2012

# Organic Rankine Cycle System Analysis for Low GWP Working Fluids

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## ABSTRACT

The last decade has seen a substantial increase in Organic Rankine Cycle system installations for low temperature waste heat power recovery. The availability of HFC245fa has played a major role in this recent surge in ORC systems since it allows the use of existing HVAC hardware (heat exchangers and compressors) to be used as ORC components (turbines, boilers and condensers) with minimal redesign.

The environmental drawback of HFC245fa is its relatively high GWP value of 950. The advent of a number of new low-GWP refrigerants for HVAC duty which GWP values in the single digits will affect the choice of future working fluids for next generation ORC systems.

This paper will analyze the potential of some of these new refrigerants as working fluids for ORC systems and compare the results against existing ORC systems using HFC refrigerants.

The analysis uses a REFPROP-based in-house developed computer program that for given heat source and heat sink input data and the selection of a working fluid, predicts anticipated ORC efficiency and performs initial component sizing calculations. The analysis accounts for pump and turbine inefficiencies, heat exchanger approach and pinch point temperatures as well as piping pressure drops as experienced in existing installations. It also allows off-design performance analysis, a feature that is important for ORC pay-back calculations given the sensitivity of ORC power output to variations in ambient conditions.

## 1. INTRODUCTION

A large percentage of present day installed ORC systems is for high temperature waste heat recovery applications. There is a growing interest for lower temperature waste heat power recovery ORC systems for applications such as low temperature geothermal, solar thermal, industrial waste heat etc. The working fluids used for high temperature ORC applications may or may not be suitable for low temperature applications. Based on our design and testing experience with a number of low temperature ORC systems [Brasz *et al*., 2005; Conry *et al*., 2011] we have developed a computer program for component sizing and overall performance prediction for low temperature ORC systems. This program is linked to the NIST Refprop 8.0 program for thermodynamic property calculations. It uses demonstrated state-of-the-art turbine and pump component efficiencies, heat exchanger approach and pinch-point temperature differences and piping loss coefficients as default values. These values can be updated to reflect future changes in technology. The calculation results reported in this paper have been obtained using the default values of the program. Most of the fluids currently used for low temperature ORC installations are existing refrigerants such as R134a and R245fa with a high direct-effect global warming potential (GWP). The HVACR industry has engaged in a large R&D effort to find alternative lower GWP working fluids with better or at least equal cycle efficiencies. In this paper we want to extend that investigation to ORC systems in order to determine which of the recently developed low GWP fluids would be suitable for future ORC duty. In this paper we start with first designing a 100 kW ORC system using various working fluids for a low temperature applications with heat source inlet and outlet temperatures of 100 $^{0}$C and 80 $^{0}$C respectively and considering a 15 $^{0}$C ambient temperature. To calculate the annualized benefit of an ORC installation the off-design performance becomes important as well. It plays a major role in the cost effectiveness and investment return times. That is why we also show how the designed ORC systems perform when operating at alternative operating conditions. In Section 7 we also discuss how assumptions such as turbine and pump efficiency and piping losses made during ORC performance prediction affect the predict power output. Low temperature ORC systems can be sensitive to changes in these parameters. We discuss a wide range of working fluids having different GWP values and different critical temperatures.

## 2. ORC SYSTEM OVERVIEW

Figure 1 shows **a** system schematic of an air-cooled ORC system with a liquid heat source. High pressure vapor enter at station 1 the ORC turbine. The expansion of the entering high-pressure vapor over the turbine creates shaft power that is transformed via an electric generator and possible an inverter/ into electrical power. The low pressure superheated vapor exiting the turbine enters the condenser. From there it leaves as slightly subcooled, low-pressure liquid and enters the pump. The pump increases the pressure and moves the refrigerant towards the preheater/ evaporator. The working fluid leaves the evaporator as a slightly superheated high-pressure vapor ready to enter the turbine, from where the cycle repeats itself. Figure 2 shows the corresponding temperature entropy diagram.

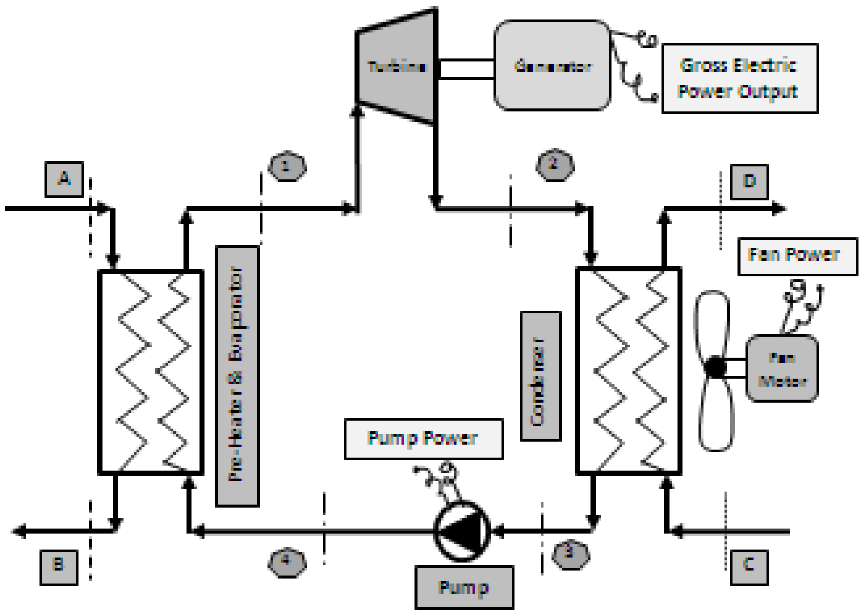


Figure 1: Schematic of the ORC system

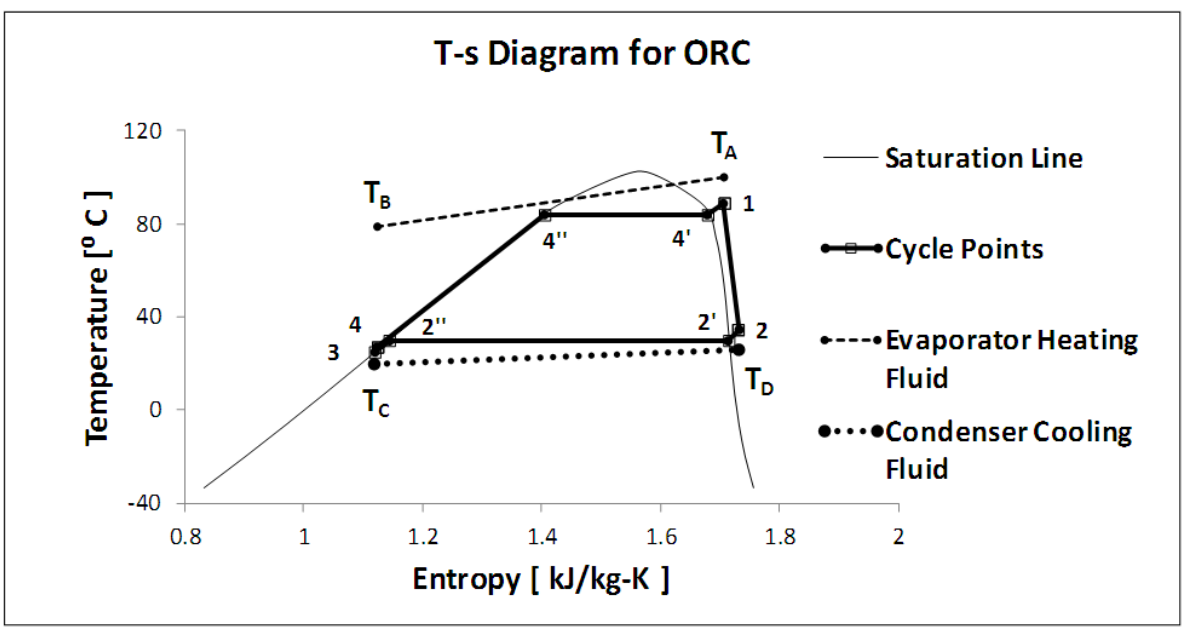


Figure 2. Temperature Entropy diagram off the ORC system

## 3. ORC WORKING FLUIDS

Table 1 lists the working fluids that have been investigated for potential use in ORC systems together with their GWP values, their flammability characteristics and their critical temperatures and pressures. The GWP values were not available for all of the working fluids. The critical temperature gives a first rough indication of ORC cycle efficiency with the higher critical temperature fluids often showing somewhat higher cycle efficiency. All fluid properties were obtained from Refprop with the exception of the properties for DR-2. The properties of this fluid were obtained from approximations based on reference [Kontomaris, 2010].

| Working Fluid [ - ] | GWP [ - ] | Flammable [ - ] | Critical Temperature [˚C] | Critical Pressure [ MPa ] |
|---|---|---|---|---|
| Pentane | 11 | Yes | 196.55 | 3.37 |
| R245fa | 950 | No | 154.01 | 3.65 |
| R134a | 1300 | No | 101.06 | 4.06 |
| R1234ze | 6 | moderately | 109.37 | 3.64 |
| R1234yf | 4 | moderately | 94.70 | 3.38 |
| R123 | 77 | No | 183.68 | 3.66 |
| *DR-2 | 9.4 | No | 171.30 | 2.90 |
| C6FK | 1 | No | 168.66 | 1.87 |
| Toluene | n.a. | Yes | 318.60 | 4.13 |
| D4 | n.a. | moderately | 313.34 | 1.33 |
| D5 | n.a. | moderately | 346.08 | 1.16 |

Table2: GWP, flammability, critical temperature and critical pressures of the various fluids used for analysis

* Note: Calculations for DR-2* are from approximations based on reference [Kontomaris, 2010], and they may vary quite a bit from a standard calculation procedure.

## 4. CALCULATION RESULTS FOR DESIGN CONDITIONS

We used the program to develop a 100 kW gross electrical turbine output power ORC system for the various working fluids listed in Table 2. We assume an available liquid heat source temperature $T_A$ of 100˚C. The program uses the evaporator pinch point as a default input to calculate the refrigerant saturation temperature in the evaporator. The resulting heat source fluid temperature leaving the evaporator in this case $T_B$ is 80 ± 0.5℃. This part of the calculation is shown in Equations 1a, 1b where the state point subscripts refer to Figure 2. The ambient temperature of 15˚C is taken as the design condition and an air cooled condenser with a saturation temperature of 30 ˚C corresponding to this ambient temperature is considered. Since we know the $T_A$, $T_B$, $\Delta T_{pinch}$ and assuming a constant heat capacity for the thermal heat source fluid, we can iteratively solve Equations 1a and 1b to obtain the evaporator saturation temperature.

$$\frac{\dot{M}_{th}C_p(T_A - T_B)}{\dot{M}_{th}C_p[T_A - (T_{evap,sat} + \Delta T_{pinch})]} = \frac{\dot{M}_{ref}\Delta H_{1,4}}{\dot{M}_{ref}\Delta H_{1,4''}} \quad \text{(1a)}$$

$$\frac{(T_A - T_B)}{[T_A - (T_{evap,sat} + \Delta T_{pinch})]} = \frac{\Delta H_{1,4}}{\Delta H_{1,4''}} \quad \text{(1b)}$$

The net electric power out, heat exchanger size, system efficiency and impeller size for the various working fluids can be seen in Figure 3, Table 3 and Table 4. The results show that the highest thermal efficiency is obtained with toluene as working fluid. The major limitation of toluene is its flammability and toxicity and to some extent its turbine size which is of the order 2 to 4 times the size of the low temperature refrigerants. The other high temperature fluids like siloxanes D4, D5 have good thermal efficiencies but they result in impractical turbine sizes, extremely high pressure ratios and very low sub-atmospheric condenser side pressures, posing added complexity in system design. Of the low temperature working fluids pentane, R245fa, R123, and DR-2* show similar level of performance with reasonable impeller sizes and speeds. Pentane has a low GWP value but its flammability might be an issue. R245fa has a high GWP among these refrigerants while R123 has a much lower GWP than most currently used refrigerants but it has ODP value of 0.02 prohibiting its use in future installations. Its new replacement fluid DR-2* has a very similar performance with a still lower GWP value and zero ODP [Kontomaris, 2010]. Considering both non flammability and low GWP as selection criteria DR-2* could be an ideal fluid to be used for low temperature ORC applications. If flammability is not an issue and just a low GWP is the selection criterion pentane would be the best ORC working fluids for low temperature applications.

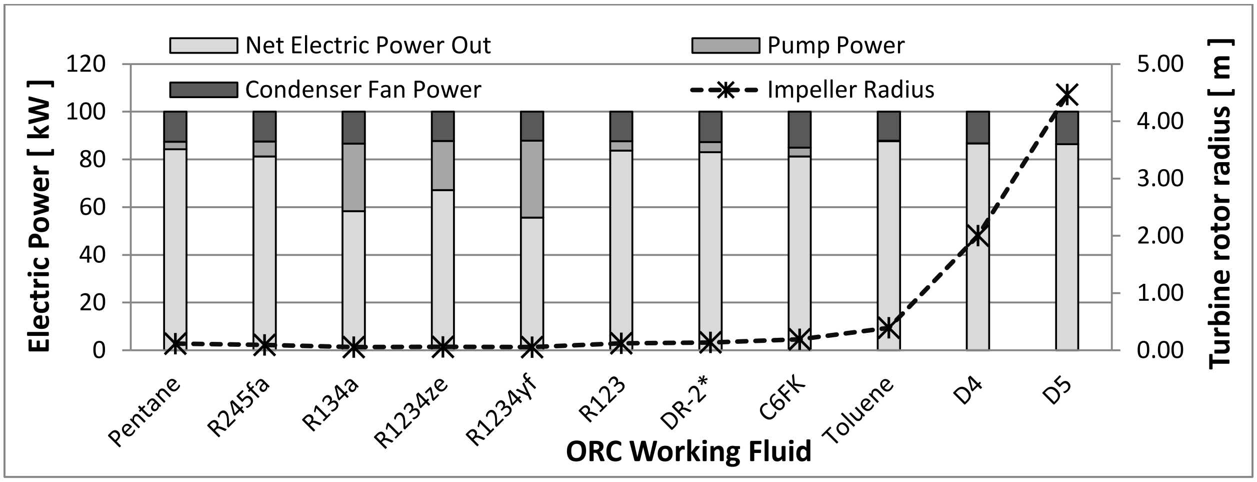


Figure 2: Power split between components of the ORC system and the turbine sizes for various working

| Working Fluid<br>[ - ] | Evaporator Capacity<br>[ kW ] | Condenser Capacity<br>[ kW ] | Pressure Ratio<br>[ - ] | Impeller Radius<br>[ m ] |
|---|---|---|---|---|
| Pentane | 1,111 | 1,007 | 4.58 | 0.12 |
| R245fa | 1,102 | 1,001 | 4.61 | 0.09 |
| R134a | 1,152 | 1,073 | 3.71 | 0.06 |
| R1234ze | 1,067 | 980 | 3.74 | 0.06 |
| R1234yf | 1,043 | 968 | 3.62 | 0.05 |
| R123 | 1,092 | 988 | 4.52 | 0.12 |
| DR-2* | 1,125 | 1,018 | 4.85 | 0.14 |
| C6FK | 1,306 | 1,202 | 5.86 | 0.19 |
| Toluene | 1,075 | 968 | 7.67 | 0.39 |
| D4 | 1,166 | 1,058 | 20.57 | 2.00 |
| D5 | 1,197 | 1,090 | 29.15 | 4.47 |

Table 3: ORC system evaporator, condenser, impeller sizing and turbine pressure ratio

| Working Fluid<br>[ - ] | Pump Power<br>[ kW ] | Condenser Fan Power<br>[ kW ] | Net Electric Power Out<br>[ kW ] | Thermal Efficiency @ Net Electric Power Out<br>[ % ] |
|---|---|---|---|---|
| Pentane | 3.17 | 12.59 | 84.24 | 7.58 |
| R245fa | 6.28 | 12.52 | 81.20 | 7.36 |
| R134a | 28.28 | 13.42 | 58.30 | 5.06 |
| R1234ze | 20.61 | 12.25 | 67.14 | 6.29 |
| R1234yf | 32.32 | 12.11 | 55.57 | 5.32 |
| R123 | 3.88 | 12.36 | 83.76 | 7.67 |
| DR-2* | 4.19 | 12.73 | 83.08 | 7.38 |
| C6FK | 3.71 | 15.03 | 81.26 | 6.22 |
| Toluene | 0.23 | 12.10 | 87.67 | 8.15 |
| D4 | 0.05 | 13.24 | 86.71 | 7.43 |
| D5 | 0.01 | 13.63 | 86.36 | 7.21 |

Table 4: Auxiliary systems power consumption and ORC system net thermal efficiencies

## 6. CALCULATION RESULTS FOR OFF-DESIGN CONDITIONS

The program also lets us calculate off design performance at various ambient temperatures. For this study a range of ambient temperatures from 0 $^0$C to 45 $^0$C are taken. We run the various ORC systems at these off-design conditions by keeping the turbine, condenser and evaporator sizes fixed. We assume that the amount of heat supplied to the preheater/evaporator stays constant. For temperatures below design condition ORC cycle efficiency improves and the turbine could increase power output if the electric generator had extra capacity. Assuming that the turbine gross power output cannot increase due to limited electrical generator capacity the net power output of the system could still increase by reducing the fan speeds of the air-cooled condensers. Figure 3 shows the variation of net electric power output from the ORC system at various ambient temperatures. As there is a significant change in output at various ambient conditions, the selection of the design point based on geographical location would have a large impact on the total amount of electrical energy produced in a year (kWh/year), which in turn affects the payback period. It is important to realize the steep drop in ORC output power with increase in ambient temperature. All thermal power plants experience a reduction in net power output at higher ambient temperatures. However, this sensitivity is much stronger for low temperature ORC systems than for any other power plant. This strong variation of output power with changes in ambient conditions requires an annual bin-hour analysis to determine the annual electrical energy production, similar to what the HVACR industry does to calculate annual electrical energy consumption of its systems.

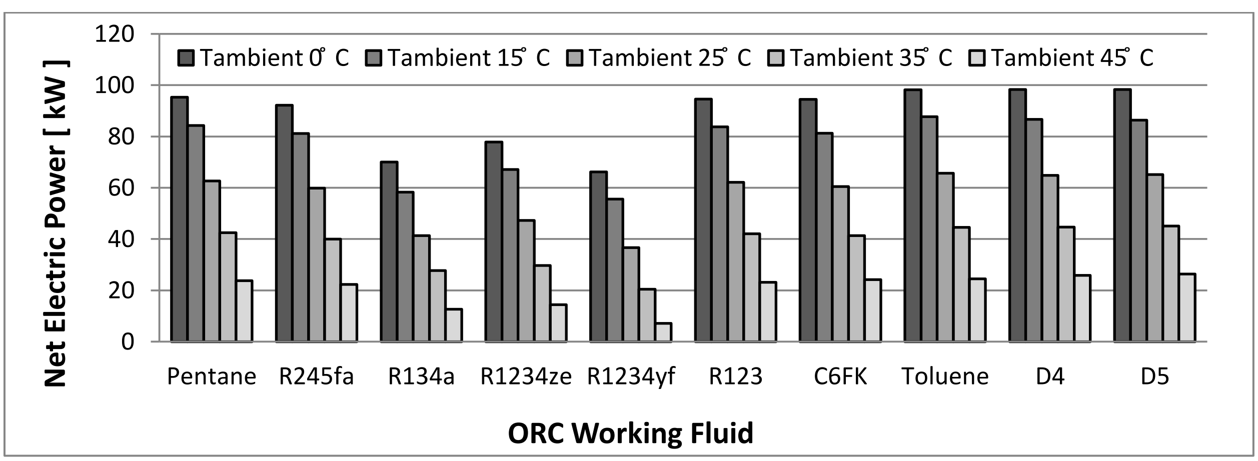


Figure 3: Net Power Output from the ORC system at various ambient temperatures

## 7. OVERLY OPTIMISTIC ORC PERFORMANCE PREDICTIONS

Cycle analysis of ORC performance analysis is often carried out with overly optimistic assumptions for turbine and pump efficiencies and heat exchanger pinch point and approach temperature differences while neglecting piping and heat exchanger pressure drops. The program used for the 100 kW ORC analysis as presented in this paper accounts for all system losses and assumes realistic machinery efficiencies and heat exchanger approach and pinch point temperature differences based on experience with actual systems. When pressure drops are neglected and overly optimistic pump efficiencies of 70%, expander efficiencies of 90% and electrical generator efficiencies of 95% are assumed, 20-25% better efficiencies are calculated as can be seen from Figure 4. However, components with these efficiency levels do not exist for the high pressure ratio, low flow conditions encountered in a 100 kW ORC system.

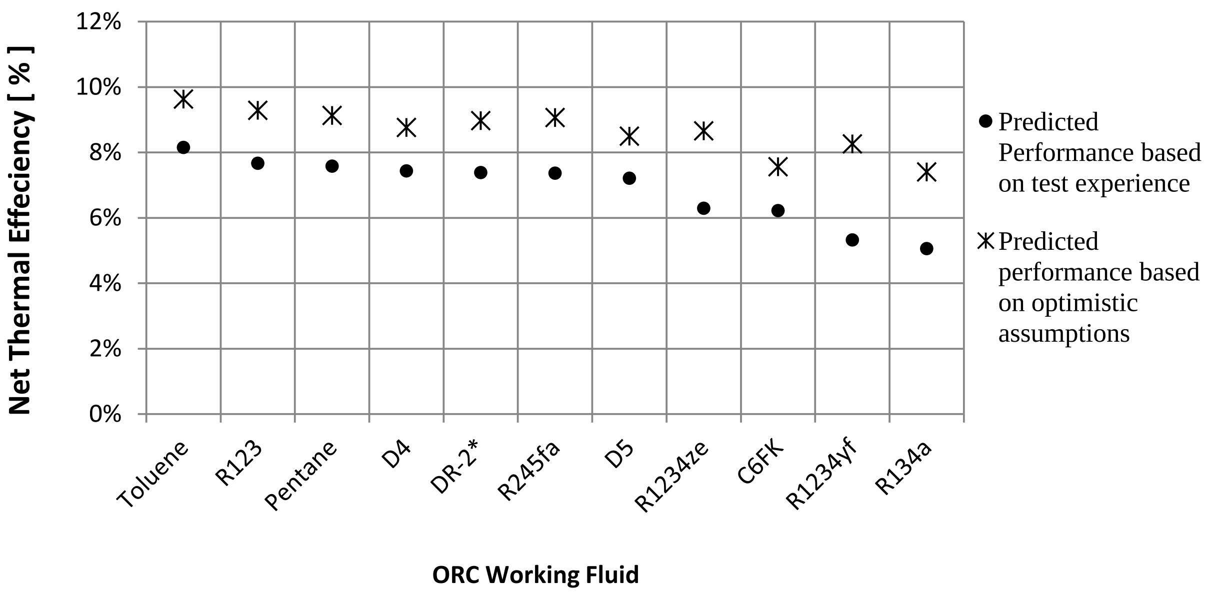


Figure 4: Net Thermal Efficiency of ORC system

## 8. CONCLUSIONS

- Most low-temperature ORC systems use working fluids, known as refrigerants in the HVACR industry, with high global warming potential (GWP ~ 1000).
- R245fa has been a popular fluid for low temperature ORC systems, thanks to its ability to use R134a compression equipment as ORC expander units. However, its GWP is 950.
- New refrigerants with substantially lower global warming potential have recently been introduced for the HVACR industry.
- An in-house performance analysis and component sizing program, calibrated against existing low temperature ORC installations, has been applied to assess the viability of some of these new low-GWP refrigerants as future working fluids for ORC systems and compared against existing known ORC working fluids.
- DR2, a new low-pressure refrigerant to replace R123 in centrifugal chillers has impressive environmental characteristics (e.g. a GWP = 4) and shows good performance for ORC duty, but it requires larger equipment size than R245fa, eliminating the direct transformation of air-conditioning compressors into ORC expanders/turbines.

- The medium pressure new refrigerants R1234yf and R1234ze(E) approach R134a in performance and result therefore in lower ORC cycle efficiencies than R245fa.
- The low-temperature ORC industry still awaits the arrival of a low GWP replacement fluid for R245fa to maintain the cost advantage that can be realized by using existing HVAC equipment for ORC duty. The newly developed fluid 1233zd(E) [Hulse, 2010] could be that fluid.

## NOMENCLATURE

| | |
|---|---|
| ORC | Organic Rankine Cycle |
| PR | pressure ratio |
| T | temperature |
| P | pressure |
| $\eta$ | efficiency |
| GWP | Global Warming Potential |
| ODP | Ozone Depletion Potential |
| $\dot{M}$ | mass flow rate |
| **Subscripts** | |
| evap | evaporator |
| cond | condenser |
| is | isentropic |
| sat | saturated |
| therm | thermal |
| ref | refrigerant |
| 1 | turbine Inlet |
| 2 | turbine outlet |
| 2’ | condenser vapor saturation point |
| 2’’ | condenser liquid saturation point |
| 3 | condenser outlet |
| 4 | pump outlet |
| 4’’ | evaporator liquid saturation point |
| 4’ | evaporator vapor saturation point |

## REFERENCES

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