Performance of a Combined Organic Rankine Cycle and Vapor Compression Cycle for Heat Activated Cooling


School of Mechanical, Industrial, & Manufacturing Engineering
204 Rogers Hall, Oregon State University
Corvallis, OR 97331, USA

* Corresponding Author
Phone: 541-713-1354, Fax: 541-758-9320, Email: wanghai@engr.orst.edu

Abstract
Heat activated cooling has the potential of utilizing thermal sources that currently go unused such as engine exhaust heat or industrial waste heat. Using these heat sources can provide enhanced energy utilization and reduced fuel usage in applications where cooling is needed. The concept developed here uses waste heat from stationary and mobile engine cycles to generate cooling for structures and vehicles. It combines an organic Rankine cycle (ORC) with a conventional vapor compression cycle. A nominal 5 kW cooling capacity prototype system was developed based on this concept and tested under laboratory conditions. In order to maintain high system performance while reducing size and weight for portable applications, microchannel based heat transfer components and scroll based expansion and compression were used. Although the system was tested off of its design point, it performed well achieving 4.4 kW of cooling at a measured heat activated COP of 0.48. Both the conversion and 2nd law efficiencies were close to the model results, proving it to be an attractive technology. The measured isentropic efficiency of the scroll expander reached 84%, when the pressure ratio was close to the scroll intrinsic expansion ratio. The reduced cooling capacity was attributed to off design operation.

Keywords: Organic Rankine Cycle, ORC, Heat Activated Cooling, COP, Waste Heat Recovery, Scroll Expander, Microchannel, R245fa, Efficiency
Nomenclatures

\( COP_c \) Cooling cycle coefficient of performance

\( COP_s \) Overall system coefficient of performance

\( \dot{E}_c \) Exergy destruction of the cooling cycle (kW)

\( \dot{E}_{cold} \) Exergy recovered by the cold stream (kW)

\( \dot{E}_{destroyed} \) Exergy destroyed inside a heat exchanger (kW)

\( \dot{E}_{hot} \) Exergy supplied by the hot stream (kW)

\( \dot{E}_p \) Exergy destruction of the power cycle (kW)

\( \dot{E}_s \) Exergy destruction of the combined cycle (kW)

\( h_1 \) Enthalpy at pump inlet (kJ/kg)

\( h_{10} \) Enthalpy at cooling condenser outlet (kJ/kg)

\( h_{11} \) Enthalpy at evaporator inlet (kJ/kg)

\( h_{2s} \) Enthalpy at pump outlet based on isentropic process (kJ/kg)

\( h_3 \) Enthalpy at boiler inlet (kJ/kg)

\( h_4 \) Enthalpy at expander inlet (kJ/kg)

\( h_{5s} \) Enthalpy at expander outlet based on isentropic process (kJ/kg)

\( h_6 \) Enthalpy at power condenser inlet (kJ/kg)

\( h_7 \) Enthalpy at power condenser outlet (kJ/kg)

\( h_8 \) Enthalpy at compressor inlet (kJ/kg)

\( h_9 \) Enthalpy at compressor outlet

\( h_{9s} \) Enthalpy at compressor outlet based on isentropic process (kJ/kg)

\( m_c \) Cooling cycle mass flow rate (kg/s)

\( m_p \) Power cycle mass flow rate (kg/s)

\( P_{pump} \) Pump outlet pressure (kPa)

\( \dot{Q} \) Heat transfer between the hot and cold streams (kW)

\( \dot{Q}_{boi} \) Boiler heat input (kW)
\( \dot{Q}_{eva} \) Evaporator cooling capacity (kW)

\( T_0 \) Temperature at dead state (room temperature) (°C)

\( T_{air,ave} \) Averaged cooling air temperature (°C)

\( T_{boil,out} \) Boiler outlet temperature (°C)

\( T_{c,ave} \) Averaged cooling stream temperature (°C)

\( T_{ccond,ave} \) Averaged condensing temperature in the cooling condenser (°C)

\( T_{h,ave} \) Averaged hot stream temperature (°C)

\( T_{oil,ave} \) Averaged heating oil temperature (°C)

\( T_{pcond,ave} \) Averaged condensing temperature in the power condenser (°C)

\( T_{room,ave} \) Averaged indoor room air temperature (°C)

\( T_{sub} \) Subcooling at power condenser outlet (°C)

\( W_{com} \) Compressor work input (kW)

\( W_{exp} \) Scroll expander work output (kW)

\( W_{net} \) Net work output in power cycle (kW)

\( W_{pump} \) Pump power consumption (kW)

\( W_{rev} \) Ideal reversible cycle work output (kW)

\( x \) Vapor quality

\( \eta_{c,II} \) Cooling cycle 2^{nd} law efficiency

\( \eta_{com} \) Compressor isentropic efficiency

\( \eta_{exp} \) Expander isentropic efficiency

\( \eta_{p} \) Power cycle conversion efficiency

\( \eta_{pump} \) Pump isentropic efficiency

\( \eta_{p,II} \) Power cycle 2^{nd} law efficiency

\( \epsilon_{recp} \) Recuperator heat transfer effectiveness
1. Introduction

Energy consumption continues to grow due to population increases and expanding economies around the world. There have been several consequences to these trends; two prominent ones are increasing energy costs and global warming. Unfortunately, while the demand for more energy continues to grow, its scarcity increases. In order to meet the future worldwide energy needs and slow the pace of global warming, the creation of sustainable energy sources and the improvement of energy efficiency have to be addressed simultaneously. In recent years renewable energy sources such as solar, wind, and geothermal have played increasingly important roles in satisfying energy needs. One method enjoying a resurgence of interest in harnessing thermal sources of renewable energy is the organic Rankine cycle (ORC), which can utilize low-to-moderate grade heat. Resources including geothermal, solar thermal, and waste heat fall into this category for the purposes of power generation [1-9]. ORC technology has proven to be economical and reliable for using thermal sources as low as 80°C [2, 3]. One interesting application that has gained attention recently involves using an organic Rankine cycle to produce fresh water. This approach converts solar energy into shaft power to drive a reverse osmosis desalination unit [10-14]. It has shown promise in southern Europe where seawater is present and optimal solar conditions exist.

In addition, there is great potential for reducing energy consumption by recovering low-grade waste heat that would be otherwise rejected to the surroundings. It has been estimated that industrial low-grade waste heat accounts for more than 50 percent of heat generated [1]. In general, heat is considered to be moderate-to-low grade if its temperature is less than 370 °C, which is relevant to steam power plants based on the Rankine cycle. Since a wide range of fluid choices exist [15-20], ORCs have the advantage of operating with good relative efficiency over a wide range of temperatures, for example, from 120 °C to 370 °C [21-25]. Therefore, the overall energy efficiency of a system can be significantly improved by incorporating an ORC into the process. Depending on the application, waste heat from a process could be used to generate useful energy such as shaft work, electricity, or cooling that can be used by another process. This reduces the energy consumption of the overall system.

This study investigates the use of low-grade waste heat to generate cooling. Essentially, an organic Rankine cycle is coupled to a vapor compression cycle to produce the cooling. Figure 1 shows the process and instrumentation diagram of the combined cycle. The approach taken is to utilize standard ORC with internal heat recuperation as the power source. In theory, The combined cycle is an alternative to the absorption cooling cycle with the advantage of providing shaft power, if needed, and is likely more efficient with high performance vapor expanders and compressors. A hot oil loop was used to simulate waste heat near 200 °C, in place of a process that produced waste heat. It vaporizes the ORC working fluid in the boiler and the superheated vapor is expanded to produce shaft work. Using HFC-245fa as a drying fluid upon expansion, the vapor still contains significant amount of sensible heat after expansion. The expanded vapor enters the power recuperator to preheat the fluid coming into the boiler. This internal heat recuperation is important for improving cycle efficiency as the required amount of heat at the boiler is reduced and the average heat input temperature also increased. The vapor then enters the power condenser to reject heat to the environment. The slightly subcooled fluid flows
through the reservoir and into the pump where the pressure is raised. The exiting high pressure fluid goes through the power recuperator and enters the boiler to complete the cycle. The power side working fluid, HFC-245fa, was selected as the power side working fluid as it has the desired thermodynamic properties for higher cycle efficiency. In addition, it has zero ozone depletion potential, relatively low global warming potential, and it is nontoxic, nonflammable, and noncorrosive.

The cooling side is a standard vapor compression cycle. Instead of using an electrical motor to drive the compressor, the compressor was directly coupled to the expander. The conversion losses associated with electrical motors are eliminated. The high pressure vapor after compression goes through the standard condensing and evaporating processes before completing the loop. The working fluid HFC-134a was used as the refrigerant. It is found widely in various mobile air-conditioning applications.

![P&ID Diagram of the Combined Power and Cooling Cycle](image)

Figure 1: P&ID Diagram of the Combined Power and Cooling Cycle

2. Combined Cycle Thermodynamic Analysis

From the thermodynamic stand point, the higher the heat input temperature and the lower the heat rejection temperature, the higher the thermodynamic performance of the system. In reality, however, there are always constraints that design engineers have to address with trade-offs having to be made. For a waste heat stream at any given temperature, it is desirable to recover as much waste energy as possible. This leads to lower temperature waste stream exhaust,
which would limit increasing the boiling temperature of the ORC working fluid due to the pinch point. In addition, the size of the system becomes an important design factor in portable energy systems; smaller systems require larger temperature differences to transfer the same amount of heat, and thereby degrade performance. The consequence could be higher entropy generation and exergy destruction, which is described later in the second law analysis. The prototype unit developed in this study was subjected to these design constraints.

2.1. 1st Law Cycle Performance

Based on the state points defined in the cycle diagram (Fig. 1), the power cycle conversion efficiency is defined as:

\[
\eta_p = \frac{W_{net}}{\dot{Q}_{boi}} \tag{2.1}
\]

\[
W_{net} = W_{exp} - W_{pump} \tag{2.2}
\]

\[
W_{exp} = \dot{m}_p (h_4 - h_{5a})\eta_{exp} \tag{2.3}
\]

\[
W_{pump} = \frac{\dot{m}_p(h_{3a} - h_3)}{\eta_{pump}} \tag{2.4}
\]

\[
\dot{Q}_{boi} = \dot{m}_p (h_4 - h_3) \tag{2.5}
\]

The cooling cycle COP is defined as:

\[
COP_c = \frac{\dot{Q}_{eva}}{\dot{W}_{com}} \tag{2.6}
\]

\[
\dot{W}_{com} = \frac{\dot{m}_c(h_{9a} - h_9)}{\eta_{com}} \tag{2.7}
\]

\[
\dot{W}_{exp} = \dot{W}_{com} \tag{2.8}
\]

\[
\dot{Q}_{eva} = \dot{m}_c (h_8 - h_{11}) \tag{2.9}
\]

The overall COP of combined cycle is:

\[
COP_z = \eta_p COP_c \tag{2.10}
\]

2.2. 2nd Law Cycle Performance

The 1st law performance of the cycle deals with energy conversion quantities of the cycle. It does not evaluate the quality of the energy conversion process. Second law analysis, however, answers this question. The 2nd law efficiency compares the efficiency of the current cycle to the efficiency of an internally reversible one such as Carnot cycle. It provides insight to identify the
sources of irreversibility, or the so-called exergy destruction, which can help improve the overall cycle efficiency. In addition, it provides a fair comparison of efficiency between different cycles operating at different conditions. The source of exergy destruction is attributable to entropy generation during a process such as heat transfer across finite temperature difference, and unconstrained expansion and compression. In addition, friction between surfaces and fluid pressure drop will also contribute to entropy generation inside each thermo-mechanical processes. Higher entropy generation for a particular process will lead to more exergy destruction, and thus lower 2nd law efficiency. By assuming each major component inside the combined cycle as a control volume and conducting the exergy balance around it, the exergy destruction rate based on steady state condition can be determined. After combining the exergy destruction rate in each component, the total exergy destruction rates of the cycle can be determined as follows:

\[
\dot{E}_p = \dot{m}_p T_0 \left( \frac{h_3 - h_4}{T_{oil,ave}} + \frac{h_6 - h_7}{T_{air,ave}} \right) \tag{2.11}
\]

Exergy destruction in cooling cycle:
\[
\dot{E}_c = \dot{m}_c T_0 \left( \frac{h_9 - h_{10}}{T_{air,ave}} + \frac{h_{11} - h_8}{T_{room,ave}} \right) \tag{2.12}
\]

Exergy destruction in combined cycle:
\[
\dot{E}_s = \dot{E}_p + \dot{E}_c \tag{2.13}
\]

According to Eqs. (2.11) and (2.12), the higher the average oil heating temperature and the lower the average outdoor cooling air temperature, the higher the exergy destruction in the power cycle. This is because the temperature differences inside the boiler and power condenser are higher for given fluid boiling and condensing temperatures, which generates more entropy during the boiling and condensing processes. Another important parameter is the enthalpy at the power condenser inlet. Its value reflects the degree of entropy generation in the expansion process. As indicated in Eq. (2.11), higher values of \( h_6 \) (enthalpy at power condenser inlet) correspond to lower isentropic expansion efficiency, and thereby higher exergy destruction in the power cycle. Similarly, higher indoor air temperatures (to be cooled) and lower outdoor air (cooling agent) temperatures would result in higher exergy destruction in the cooling cycle because of larger temperature differences in both the evaporator and cooling condenser. A compressor with lower isentropic efficiency would also increase the value of \( h_9 \) (enthalpy at the cooling condenser inlet), thus leading to higher cycle exergy destruction.

For any given heat exchanger, the exergy transfer can be described in the following:

Exergy supplied by the hot stream:
\[
\dot{E}_{\text{hot}} = \dot{Q}\left(1 - \frac{T_0}{T_{h,ave}}\right) \tag{2.14}
\]

Exergy recovered by the cold stream:
\[
\dot{E}_{\text{cold}} = \dot{Q}\left(1 - \frac{T_0}{T_{c,ave}}\right) \tag{2.15}
\]

Therefore the exergy destroyed inside the heat exchanger is:
\[
\dot{E}_{\text{destroyed}} = \dot{E}_{\text{hot}} - \dot{E}_{\text{cold}} = \dot{Q}\left[\left(1 - \frac{T_0}{T_{h,ave}}\right) - \left(1 - \frac{T_0}{T_{c,ave}}\right)\right] \tag{2.16}
\]
This indicates the higher the temperature difference between the hot and cold streams, the more exergy destruction occurs. In order to reduce the amount of exergy destroyed, it is desirable to keep the temperature difference to a minimum. Two approaches can be used to achieve this goal. One is to increase the heat transfer area, which will reduce the required temperature difference for a given amount of heat load. The other is to keep or even reduce the heat transfer area, but significantly increase the heat transfer coefficient. The first approach is achievable in stationary applications where size and weight are not an issue. It becomes impractical for portable applications where component size and weight significantly impact the overall design objectives. In this case, the alternate approach may be realized by using microchannel technology to enhance heat transfer. Several past studies [26-31] have shown the advantages of using microchannel heat exchangers, specifically for heating and cooling applications. They can achieve both goals of reducing exergy destruction, and minimizing size and weight. During this study microchannel heat transfer components were designed for all heat transfer processes, including the boiler, power recuperator, condensers and evaporator. In addition, scroll technology has been used to build the expander and compressor. Their inherent high isentropic efficiency at their intrinsic pressure ratio minimizes the entropy generation during the expansion and compression processes.

The 2\textsuperscript{nd} law efficiency of the cycle can be expressed in different ways. However, the more straightforward approach is to compare the actual cycle performance to the ideal (Carnot) cycle performance. Expressions of 2\textsuperscript{nd} law efficiency for both the power and cooling cycles are given as,

\begin{equation}
\eta_{p,II} = \left( \frac{\eta_p}{1 - \frac{T_{air,ave}}{T_{oil,ave}}} \right)
\end{equation}

\begin{equation}
\eta_{c,II} = COP_C \left( \frac{T_{air,ave}}{T_{room,ave}} - 1 \right)
\end{equation}

In addition, the 2\textsuperscript{nd} law efficiency of the power cycle can be determined by comparing the net work produced by the current power cycle with the reversible work produced by Carnot cycle. Thus:

\begin{equation}
\eta_{p,II} = \frac{W_{net}}{W_{rev}} = \left[ \left( h_4 - h_{5s} \right) \eta_{exp} \left( h_{2s} - h_1 \right) \eta_{\text{pump}} \right] \frac{m_p (h_4 - h_3)}{T_{air,ave}}
\end{equation}

\textbf{2.3. Thermodynamic Model}

A thermodynamic model was developed using EES (Engineering Equation Solver) to simulate the combined cycle performance at given operating conditions. Because the goal was to make the system as small and light as possible while maintaining relative high efficiency, a model of the system components was developed to estimate their size and weight and then use these values to determine the overall system dimensions and weight. Efficiency was not the primary concern because the combined cycle was designed to recover waste engine exhaust heat and
use it to generate cooling. A parametric study was conducted to investigate the cycle efficiency as functions of design parameter variations. The parameters investigated included expander efficiency, recuperator effectiveness, boiler superheat and condensing temperature. A number of working fluids were investigated for both the power and cooling cycle. Although thermodynamic performance was the primary concern, other characteristics such as safety and environmental impact were also taken into account. A single fluid system using HFC-245fa throughout both power and cooling cycles was compared with a dual fluid system where HFC-245fa was used in the power cycle and HFC-134a was used in the cooling cycle. Modeling indicated that both systems were of comparable conversion efficiency, although the single fluid system had practical advantages associated with using a single fluid and one condenser. The high boiling temperature of HFC-245fa in the cooling cycle was shown to increase the required size of cooling components such as the compressor and evaporator. As a result, the dual fluid system was selected for its advantages in giving smaller and lighter portable system.

The model simulating the dual fluid system has taken into account modest pressure drops across the heat transfer components and neglected pressure drops between components. The net power output did not take into account the parasitic fan power, although the pump power was included. The impact of different design parameters and component efficiencies on the power cycle efficiency and overall system COP are presented in Figs. 2 - 4. The rest of the parameters were held constant while varying one parameter. Unless it is the variable to be investigated, the default values for a few of the important parameters are listed in Table 1. The vapor after the boiler is kept saturated as the pump outlet pressure increased from 2,000 kPa to 3,250 kPa. As indicated in Fig. 2, both pump outlet pressure and superheat coming out of the boiler increased the power cycle efficiency and overall system COP. Both raised the heat input temperature for the power cycle leading to higher cycle conversion efficiency. In practice, however, this is not always controllable. In addition, for waste heat recovery applications, higher cycle conversion efficiency may not lead to higher overall waste heat recovery efficiency, as the waste heat stream exits with energy content remaining. Therefore, these efficiencies are parameters to be optimized for a particular application. Superheat has a positive effect on the cycle efficiency, because a high efficiency (85%) recuperator was incorporated into the cycle after the expansion.

Table 1: The Default Value of the Important Parameters

<table>
<thead>
<tr>
<th>P_{pump} (kPa)</th>
<th>T_{boil, out} (°C)</th>
<th>η_{exp}</th>
<th>ε_{recp}</th>
<th>T_{pcond, ave} (°C)</th>
<th>T_{sub} (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>2,750</td>
<td>190</td>
<td>75%</td>
<td>85%</td>
<td>67</td>
<td>12</td>
</tr>
</tbody>
</table>

According to Fig. 3, the expander efficiency has a very significant impact on the system performance. The overall system COP increases almost 50% when the isentropic efficiency changes from 60% to 85%. Although it is common to have a large-scale turbine running at very high isentropic efficiency, it is very difficult to achieve high efficiencies for turbine based expanders at the 1 - 10 kW level. Following the successful development of the first generation scroll expander, a reasonable isentropic efficiency of 75% was chosen for the model. The recuperator effectiveness also has very positive effects on the system performance because a
significant amount of sensible heat can be recovered from the expanded vapor. This is one of the benefits of using a drying fluid like HFC-245fa in the ORC. However, higher heat transfer effectiveness requires a larger recuperator and will increase system size and weight. As a tradeoff, the heat transfer effectiveness of the recuperator was designed at 85%.

Both the condensing temperature and fluid subcooling in the vapor compression cycle have significant impact on the system performance as shown in Fig. 4, especially the condensing temperature. This was due to the fact that the condensing temperature has an effect on both the power and cooling cycles. It determines the overall heat rejection temperature, which is another key parameter to improve cycle efficiency besides the heat input temperature; the lower the condensing temperature the higher the overall system COP. The condensing temperatures in the model were higher than standard commercial operating conditions. This is because the system was designed based on extreme conditions with outside air temperature reaching 48.9 °C (120 °F). In addition, a relatively large temperature difference from the condensing fluid to the atmosphere was used in order to reduce the size and weight of the condensers. The model has also demonstrated that more subcooling of the refrigerant coming out of the cooling condenser increases the cooling cycle COP and in effect increases the overall system COP. Additional subcooling keeps more liquid as the working fluid throttles through the thermostatic expansion valve (TXV in Fig. 1), which reduces the mass flow rate of the refrigerant for any given cooling load. Although subcooling is not needed in the ORC itself, 4°C of subcooling was built into the model in order to prevent pump cavitations during operation.

![Figure 2: Effect of Pump Pressure and Boiler Temperature on System Performance](image-url)
The effects of the major design parameters on 2nd law efficiencies are presented in Figs. 5 - 7. The power cycle 2nd law efficiency is shown on the left vertical axis, while the cooling cycle 2nd law efficiency is shown on the right vertical axis. The overall system COP is shown on the top horizontal axis.
law efficiency is shown on the right vertical axis. The graphs show the 2\textsuperscript{nd} law efficiencies in the cooling cycle are much lower than those in the power cycle. This can be attributed to the relatively larger temperature differences designed into the cooling cycle heat exchangers, i.e. the evaporator and condenser, in order to minimize component size and weight.

In the computational runs, the fluid exiting the boiler was maintained at saturated vapor conditions as the pump outlet pressure was varied. For investigating the superheat effects, the pump outlet pressure was set at 2,750 kPa. According to Fig. 5, pump outlet pressure has little effects on the 2\textsuperscript{nd} law cycle efficiencies, while the 2\textsuperscript{nd} law efficiencies decrease slightly with the fluid temperature (superheat) coming out of the boiler. This is because a fixed 20 °C of difference between the average heating source temperature and fluid temperature coming out of the boiler was given as an input. If the average temperature of the heating source is a fixed value instead, an increasing trend of the 2\textsuperscript{nd} law efficiency for the power cycle would be expected as the boiler outlet pressure/temperature increased. This is because entropy generation decreases as the temperature difference in the boiler decreases. To minimize the size and weight increase of the boiler as a result of a smaller driving temperature difference, a boiler based on microchannels with an enhanced heat transfer coefficient was designed.

The effects of expander isentropic efficiency and recuperator effectiveness are shown in Fig. 6. There is practically no change of 2\textsuperscript{nd} law efficiency in the cooling cycle. Apparently, expander isentropic efficiency has the most impact on the 2\textsuperscript{nd} law efficiency of the power cycle (as already described in a previous section), while increasing effectiveness of the recuperator also steadily increases the 2\textsuperscript{nd} law efficiency. This indicates that internal recuperation is beneficial when there is sensible heat available after expansion. With a drying fluid running at superheat, there is always a significant amount of sensible heat left after the expander. The recuperator essentially delivers a higher temperature liquid to the boiler, thus reducing entropic generation during the heat input process.

The effects of condensing temperature and subcooling on 2\textsuperscript{nd} law efficiencies are plotted in Fig. 7. Both the power and cooling cycle 2\textsuperscript{nd} law efficiencies decreased as the condensing temperature is increased (maintaining the surroundings temperature constant). This again results from the larger temperature difference between the condensing fluid and the atmosphere thus causing an increase in entropy generation. As more subcooling (lower T\textsubscript{sub}) is incorporated into the vapor compression cycle condenser, higher 2\textsuperscript{nd} law efficiency is obtained as a result of less entropy generated through the thermostatic expansion valve.
Figure 5: Effect of Pump Pressure and Boiler Temperature on 2\textsuperscript{nd} Law Efficiencies

Figure 6: Effect of Expander Efficiency and Recuperator Effectiveness on 2\textsuperscript{nd} Law Efficiencies
3. Prototype System Development

Due to the mobile nature of this application, portability was a design goal. This is why system weight and size were minimized. In an attempt to avoid compromising system performance, the heat transfer components were designed with microchannels and a high efficiency scroll expander was implemented. During tests conducted in the lab, a hot oil circulator was used to simulate the heat source. The combined cycle prototype unit (on the right) and the hot oil circulator (on the left) are shown in Fig. 8. The view is from the front side of the prototype unit, where the evaporator panel was situated. The louver panel in front of the brazed aluminum microchannel (BAM) evaporator protects the air fins and directs the flow. Two impellers were used behind the evaporator to draw air from the center of the front surface; and blow the cooled air out to the room through both the top and bottom slots. As also can be seen from the picture, there are 4 axial fans sitting on the top of the unit to provide air flow to cool the condensers. They are designed to provide a draw-through system, which means cooling air is pulled through the power and cooling condensers on each side of the unit and exits from the top. One of the primary concerns was hot air recirculation; the current design has the advantage of minimizing the amount of hot air pulled back into the system. A back view of the prototype unit is shown in Fig. 9. The entire volume is filled with components, plumbing, valves, instruments, controls, electrical, and data acquisition elements. Once the system is properly configured, the only external connections are a 110 VAC power cord, a USB cable connected to a laptop used for data acquisition and controls, and the white insulated lines connecting the boiler and the hot oil circulator.
Both condensers and the evaporator are based on BAM coils which are constructed of extruded aluminum tubes containing many small refrigerant ports. It greatly enhances the internal wetting surface, compared to conventional copper tubes for refrigerant flows. A BAM coil can deliver greater heat-exchange capacity than a conventional tube and fin coil in the same space. In other words, a BAM coil can be smaller than a tube and fin coil with the same capacity. In this way, BAM coils are significantly lighter than their traditional counterparts. In addition, the air side flow length of BAM coils is significantly shorter than that of their counterparts. This leads to advantage of using smaller fans and lower fan power requirement. Additional advantages of the BAM coils include their smaller internal volume,
which can reduce the refrigerant charge by up to 45%. Figure 10 shows a typical microchannel condenser based on BAM coils.

![Microchannel Condenser Based on BAM Coils](image)

**Figure 10: A Microchannel Condenser Based on BAM Coils**

Microchannel heat exchangers were designed for every heat transfer component in the prototype unit. Figure 11 shows the custom made integrated boiler and recuperator in the power cycle. The device has a rectangular cross section of 8.9 x 7.6 cm and is 33 cm long. The individual channel size inside the device is on the order of 200 µm, which significantly increases the heat transfer coefficients and reduces the overall heat exchanger size and weight. Unfortunately, after the diffusion bonding the post welding of headers caused numerous leaks in the device. A great amount of effort was spent to fix it, but the problem persisted. Two separate plate heat exchangers were used in its place so the performance tests could be carried out. An alternate design avoiding welds is being developed for later integration into the cycle.

![Integrated Microchannel Boiler and Recuperator](image)

**Figure 11: The Integrated Microchannel Boiler and Recuperator**
Two critical components of the power and cooling side were the scroll expander and scroll compressor, respectively. The expander is a positive displacement scroll compressor that has been modified to operate in reverse. The compressor is very similar to the expander, but operates in its designed direction. Specific scrolls were chosen for the expander and compressor based on the anticipated power levels, volumetric flow rates, and rotational speeds. Figure 12 shows a picture of the scroll expander. The diameter and height of the canister are 16.5 cm (6.5 inches) and 20.3 cm (8 inches), respectively. The expander and compressor are mechanically coupled with an inline, non-interfering, torque sensor in between. It measures the rotational speed, torque, and hence power of the expander. The expander output and compressor input shafts are on the side of the units. This is accomplished with two sets of spherical miters gears inside the canisters. The orientation of the unit is important for lubrication. An alternative scheme is to have a generating-motoring system where the two units would not be directly mechanically coupled. In this case, the expander would drive a motor-generator, and the compressor would run from electrical power input. However, this would reduce the overall system efficiency due to electrical conversion losses. Typical efficiency values for motors and generators are around 90%, hence a two-way conversion drops the efficiency immediately to 81% for just the power delivery to the compressor. In contrast a simple gear configuration has a mechanical efficiency between 95-97% [32]. For the expander, the high pressure and temperature vapor enters from the top center and exits from the side. In contrast, for the compressor, the low pressure vapor enters from the side and discharges from the top center.

Figure 12: The scroll expander with power take off (bottom left), inlet (top), and outlet (right)
A compliant scroll design was used for the expander in this project. A unique feature of this class of device is that the scroll plates contact each other to eliminate gaps between the orbiting and fix plates. A force on the orbiting plate must be applied at the correct magnitude in order to keep the plates from separating axially, but minimize excess frictional effects [33, 34]. Helping reduce the friction and seal the contact surfaces between the scroll plates is oil that is circulated with the working fluid. The orbiting scroll is off center and counterweights were implemented for dynamic balance. To support the rotational components ball bearings were used for their high operating speeds, small losses, and ability to handle combined axial-thrust loads. A shaft seal is required for transmitting the power through the canister wall. To minimize seal frictional losses the power take off shaft diameter is minimized at the shaft seal interface. The thermal efficiency of an ORC is highly dependent on the efficiency of the expander [35 - 37], thus it is important to minimize frictional losses and vapor leaks within the scroll wraps using lubrication and well designed mechanical components.

A small positive displacement rotary piston pump was used to pressurize the liquid in the power cycle ahead of the exhaust vapor recuperator. It was driven by a brushless DC motor where the pump speed could be adjusted through a motor controller. As the speed was increased for higher flow rates, the pump discharge pressure increased at the same time. A bypass line with a needle valve was included in the system design, as shown in Fig. 1, to allow the pump discharge to circulate back to the pump inlet. This achieved independent control of flow and pressure at the pump outlet. During testing, target pressures could not be achieved even with a very small orifice needle valve on the bypass line. This was primarily due to the light load placed on the expander by the compressor. A throttling valve was temporarily added at compressor discharge line to simulate additional load and raise the pressure to the design target. It provided the ability to control expander and compressor rotational speed and proved that desired pressure and flow could be achieved once the cooling side compressor load was in the target range. The throttling valve was removed for performance testing.

In addition, instrumentation as shown in Fig. 1 was integrated into the system to monitor performance and used to calculate thermodynamic properties such as enthalpy and entropy. Temperature and pressure were measured before and after each major component (instrumentation is described below). Flow rate was also measured for each respective cycle.

4. Data Reduction and Uncertainty Analysis

The collected pressure and temperature data was used in EES calculations, which use the fundamental equation of state [38, 39] to determine state properties such as enthalpy and entropy at each individual measurement point. These quantities were then used to calculate expander isentropic efficiency, cooling capacity, thermal efficiencies and coefficients of performance (COP) values.

All temperatures were measured with type K thermocouples made of special limits of error material, which have an uncertainty of ± 1.1 °C. All pressure transducers with inherent 0.25% accuracy were calibrated against a digital pressure gauge with an accuracy of ±0.05% (of full
scale range), or \( \pm 1.7 \) kPa. The mass flow rates for both the power and cooling sides were measured by turbine flow meters made by AW Company (model TRG-11.300-5) with an accuracy of \( \pm 1\% \) of the actual measured flow. The expander power output was measured by a Futek torque sensor (model TRS 605) with rated torque capacity of 10 N-m. It had an accuracy of \( \pm 0.3\% \) of the full scale range, or \( \pm 0.03 \) N-m.

The uncertainty analysis was focused on the bias errors introduced by the instruments, as the random errors during the experiments were minimal and averaged out. According to the theory of error propagation, the root-sum-square method such as proposed by Kline and McClintock [40] was used to combine individual errors. The resulting uncertainties for the quantities of interest are listed in Table 2 (Refer to Appendix for detailed equations).

<table>
<thead>
<tr>
<th>Quantity</th>
<th>( \eta_{\text{exp}} )</th>
<th>( \eta_p )</th>
<th>( \eta_{p,\text{II}} )</th>
<th>( \epsilon_{\text{recp}} )</th>
<th>( Q_{\text{evap}} )</th>
<th>COP(_c)</th>
<th>COP(_s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Uncertainty</td>
<td>2.5%</td>
<td>2.5%</td>
<td>2.5%</td>
<td>2.8%</td>
<td>2.2%</td>
<td>2.5%</td>
<td>3.5%</td>
</tr>
</tbody>
</table>

5. Performance Analysis and Discussions

A series laboratory tests were conducted to assess the prototype system performance over a range of conditions. Due to facility limitations, the system has not yet been tested under its design conditions with outdoor condenser air temperature of 48.9 °C and indoor evaporator air temperature of 32 °C. Rather, the system was tested indoors with air temperature at approximately 22 °C. Because of internal leaks associated with the custom made microchannel boiler and recuperator, two standard plate-type heat exchangers were used during the system tests.

Figures 13 and 14 show a typical pressure-enthalpy (P-h) diagram for the power and cooling cycle, respectively. The graphs provide in visual form the cycle conversion performance. As the boiling takes place from state point 3 to point 4 in Fig. 13, the corresponding distance on the horizontal axis (enthalpy) indicates the amount of heat input in the boiler. In contrast, the work generated by the expander is projected on the horizontal axis between state points 4 and 5. The ratio of these two is the power cycle conversion efficiency. For a given heat input to the boiler, as the length of the projection on the horizontal axis increases, the better the energy conversion efficiency. Also, from a second law point of view, less entropy is generated in the process as the slope between state points 4 and 5 decreases. Similarly in Fig. 14, the horizontal distance from state point 11 to point 8 represents the heat absorbed in the evaporator, while the horizontal distance from points 8 to 9 represents the work consumed by the compressor. The ratio of these two quantities gives the coefficient of performance for the cooling cycle.
Expander power output and recuperator effectiveness based on liquid side enthalpy change are plotted in Fig. 15. The power level increases almost linearly with the mass flow rate of the working fluid. As the pump speed was increased for higher flow rate, the pump outlet pressure also increased. The fluid saturation temperature inside the boiler increased and, as a result,
both expander inlet pressure and temperature increased. Although the expander power output was primarily due to an increase of the mass flow rate, the change of pressure and temperature at expander inlet had secondary and beneficial effects. In addition, as illustrated in Fig. 16, the isentropic efficiency of the expander changes with the expander inlet and outlet pressure ratio. It also plays a role in determining the actual power output. Unlike modeling which can specify the change of one parameter at a time, it is not unusual that changing one parameter in a physical system affects several others during testing. As indicated in Fig. 16, the expander isentropic efficiency approaches 84% at a pressure ratio around 3.3 and decreases gradually as the pressure ratio is increased.

The heat transfer effectiveness of the recuperator was evaluated based on the collected data at both the liquid and vapor sides. If the device was perfectly insulated and allowed to reach steady state, the heat transfer rates on both sides would be expected to be the same. The values are plotted in Fig. 17 with the horizontal axis showing effectiveness based on the liquid side and the vertical axis showing effectiveness based on the vapor side. According to the plot, effectiveness was consistently between 70 and 80 percent based on the enthalpy change on the liquid side, while effectiveness was on average 9% higher based on the vapor side. Although the recuperator was insulated with 0.5 inches thick melamine foam, it appears that the performance of the recuperator may be degraded by approximately 10% by heat loss during operation. This can be attributed to the significant amount of air flow inside the demo unit described in the previous section (draw-through air). As also shown in Fig. 15, the effectiveness of the recuperator increases with the mass flow rate, indicating heat loss became a smaller factor as fluid heat capacity increased. In other words, the thermal resistance associated with the heat loss became relatively larger as the mass flow rate of the power cycle increased.

The power cycle efficiencies at various pump pressures are plotted in Fig. 18. For most of the cases where pressure was between 1,700 kPa and 2,000 kPa, the 1st law efficiency was around 10% and 2nd law efficiency was around 30%. Lower pump pressure seems to affect both 1st and 2nd law efficiencies in a negative way, which confirms what was obtained in the thermodynamic analysis – the higher the pump pressure, the higher the fluid saturation temperature in the boiler. This not only raises the heat input temperature, which improves the 1st law efficiency, but also reduces the temperature difference in the boiler which reduces entropy generation. Because the pump outlet pressure affects other important system parameters, higher pump pressure could also cause lower cycle performance due to poor isentropic efficiency of the expander at high pressure ratios, as illustrated in Fig. 16. These parameters include the pressure ratio at the expander inlet and outlet, and fluid superheat coming out of the boiler.

Although the power cycle of the system performed relatively well during the tests, the performance of the cooling cycle was less than expected. Figure 19 shows the cooling capacity achieved, cooling cycle COP, and overall system COP. As illustrated, the cooling capacity increased with expander power as expected. However, both the cooling COP and overall system COP decreased as the expander power was increased. The direct cause for this was that the cooling capacity did not increase correspondingly with the power generated by the expander. This could be an inherent feature of the vapor compression cooling cycle; i.e. capacity is a function of several factors, not just power into the compressor alone. In general,
when more power is available to the compressor, more fluid can be pumped, resulting in more cooling capacity. However, this relies on whether the thermostatic expansion valve can provide the flow needed, so long as the rest of the parameters such as compressor efficiency and evaporating and condensing temperatures remaining unchanged. Both temperatures (essentially saturation pressures) can affect the amount of fluid available for cooling across the thermostatic expansion valve. For the data sets in Fig. 19, as expander power was increased, the difference between the evaporating and condensing pressures were changed slightly in favor of more cooling along with the higher expander and compressor rotational speed which also favored higher cooling capacity. The counteracting factors are lower isentropic efficiency for the compressor due to higher pressure ratio and lower refrigerant vapor density at the compressor inlet due to lower pressure and temperature.

According to the figure, the cooling capacities for most of the cases were between 3.5 and 4.5 kW, which are below the designed 5.3 kW cooling capacity. Although it is possible that the evaporator and/or compressor were undersized, the low cooling capacity currently observed is likely attributed to the fact that the cooling cycle was running considerably off its design point. The 5.3 kW design specification of cooling capacity was based on indoor and outdoor air temperature of 32 °C and 48.9 °C, respectively. This would provide enough driving force across the thermostatic expansion valve to achieve the needed mass flow rate. Because the unit was tested inside a lab with both the evaporating and condensing temperatures the same, there is not enough driving pressure difference to achieve the target flow rate which was believed to be the primary reason for the reduced cooling capacity at the conditions tested.

![Figure 15: Expander Power Output and Recuperator Effectiveness vs. Mass Flow Rate](image-url)
Figure 16: Expander Isentropic Efficiency vs. Pressure Ratio

Figure 17: Calculated Recuperator Heat Transfer Effectiveness
Although the actual prototype system was operating at different conditions from those in the system model, the test data provides valuable information on performance both at the system and component levels. The data supported the design target that the isentropic efficiency of the expander can attain 75 percent at a pressure ratio of 5. In addition, test results showed that the commercially available plate type recuperator was under performing. Compared to 85%
effectiveness in the model, the actual effectiveness of the commercially available plate recuperator was typically around 75% including the heat loss. A newly designed integrated boiler and recuperator based on microchannels is planned to replace the plate type boiler and recuperator when it becomes available. It is anticipated higher heat transfer effectiveness (85%) can be achieved with the microchannel devices as well as better insulation.

The power output from the expander was consistent with the mass flow rate values recorded during experiments, although both pressure and temperature were lower than the design specification during the tests. This was compensated by the lower condensing pressure and temperature in the power cycle. Essentially the power cycle conversion efficiencies (1\textsuperscript{st} law) at higher pump outlet pressures match the 1\textsuperscript{st} law efficiency in the system model reasonably well. A similar trend occurred in the 2\textsuperscript{nd} law efficiencies which were only slightly below the model results. Since the cooling cycle is sensitive to the operating conditions, running the cooling cycle off its design point has led to reduced cooling capacity.

6. Conclusions

The concept of combining an organic Rankine cycle (ORC) with a vapor compression cycle for heat activated cooling was demonstrated in this study. A small scale prototype system based on this concept was developed with guidance from a system model. Highly efficient microchannel heat exchangers along with the use of a scroll expander and compressor were shown to be effective in achieving high system performance at reduced component size and weight. The measured isentropic efficiencies of the scroll expander are especially valuable at its given scale with values ranging from 70% to 84% depending on the imposed pressure ratios. The system performance based on tests conducted in the lab environment shows promise, although the original cooling capacity specification was not attained due to the system operating off the design point. However, this study has shown the combined cycle is viable and can potentially become part of an overall energy solution for heat activated cooling. By recovering waste heat from diesel engines and other power cycles, the system can generate cooling as well as power to improve the overall efficiency and the utilization of fuel. Based on an overall system COP of 0.5, the combined cycle can convert half the amount of waste heat into cooling, which can be significant in many applications. Higher cooling capacity is expected for the system when tested under design conditions and when the full suite of microchannel heat transfer components are integrated into the power cycle. This will further increase the overall COP of the system.

Acknowledgement

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Appendix

Below are listed equations used for uncertainty analysis. It is based on the theory of error propagation and use Root-Sum-Square method to combine the errors.

\[
\frac{u_{\text{COP}}}{\text{COP}} = \sqrt{\left(\frac{u_{\eta_p}}{\eta_p}\right)^2 + \left(\frac{u_{\text{COP}}}{\text{COP}}\right)^2}
\]

\[
\frac{u_{\eta_p}}{\eta_p} = \sqrt{\left(\frac{u_{\dot{w}_{\text{net}}}}{\dot{W}_{\text{net}}}\right)^2 + \left(\frac{u_{\dot{Q}_{\text{boi}}}}{\dot{Q}_{\text{boi}}}\right)^2}
\]

\[
\frac{u_{\text{COP}}}{\text{COP}} = \sqrt{\left(\frac{u_{\dot{w}_{\text{com}}}}{\dot{W}_{\text{com}}}\right)^2 + \left(\frac{u_{\dot{Q}_{\text{evap}}}}{\dot{Q}_{\text{evap}}}\right)^2}
\]

\[
\frac{u_{\dot{Q}_{\text{boi}}}}{\dot{Q}_{\text{boi}}} = \sqrt{\left(\frac{u_{m_p}}{m_p}\right)^2 + \left(\frac{u_{h_4}}{h_4}\right)^2 + \left(\frac{u_{h_3}}{h_3}\right)^2}
\]

\[
\frac{u_{\dot{Q}_{\text{evap}}}}{\dot{Q}_{\text{evap}}} = \sqrt{\left(\frac{u_{m_c}}{m_c}\right)^2 + \left(\frac{u_{h_8}}{h_8}\right)^2 + \left(\frac{u_{h_1}}{h_1}\right)^2}
\]

\[
u_h = \sqrt{(h(T + u_T, p) - h(T, p))^2 + (h(T, p + u_p) - h(T, p))^2}
\]

Reference


[40] Figliola, R., and Beasley, D., Theory and Design for Mechanical Measurements, 1995