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Lars J. Brasz  
*SUNY at Buffalo*

William M. Bilbow  
*WMB-Enterprises*

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# RANKING OF WORKING FLUIDS FOR ORGANIC RANKINE CYCLE APPLICATIONS

Lars J. BRASZ<sup>1</sup>, William M. BILBOW<sup>2</sup>

<sup>1</sup>SUNY at Buffalo,  
Buffalo, NY 14260  
[ljbrasz@buffalo.edu](mailto:ljbrasz@buffalo.edu)

<sup>2</sup>WMB-Enterprises,  
Baldswinville, NY 13027  
[wbilbow@wmb-enterprises.com](mailto:wbilbow@wmb-enterprises.com)

## ABSTRACT

Renewed interest in waste heat power recovery has resulted in the development of Organic Rankine Cycle (ORC) products with the recently commercialized refrigerant R-245fa [1,2,3]. A thermodynamic cycle analysis program predicting the performance of ORC systems has been written in support of the development of these products. This cycle program allows the use of different organic working fluids and various sources of waste heat. It also accounts for actual turbine and pump efficiencies, refrigerant line pressure losses and finite boiler and condenser surface area. Using this cycle program, the relative thermodynamic efficiency of various potential working fluids is determined for three waste heat power recovery applications: gas engine exhaust, landfill flare gas and condensing steam. The analysis is carried out for both recuperated and non-recuperated ORC cycle configurations and includes water-cooled as well as air-cooled condenser options.

The differences in ORC thermodynamic cycle efficiency of various refrigerants were found to be substantially larger than the differences in thermodynamic efficiency of these same fluids for vapor compression cycle applications. Second law analysis of the results explains these findings. ORC thermodynamic efficiency correlates well with the critical temperature of the working fluid. Future ORC systems could benefit from the development of safe and environmentally acceptable working fluids with higher critical temperatures than currently available.

## 1. INTRODUCTION

The thermodynamic cycle efficiency of the refrigeration cycle is influenced by the choice of refrigerant. The ban on CFC's and HCFC's by the Montreal protocol forced the HVAC industry to work with thermodynamically less efficient refrigerants. Replacing CFC11 with HFC134a in centrifugal chillers caused a 6% drop in cycle efficiency. Similarly replacing HCFC22 with HFC410A in air-cooled equipment caused a 5% drop. Fortunately, advances in compression and heat transfer technology have in most cases more than compensated for the lower cycle efficiency of the environmentally friendlier refrigerants. Modern HVAC products have better overall system efficiency despite their use of less efficient refrigerants.

Organic Rankine Cycle systems face similar challenges as HVAC products. The non-flammable and non-toxic refrigerants promoted in the past as attractive working fluids [4], i.e. CFC113, CFC114, CFC11 and HCFC123, have already been outlawed or will be outlawed in the near future because of their ozone layer depletion potential. This paper compares the thermodynamic cycle efficiency of these refrigerants with some of the newer refrigerants, i.e. HFC245fa, HFC236fa and HFC134a, when used as working fluids in organic Rankine cycle applications.

To cover a number of applications this study considered two types of cycles: the simple organic Rankine cycle and the recuperated cycle. Different evaporation temperatures were selected, based on the type of waste heat available (high temperature gas, waste heat steam at ambient pressure and low temperature liquid). The condensation temperatures varied based on the choice of condenser, i.e. water-cooled versus air-cooled.

## 2. SIMPLE (NON-RECUPERATED) ORGANIC RANKINE CYCLE

The Organic Rankine cycle allows power recovery of waste heat streams that are too low in temperature to be cost-effectively converted into power by conventional steam cycles. The use of organic fluids, such as refrigerants, enables efficient and cost-effective power recovery of waste heat streams. A schematic picture of the simple or non-recuperated organic Rankine cycle is shown in Figure 1 and consists of four major components: turbine, condenser, pump and evaporator.

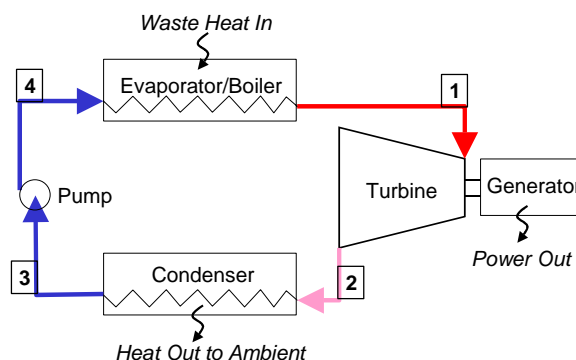


Figure 1. Schematic of non-recuperated ORC system

The following processes take place in the simple (non-recuperated) ORC system. High-pressure vapor enters a turbine (state point 1). In the turbine mechanical energy is recovered through controlled expansion of high-pressure vapor and transferred into electrical energy through an electrical generator. The low-pressure vapor leaving the turbine at state point 2 enters a condenser where heat is rejected from the working fluid to and condensation takes place. The working fluid leaves the condenser as low-pressure, slightly sub-cooled liquid. (state point 3). A pump raises the pressure of the liquid to state point 3 and feeds it to an evaporator. In the evaporator heat is extracted from a waste heat source to preheat, evaporate and superheat the working fluid, thus modifying high pressure liquid into high pressure vapor (state point 1) ready to enter the turbine again. The pressure-enthalpy and the temperature-entropy diagrams of the simple organic Rankine cycle with R245fa as the working fluid are shown in Figures 2 and 3, respectively. The temperature-entropy diagram includes the heat source (AB) to identify potential pinch-point problems.

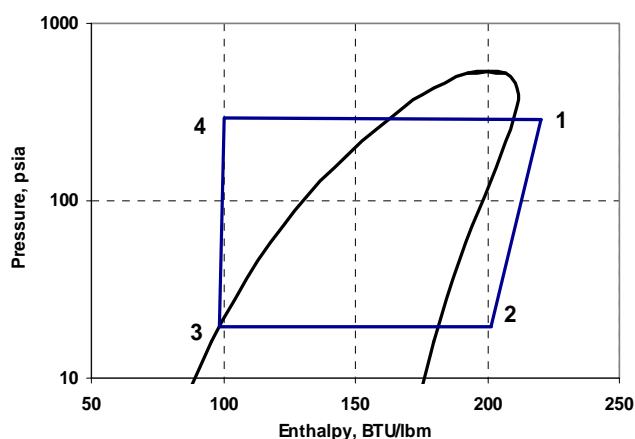


Figure 2. Pressure enthalpy diagram of ORC cycle

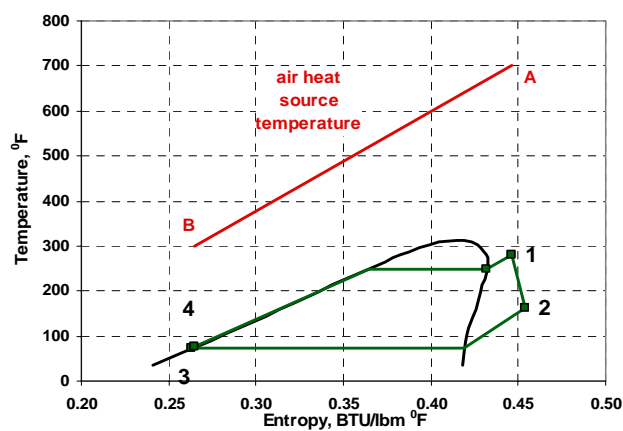


Figure 3. Temperature entropy diagram of ORC cycle

### 3. RECUPERATED ORGANIC RANKINE CYCLE

For higher temperature waste heat streams, the thermal efficiency of the organic Rankine cycle system can for some working fluids be improved dramatically by the use of a recuperative preheater. The recuperated ORC system uses the heat required to de-superheat the turbine exhaust vapor to preheat the cold condensed liquid leaving the pump before being fed to the evaporator. Figure 4 shows a schematic diagram of a recuperated organic Rankine cycle system and Figures 5 and 6 show the corresponding pressure-enthalpy and temperature entropy diagrams for a recuperated ORC system with R245fa as the working fluid.

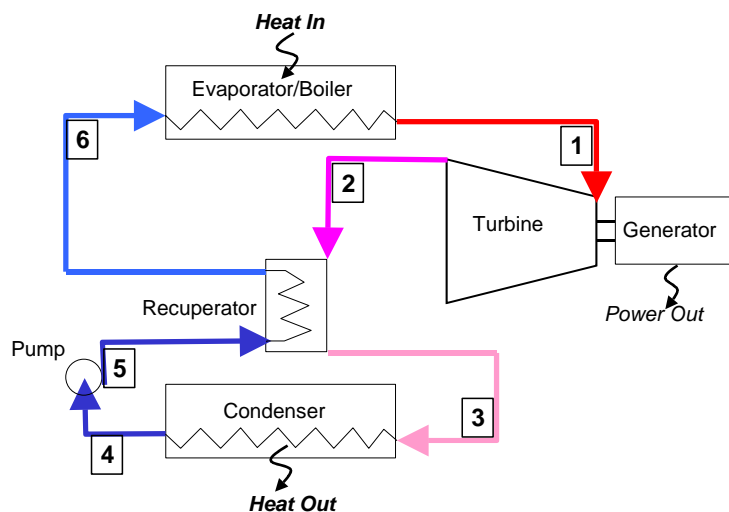


Figure 4. Schematic of recuperated ORC system

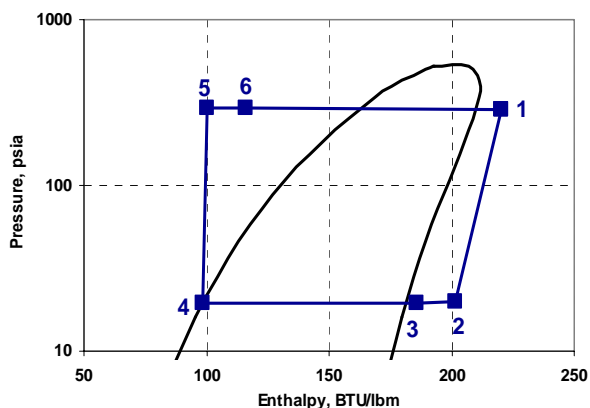


Figure 5. PH diagram of recuperated ORC cycle

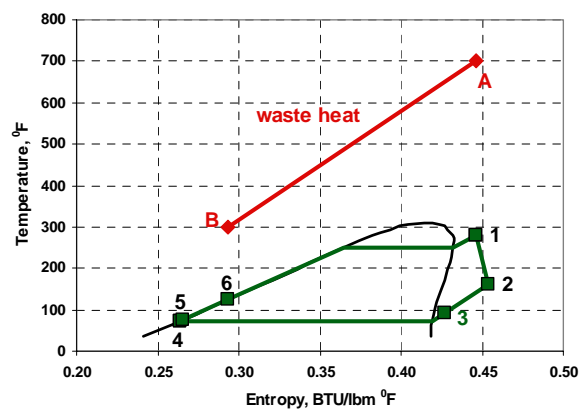


Figure 6. TS diagram of recuperated ORC cycle

#### 4. THE LIMITATION OF THE THERMAL EFFICIENCY CONCEPT FOR ORC APPLICATIONS

The objective of an ORC system is to maximize the power recovery of a waste heat stream. ORC thermal efficiency is defined as net power out divided by thermal waste heat in:

$$\eta_{thermal} = \frac{\dot{P}_{net,out}}{\dot{Q}_{thermal,in}} \quad (1)$$

The second law of thermodynamics limits the maximum efficiency that can be achieved with low temperature waste heat streams. When heat is added at a constant temperature, e.g. when a latent heat source such as condensing steam is available, the maximum efficiency is determined by the Carnot cycle efficiency:

$$\eta_{max} = \frac{T_H - T_L}{T_H} \quad (2)$$

where  $T_H$  is the absolute temperature of the condensing steam from which heat is absorbed and  $T_L$  is the absolute temperature at which heat is rejected. The availability of atmospheric steam at a 59 °F day will result in a maximum theoretical efficiency

$$\eta_{max} = \frac{T_H - T_L}{T_H} = \frac{212 - 59}{212 + 460} = 0.24 \quad (3)$$

System losses are caused by turbine and pump inefficiencies, temperature differences in heat exchangers between the working fluid and the heat source and heat sink media as well as pressure drops in connecting piping. As a consequence ORC thermal efficiencies are substantially lower than the theoretical maximum.

In many instances sensible heat is the predominant component of a waste heat stream. The maximum thermal efficiency of an Organic Rankine Cycle system with a sensible waste heat source, which is cooled down from an inlet absolute temperature  $T_{in}$  to an outlet absolute temperature  $T_{out}$  and a heat rejection absolute temperature  $T_L$ , is [5]:

$$\eta_{max} = 1 - \frac{T_L}{T_{in} - T_{out}} \ln\left(\frac{T_{in}}{T_{out}}\right) \quad (4)$$

This equation shows that the maximum efficiency is higher when  $T_{out}$  is closer to  $T_{in}$ , in other words, when less heat is extracted from the waste heat stream. In practice maximizing waste heat power recovery is more important than thermal efficiency. Assuming that there is no other potential use for the waste heat stream than power recovery,  $T_{out}$  should be selected to maximize waste heat power recovery instead of maximizing thermal efficiency of the system, i.e. as low as economically possible.

#### 5. WASTE HEAT STREAMS

The most common waste heat streams that lend themselves for power recovery by use of organic Rankine cycle equipment are:

1. Exhaust gases from engines or incinerators. Engine exhaust temperatures, either from a gas turbine or recip engine, vary between 600 and 800 °F. Incinerator exhaust gas temperatures are even higher. A refrigerant evaporator extracts the thermal energy from these waste heat streams by cooling them down. The actual gas composition of the exhaust waste heat dictates how much heat can be extracted from these waste heat streams before condensation of acidic byproducts causes heat exchanger corrosion concerns. Practical limits for minimum cooled-down gas temperature vary, depending on the chemical composition of the waste heat gas stream, between 175 to 300 °F.

2. Low temperature wet steam from industrial processes at just-above atmospheric pressures. Under these conditions all heat is supplied at a constant temperature but this temperature is at a much lower level, around 225 °F.
3. Hot liquid waste heat. In these cases a secondary fluid, a brine or oil, is circuited through a process or engine as a cooling medium and is traditionally reduced in temperature in some type of radiator. Replacing the radiator with an organic Rankine cycle enables power recovery from this waste heat. Typical temperatures for these types of liquid waste heat streams are from around 225 °F incoming to about 150 °F returning.

The condenser of an organic Rankine cycle can be either air-cooled or water-cooled. Water-cooled condensation results in lower condensing temperatures (around 72 °F at a 59 °F ISO day) and therefore higher cycle efficiency than can be achieved with the 95 °F condensation temperature more typical for air-cooled condensers under those conditions. The trade-off to the higher cycle advantage of water-cooled condenser ORC systems is the need for a secondary water loop with pump and cooling tower and its corresponding water consumption. Also, the power consumption of the water pump and the cooling tower fans is typically larger than the power required by the air-cooled condenser fans.

## **6. WORKING FLUID SELECTION AND BOUNDARY CONDITIONS/ASSUMPTIONS FOR ORC ANALYSIS**

The analysis was limited to non-flammable, non-toxic organic Rankine cycle working fluids. Some of the now outlawed fluids are compared against some of the newer HFC refrigerants that are currently considered for ORC duty. The analyzed fluids have been limited to commercially available, non-flammable, non-toxic fluids.

The analysis has been done for the following eight cases:

1. hot gas:  $T_{in} = 700$  °F,  $T_{out} = 300$  °F without recuperation, with air-cooled condenser
2. hot gas:  $T_{in} = 700$  °F,  $T_{out} = 300$  °F without recuperation, with water-cooled condenser
3. hot gas:  $T_{in} = 700$  °F,  $T_{out} = 300$  °F with recuperation, with air-cooled condenser
4. hot gas:  $T_{in} = 700$  °F,  $T_{out} = 300$  °F with recuperation, with water-cooled condenser
5. condensing steam:  $T_{cond} = 225$  °F with air-cooled condenser
6. condensing steam:  $T_{cond} = 225$  °F with water-cooled condenser
7. hot liquid:  $T_{in} = 225$  °F,  $T_{out} = 150$  °F and air-cooled condenser
8. hot liquid:  $T_{in} = 225$  °F,  $T_{out} = 150$  °F and water cooled condenser'

Recuperation is not considered for cases 5-8 since at these low evaporator saturation temperatures not much condenser superheat is available for preheating.

The cycle calculations are initially done for the "ideal" case, meaning 100% turbine and pump efficiency and no pressure drop in heat exchangers and connecting piping. In the ideal case the power required to move the heat source and heat sink fluids over the evaporator and condenser is assumed negligible. In other words, parasitic power consumption of condenser fans and cooling water pumps and the negative effect of heat source back-pressure on engine output are neglected. Minimum pinch point temperature of heat exchangers was assumed to be 20 °F for the evaporators and recuperators and 35 for air-cooled condensers and 5 °F for water-cooled condensers, both for the ideal case and the actual case. In other words the ideal case had the same saturation temperatures as the actual case but all internal refrigerant flow losses as well as the parasitic power consumption of auxiliary equipment were neglected.

## **7. COMPARING ORC SYSTEMS WITH EVAPORATION AT 95% OF CRITICAL PRESSURE**

The thermodynamic calculations were carried out with a cycle program developed for an organic Rankine cycle system designed with R245fa. The original purpose of this program was to analyze system performance at various pressures and temperatures. High evaporator saturation pressures and temperatures are possible for cases 1 through 4 and tend to increase ORC system efficiency. For these conditions the main thermodynamic irreversibility in the ORC system is due to the temperature difference between the heat source and the refrigerant. Using refrigerants with a higher critical temperature reduces the temperature difference between the heat source and the refrigerant in

the evaporator, thus reducing the heat transfer irreversibility at the price of a larger and therefore more expensive evaporator. Figure 7 shows the lower heat transfer irreversibility for an ORC system with CFC113 compared to HFC245fa, both evaporating at 95% of its critical temperature, resulting in a 23% thermal efficiency for CFC113 versus 18% for HFC245fa for the ideal cycle with a 95 °F condenser saturation temperature.

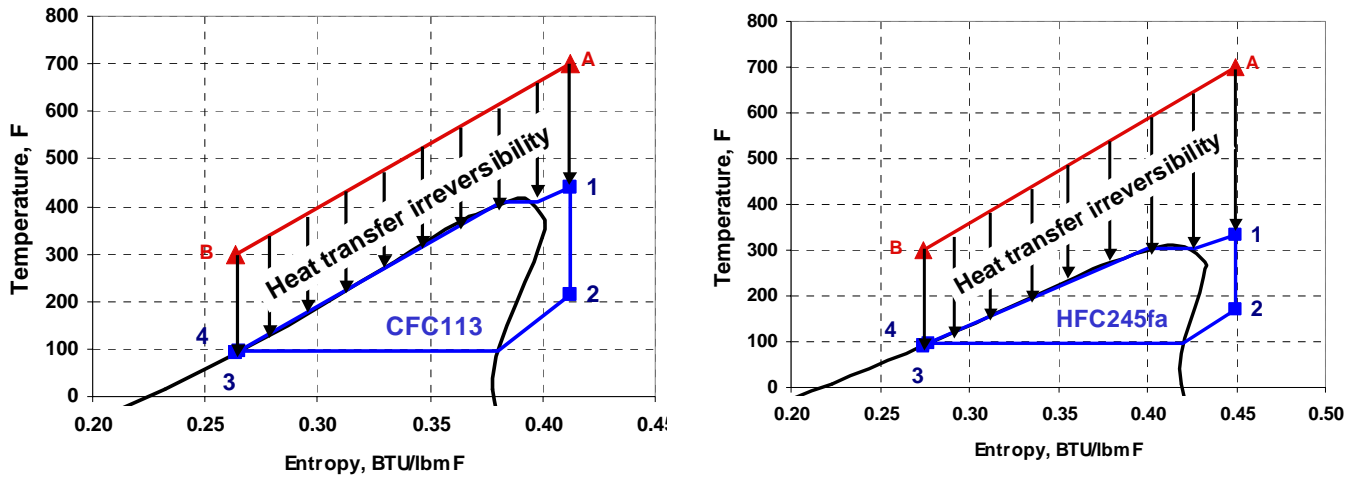


Figure 7. Effect of refrigerant properties on evaporator heat transfer irreversibility.

It becomes clear from Figure 7 that refrigerants with a higher critical temperature will have a better thermal efficiency. Figure 8 plots the thermal ORC efficiencies for a number of refrigerants as a function of critical temperature. The assumptions for these calculations are:

- a sensible heat source which is cooled down in the evaporator from 700 to 300 °F.
- a refrigerant saturation pressure in the evaporator equal to 95% of the critical pressure.
- a condenser saturation temperature of 72 °F.

The calculations have been carried out for the recuperated and the non-recuperated cycle, both ideal (no component losses) and actual (80% turbine and 40% pump efficiency and 2% pressure drop in all connecting piping)

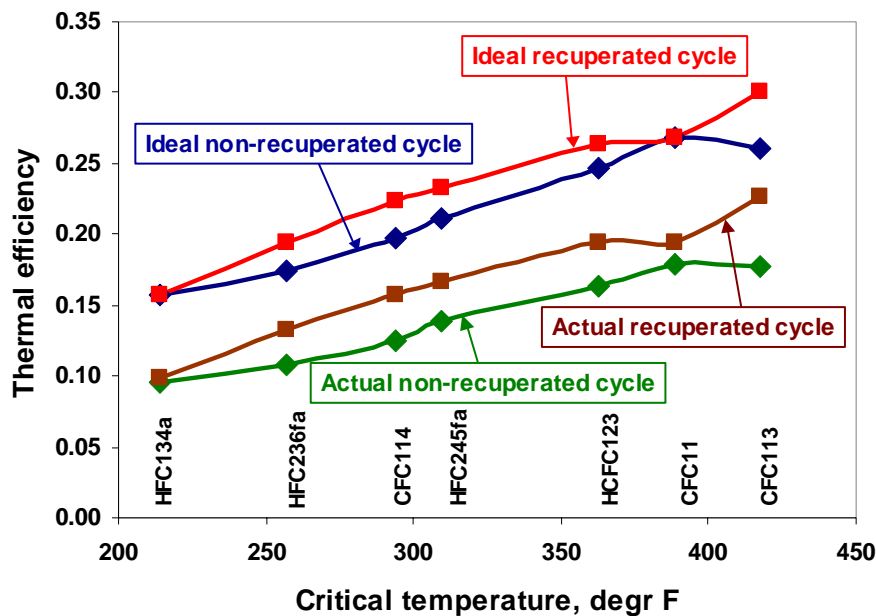


Figure 8. Thermal efficiency of ORC systems as a function of the critical temperature of the working fluid

## 8. COMPARING ORC SYSTEMS WITH EVAPORATION AT EQUAL SATURATION TEMPERATURES

When the waste heat is free, thermal efficiency is only important as far as it is capable of reducing overall system cost per kW of net power produced. Higher refrigerant operating temperatures can easily result in higher equipment cost. It was therefore decided to limit the design saturation temperature in the evaporator to 250 °F. This allows the use of existing HVAC turbo-machinery compressors as radial inflow turbines without major modifications. The remaining studies of cases 1 through 8 follow this refrigerant temperature limitation. All preheaters/evaporators are for analysis purposes considered to be of the counter-flow heat exchanger type.

Combined co-flow/counter-flow evaporator designs are sometimes used with higher waste heat source gas temperatures to limit the heat exchanger metal temperatures.

The lower waste heat supply temperatures of cases 5 through 8 will obviously result in evaporator saturation temperatures less than 250 °F. For these cases the evaporator saturation temperatures are not necessarily equal since some of the refrigerants suffer from a so-called "pinch-point" problem. The actual saturation temperature in the evaporator was determined by requiring a minimum temperature difference between the heat source and the refrigerant of 20 °F. This minimum temperature difference, the pinch-point, occurs in evaporators at the point where preheating is completed and evaporation starts. For counter-flow recuperators a pinch point occurs between the low-pressure vapor outlet side and the high-pressure liquid inlet side. This minimum temperature difference was assumed to be 20 °F in the calculations. The results of the calculations are summarized in Tables 1 and 2.

**Table 1. Thermal efficiency of various Organic Rankine Cycle applications for different refrigerants assuming reversible operation**

*(100% efficient pump and turbine and no line losses)*

	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6	Case 7	Case 8
Condenser:	A/C	W/C	A/C	W/C	A/C	W/C	A/C	W/C
Heat source:	700 F => 300 F		700 F => 300 F		225 F		225 F => 150 F	
Cycle:	non-recuperated		recuperated		non-recuperated		non-recuperated	
R11	.184	.209	.186	.210	.140	.168	.069	.109
R123	.175	.199	.187	.211	.136	.163	.070	.110
R113	.173	.196	.191	.217	.135	.161	.069	.108
R114	.161	.183	.183	.207	.128	.153	.072	.109
R245fa	.166	.189	.183	.208	.131	.157	.072	.110
R236fa	.155	.177	.175	.199	.124	.148	.074	.111
R134a	n/a	n/a	n/a	n/a	.128	.152	.078	.116

**Table 2. Thermal efficiency of various Organic Rankine Cycle applications for different refrigerants assuming realistic component efficiencies**

*(40% pump efficiency, 80% aerodynamic turbine efficiency, 95% mechanical turbine efficiency, 96% electrical generator efficiency, 2% pressure drop in the lines between components)*

Working fluid	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6	Case 7	Case 8
R11	.130	.148	.136	.156	.100	.120	.046	.078
R123	.123	.141	.137	.157	.096	.116	.047	.079
R113	.124	.142	.143	.164	.097	.117	.047	.078
R114	.107	.124	.126	.147	.086	.105	.046	.076
R245fa	.114	.131	.131	.151	.091	.110	.047	.078
R236fa	.098	.115	.115	.135	.081	.100	.047	.078
R134a	n/a	n/a	n/a	n/a	.072	.092	.045	.075



## 9.0 CONCLUSIONS

1. Actual values of ORC thermal efficiencies are low due to the limited thermodynamic availability of waste heat.
2. Maximizing thermal efficiency for sensible heat sources leads to under-utilization of the waste heat stream.
3. Dramatically higher thermal efficiency can be obtained for high-temperature waste heat streams by using the higher evaporation temperatures possible for working fluids with higher critical temperatures.
4. At constant evaporation saturation temperatures the ranking of refrigerants for the non-recuperated ORC systems depends both the critical temperature which controls the pre-heater/evaporator heat transfer irreversibility and the slope of the saturated vapor line in the temperature entropy diagram which controls the de-superheating heat transfer irreversibility.
5. Recuperating the ORC system reduces the effect of de-superheating irreversibility on ORC thermal efficiency.
6. If the critical temperature of the working fluid is higher than the waste heat source temperature evaporator pinch-point problems are likely for high critical temperature refrigerants. Working fluids with a lower critical temperature lead to more efficient systems for the ideal ORC cycle (100% turbine and pump efficiency and no piping losses)
7. Relatively low pump efficiencies and actual piping pressure drops penalize the higher-pressure working fluids, i.e. the refrigerants with lower critical temperatures. Inclusion of actual component losses will favor the higher critical temperature working fluids despite their theoretical pinch-point problems.
8. Future ORC systems could benefit from the development of safe and environmentally acceptable working fluids with higher critical temperatures than currently available.

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