’s paper was based on, $\dot{m}_r$ is negligible as the working fluid needs to be
heated to a saturated vapour state. But in a TFC, the working fluid only needs to be heated to a saturated liquid state. This can be seen more clearly in figure 4. It is this extra heat input between points 2 and 3 that shows the need for this distinction in the modelling, which must also be applied when considering the heat rejection on the condensing side.

\[ m_r (h_{3s} - h_3) = \dot{m}_s c_p (T_{st} - T_{sa}) \]  
\[ m_{r+wf}(h_3 - h_1) = \dot{m}_s c_p (T_{sa} - T_{so}) \]  
\[ V = \frac{\dot{m}_{r+wf}}{\rho_{3,tf} - \rho_{1,tf}} \]  
\[ \dot{m}_r = (\rho_{v, ev} + \rho_{4, tf} + \rho_{3, tf}) V \]  

By rearranging Equation 3-6, a value for the recirculating mass flow rate can be calculated. This can be seen in Equation 7. From this, the total flow rate, \( m_{r+wf} \), and thus the working fluid flow rate delivered to the expander, \( m_{wf} \), can be calculated.

\[ \dot{m}_r = \frac{(T_{st} - T_{so})(\rho_{v, ev} + \rho_{4, tf} + \rho_{3, tf})\dot{m}_s c_p}{(\rho_{3, tf} - \rho_{1, tf})(h_3 - h_1) + (h_{3s} - h_3)} \]  

Figure 3 shows the interaction of the \( \dot{m}_{wf+r} \) and \( \dot{m}_r \) in the TFE pump cycle.
The points shown in Figure 3 are defined as follows:

- **Point 1** - Saturated liquid at P1, T1
- **Point 2** - Saturated liquid at P2, T2
- **Point 3** - Saturated vapour at P2, T2
- **Point 4** - Superheated vapour at P1, T4. Isenthalpic expansion through Control valve 2, h3=h4
- **Point 5** - Saturated vapour at P1, T1

By applying the analysis method described above, it is possible to calculate the conditions at all points in the cycle and the mass flow rate of the working fluid. This allows the performance of the cycle to be investigated as described in Section 4.

4. Results and Discussion

The model described above has been used to calculate the cycle efficiency, boiler efficiency, overall efficiency and net power output, as defined in Equations 8, 9, 10 and 11. For the TFC TFE pump case, there is no additional heat input from the evaporator as the TFE pump delivers the fluid in saturated liquid form at the desired temperature ($q_{in, evap} = 0$). The $h_{s,0}$ term denotes the enthalpy at the source fluid dead state, this is taken to be 25°C.

\[
\eta_{cycle} = \frac{w_{out} - w_{in}}{q_{in, pump} + q_{in, evap}} \tag{8}
\]

\[
\eta_{boiler} = \frac{\dot{m}_{wf} q_{in, pump} + q_{in, evap}}{\dot{m}_{s} (h_{s,in} - h_{s,0})} \tag{9}
\]

\[
\eta_{oa} = \eta_{boiler} \cdot \eta_{cycle} \tag{10}
\]

\[
W_{net} = W_{out} - W_{in} \tag{11}
\]

In each case, the maximum source temperature inlet is varied from 90 to 300°C. The evaporation temperature/pressure is varied from an initial state at T1 plus twice the pinch point to 95% of its maximum value, this is to reduce the number of anomalies at temperatures close to the working fluids critical value. The evaporation pressure/temperature is varied to find maximum net power and highest overall efficiency. The results of this can be seen in Section 4.1 below.

Four configurations have been considered, SORC with mechanical pump, SORC with TFE pump, TFC with mechanical pump and TFC with TFE pump. All mechanical pumps modelled have an efficiency of 50%. These results allow a direct comparison with the existing results in the paper by Richardson. The results for the SORC configurations from the current model are in good agreement with the previously published data.

4.1. Cycle and boiler efficiency

The cycle efficiency in figure 5, shows that the TFE pump results in significantly different behaviour when placed in a TFC compared to a SORC. This is due to the additional heat transfer in the evaporator which doesn’t occur in the TFC. As the source fluid temperature increases, the mechanical pump has an increasing advantage over the TFE pump in an SORC, this is due to two main reasons. The first is, as the expander inlet temperature increases the amount of usable heat for the TFE pump is reduced. This results in the reduced efficiency of the pump. Secondly, as the expander inlet temperature (with the source fluid inlet) is increased, the BWR (back work ratio) of the mechanical pump configuration increases, due to needing a higher mass flow. The TFE pump in a TFC consistently has a higher cycle efficiency, due to the exclusion of work input required by the mechanical pump.
Figure 6 shows how effectively the amount of usable heat is recovered. It is not surprising that the TFE pump has lower efficiencies than its counterpart. This is due to the increased input heat required for the recirculation of the TFE pump. Increasing the recirculation mass from saturated water at P1 to saturated vapour at P3. The TFE pump, as with cycle efficiency, performs better in the TFC. This is again due to the additional heat input required to the mass flow rate of the working fluid in the SORC. In both efficiency curves a noticeable change in gradient can be seen. This is at the point where the pressure limit of the working fluid has been reached and no more heat can be removed from the source fluid. This point is higher for the SORC than TFC as the optimum boiler pressure is always lower for a given source fluid inlet temperature.

Compared to previous data, the SORC values are slightly different. This is due to the additional modelling needed of the recirculating mass flow rate. It is this additional mass flow rate that effects both the heat input and heat output of the cycle.

4.2. Overall efficiency

As the overall efficiency is proportional to both the cycle and boiler efficiency, all the curve characteristics can be seen. The sudden change in gradient for the TFC happens at 450K, and as previously stated is related to the critical pressure limit of the working fluid. Similarly the SORC tails off at 525K. Most importantly from Figure 7, the TFC has higher efficiencies at lower source fluid temperatures, <480K. This links directly to the low grade waste heat recovery aim of this paper. On top of this, the TFE pump outperforms the mechanical pump up until 440K, this is still within the low grade heat source realm.
Figure 7. Overall efficiency against source fluid inlet temperature for cycle configurations

4.3. NET Work output

Due to the higher overall efficiencies seen in Figure 7, the net work outputs in Figure 8 have a very similar conclusion. The TFC produces more work output at lower temperatures than the SORC, and the TFE pump in a TFC produces more work output at lower temperatures than a mechanical pump. The higher mass flow rates of the working fluid and the improved temperature matching between the source and working fluid during heat addition allow the TFC to perform better. This is important as the increase in two-phase expander efficiencies have made this cycle more viable. Now with the inclusion of a different pump design and through further modelling, these results suggest that it may be possible to produce more net work output using a TFE pump in a TFC. This current study does however take a relatively simple approach to predicting the performance of these systems. In order to assess the potential benefits of using the TFE in power generation cycles more detailed modelling of both the unsteady TFE flow processes and the efficiency of components such as the expander and mechanical power must also be considered. This will be the focus of future research.

Figure 8. Net Work Output Per Unit Source Mass Flow Rate against source fluid inlet temperature for cycle configurations
5. Conclusion

The aim of this paper was to work on previous TFE pump modelling and apply this to a TFC, with reference back to the existing results. The SORC cycle was validated with previous data produced. A reworked model was used to suit a TFC and the TFE pump was inputted. The results show a performance increase at lower temperatures within a TFC and more specifically with a TFE pump used. At temperatures <440K, the TFC with a TFE pump has 0.2% higher overall efficiencies than a mechanical pump. This shows again in the net work output where the TFE pump produces 0.2% greater net work output than the mechanical counterpart.

Physical comparisons of the two pumps is indeed another necessary factor. Figure 2 shows that the TFE pump can be compact in size with the cylinder volumes being the determining factor of both the size and mass flow rate. In comparison to a mechanical pump with a heat exchanger, the TFE would have a smaller total area due to both the pressurising and heating/condensing processes happening within the same pump. This does add a layer of complexity both in the design and cost of the product, with further controllers needed to adjust the valve opening times.

Future work needs to be conducted in order to validate these initial findings, which should focus on the unsteady fluid flow through the 4 valves in the TFE pump, and their effect on pressure changes within the system. This can then be applied to a whole system analysis to show the effect on additional heat input through the mass flow rate needed in the source and coolant cycles, and ultimately the net work output that can be achieved.

Nomenclature

Symbols

- \(c_p\): Specific heat capacity (J/kg K)
- \(s\): Specific entropy (J/kg)
- \(h\): Specific enthalpy (J/kg)
- \(q\): Specific heat transfer (J/kg)
- \(m\): Mass flow rate (kg/s)
- \(T\): Temperature (°C)
- \(P\): Pressure (pas)
- \(u\): Specific int energy (J/kg)
- \(pp\): Pinch point temperature difference
- \(w\): Specific work transfer (J/kg)
- \(\eta\): Efficiency
- \(\rho\): Density

Subscripts

- \(boiler\): System within boiler
- \(c\): Condenser
- \(co\): Coolant fluid
- \(cycle\): System within cycle
- \(ev\): Evaporator
- \(exp\): Expander
- \(oa\): System of all
- \(0\): Dead state conditions (25°C, 1atm)
- \(s\): Source fluid
- \(wf\): Working fluid
- \(in\): Input of heat or work
- \(out\): Output of heat or work
- \(l\): Liquid state
- \(v\): Vapour state
- \(sat\): Saturated state
- \(r\): Recirculation

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