DESIGN AND BUILD OF A 1 KILOWATT ORGANIC RANKINE CYCLE POWER GENERATOR

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ABSTRACT

Organic Rankine Cycle (ORC) systems are capable of utilising low-enthalpy geothermal sources. The aim of the Above Ground Geothermal and Allied Technologies (AGGAT) research programme is the development of ORC systems within New Zealand. An experimental scale ORC system, known as ORC-B, was built at the University of Canterbury to assist with the research and development of the system design and component selection process.

The unit is a 1 kW ORC consisting of four key components: evaporator, expander, condenser and pump. Selection of the working fluid was found to be a constraining factor in the design. A refrigerant mix known as HFC-M1 was selected due to its desirable performance, high safety and ease of availability in New Zealand. The heat source for the system is the exhaust of a Capstone gas turbine. Heat is transferred to the unit's plate-type evaporator by a thermal oil extraction loop. Cooling is supplied to the plate-type condenser from the available water supply. A scroll expander is used to extract work from the system and the evaporator is fed by a high-pressure plunger pump. The instrumentation selected allows measurement of the performance of the system as well as PID control. The system has been designed and constructed and is now ready for commissioning and testing.

1.1 ORC Background

ORC systems are typically used for four major applications: waste heat recovery, geothermal power plants, biomass combustion plants and solar thermal plants. The primary difference between an ORC and a traditional steam Rankine cycle is the use of an organic working fluid, such as ammonia, pentane or a halocarbon. The application of organic working fluids allows the extraction of energy from a low temperature resource at a higher efficiency than conventional steam cycle technology (Macián, Serrano, Dolz, & Sánchez, 2013). Despite ORC systems and steam Rankine cycles being conceptually very similar, the use of an organic working fluid requires a higher level of caution to avoid leakage or contamination as the consequences are more severe.

There are several ORC binary cycle plants currently installed in New Zealand utilizing geothermal resources. All these systems so far have been designed by overseas companies because the necessary expertise was not available in New Zealand.

1.2 University of Canterbury Background

The University of Canterbury recently began aiding HERA and AGGAT in developing the research necessary for designing ORC systems.

Together the three groups aim to develop the expertise required to design and fabricate ORC systems within New Zealand, making the technology more affordable and accessible.

The University built a small scale ORC test-bed in 2012 named ORC-A. This unit was used to prove the feasibility of the system concept and to identify design requirements necessary for further development. This system successfully produced 400 W of electricity and identified many important requirements for ORC design and control.

Figure 1 - Initial ORC-A system for testing the system concept

The University and AGGAT are now developing a second test bed to further improve upon the knowledge gained from the first unit. The ORC-B will be used to test different working fluids, turbines and control systems to gain greater capability for developing ORC systems.

2. ORC DESIGN PROCESS

The AGGAT guideline, explaining the design process for an ORC plant, is shown in Figure 2. This design process is primarily for use with full scale plants as it ensures the plant will be economically successful.

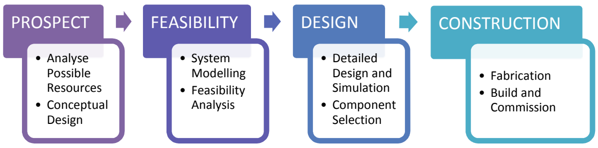

Figure 2 - ORC design guideline

This design guideline was used where applicable throughout the design of the ORC-A system. The feasibility analysis completed was brief for this system as it was pre-determined that a test bed would be designed.

All of the steps are linked as this is a highly iterative process: any changes require reconsideration of the previous steps.

3. PROSPECTING

The preliminary phase of ORC design requires the prospective heat resource to be assessed for use with an ORC system. This involves visiting the site and collecting any important information regarding the potential resource.

It is important to fully understand the resource before commencing the design of the ORC as typical waste heat and geothermal resources have limiting factors or special requirements, such as reinjection or minimizing effect on the resource. The details of water or air cooling also need to be considered as these will affect the size and efficiency of the plant.

3.1 Capstone Gas Turbine

The selected heat resource for the ORC-B is the waste heat produced by the exhaust of a 30 kW Capstone Diesel Turbine. This heat resource is easily assessed as the waste heat specifications are well documented. There are minimal special requirements for this heat resource, as the heat would otherwise be vented to atmosphere.

Table 1 - Capstone turbine waste heat resource parameters

Parameter	Specification
Resource Type	Hot exhaust gases. Clean gas.
Temperature	220°C
Mass Flow	0.3 kg/s
Total Waste Heat	90 kW
Limiting Factors	1. Exit temperature must remain greater than 100°C for the gas to leave the exhaust flue. 2. Heat exchanger back pressure must be less than 1kPa to prevent reduction in the Capstone efficiency. 3. System safety is paramount as the location is accessible by students.
Extractable Resource	30 kW of thermal energy.

The extractable heat resource is limited in order to meet the requirements outlined in Table 1. A limit of 30 kW is set to ensure the extracting heat exchanger causes minimal pressure drop and is compact. The limit also prevents the exhaust temperature dropping below 100°C. Preliminary calculations show that a fin and tube heat exchanger will be capable of extracting up to 30 kW without exceeding the tolerable pressure drop.

3.2 Cooling Resource

The primary cooling options available on-site were: air cooling, water cooling or air cooling using a water tower. Due to the small scale of this system it was determined that water cooling would be the easiest option. The estimated water wastage of 100 tons/year was deemed to be acceptable. A cooling tower would be recommended for a larger system to reduce wastage.

Table2 - Water cooling parameters

Parameter	Value
Water Temperature	15°C
Water Flow Rate	0.5 kg/s

3.3 Conceptual Design

The data from the initial prospecting allows the conceptual design of an ORC system that will meet the required specifications. An ORC system coupled with a thermal oil extraction loop is proposed. The thermal oil extraction loop allows greater controllability of the system and allows the ORC system to be located further from the Capstone turbine.

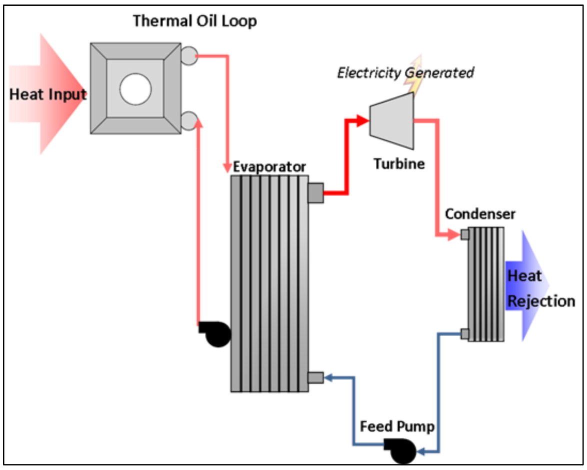

Figure 3 - Conceptual design of the ORC unit

3.3.1 Working Fluid

To further develop the conceptual design of the ORC system the potential working fluids need to be determined. The working fluid is a limiting factor in ORC design as it affects the thermodynamic design and performance of all components within the system. The refrigerant also influences the required pressure rating and material type of all components in the system.

Common refrigerants used for ORC systems were analysed and are summarized in Table 3.

Table 3 - Refrigerant Selection

Working Fluid	System Efficiency	Advantages	Disadvantages
R-134a	5.0 %	Readily available, Handling safety	Poor performance, High pressure
R-245fa	5.7%	Good performance	Not readily available in New Zealand
R-365mfc	5.5%	Handling safety	Not readily available in New Zealand
HFC-M1 Blend (50/50)	5.7%	Good performance, Available in New Zealand	Thermodynamic properties are not well documented
n-Pentane	5.9%	Excellent performance	Flammability risk
R-141b	6.0%	Excellent performance	Ozone depleting

The ORC-A system used R-134a as the working fluid as it is readily available, low cost and widely used in the refrigeration industry. The thermodynamic performance of R-134a at temperatures exceeding 50°C was found to be incredibly poor. Higher operating pressures also decreased the safety factor of experimental components. For these reasons it was decided that a more efficient refrigerant should be chosen.

A refrigerant mix known as HFC-M1 was selected for use with the ORC test bed. HFC-M1 is a mix of 50% R-245fa and 50% R-365mfc. This mix of refrigerants is more readily available in New Zealand than its virgin constituents as it is used as a blowing agent by the polymer industry. A boiling temperature of 30°C at atmospheric conditions also increases the ease of handling of this refrigerant.

The refrigerant offers several benefits over water, such as the lower boiling temperature and dry saturation curve. This is beneficial to the system performance as it means the vapour flowing through the turbine is not a wet mixture.

3.3.2 Heat Extraction Loop

A thermal oil extraction loop will be used to transport the thermal energy between the Capstone exhaust and the ORC system evaporator. This provides many benefits to the system design, such as: preventing degradation of the refrigerant due to high film temperatures, additional safety as the oil is non-flammable, as well as making the system easier to control in a laboratory environment. It also allows the ORC system to be designed around a liquid heat source which is more relevant for geothermal applications.

The addition of a thermal extraction loop adds extra complexity to the cycle; an additional heat exchanger is required as well as an oil pump, which reduces the overall electrical output of the system. This is not necessary for future geothermal plants.

The thermal oil selected from Petro-Canada is known as Calflo HTF. The oil was selected as it is readily available, operates at the required temperature range and has a reasonably low viscosity. A reduced viscosity is desirable as it minimizes the required pump work and increases the heat transfer.

3.3.3 Thermodynamic Modeling

A thermodynamic model of the conceptual design shown in Figure 6 was developed in Engineering Equation Solver (EES) and validated against experimental data from Quoilin (Quoilin, Lemort, & Lebrun, 2010). The model was then adapted to use both the available resource temperatures and expected performance characteristics of the system components. Estimates could then be made for the duty of each of the four major components within the system. The thermodynamic cycle simulated by this model is shown in Figure 4.

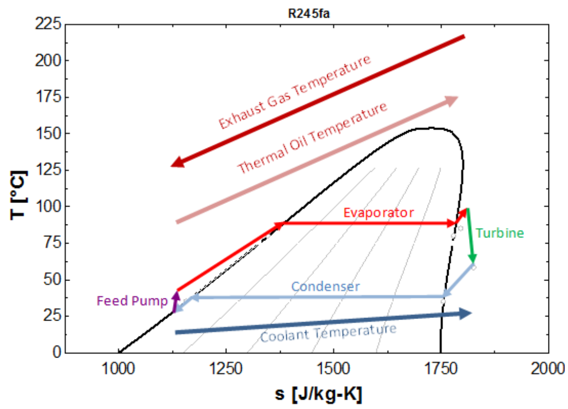

Figure 4 - Thermodynamic T-s diagram of conceptual design

The duty of each component and proposed operating conditions are shown in Figure 5.

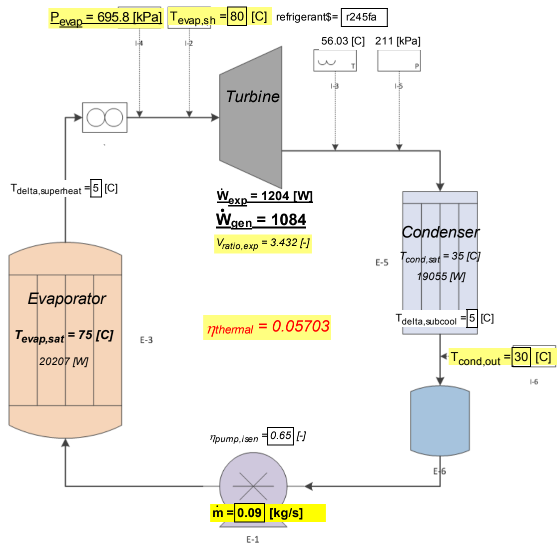

Figure 5 - Steady state model of conceptual design

3.2.3 System Efficiency

The system efficiency for an ORC system is a key performance parameter and is defined as:

$$\eta_t = \frac{\dot{W}_{gen} - \dot{W}_{input}}{Q_{in}} \quad (1)$$

The thermal efficiency estimated for this design is 5.7%. The efficiency of an ORC system is typically very low due to the low resource temperatures which they utilize. The

Carnot efficiency of a cycle is the theoretical maximum efficiency for a thermodynamic cycle.

$$\eta_t = \frac{T_h - T_c}{T_h} \qquad (2)$$

For the operating conditions proposed the cycle Carnot efficiency is 10.1%, meaning that a cycle efficiency of 5.7% is can be considered satisfactory for testing purposes. Thermodynamic efficiency is often used as a benchmark for ORC system design, but is not the only consideration. Maintenance, ease of manufacture, capital expense and environmental impact all need to be considered when optimizing the system design. These considerations were balanced with thermal efficiency throughout the design process and resulted in the design efficiency of 5.7%.

3. FEASIBILITY ANALYSIS

Once the conceptual design is completed a feasibility analysis is necessary to evaluate whether it is worthwhile continuing the project. This analysis should consider the impact of installing an ORC system on the resource it is utilizing, as well as if it is financially viable or profitable to do so.

The ORC-B system is required for test purposes only and does not need to be a profitable venture. The feasibility study considered: available space, cost considerations, availability of components and applicability for research into small ORC test beds. These considerations were combined with the knowledge gained from the ORC-A test bed and it was determined that this project would be feasible.

4. DETAILED DESIGN

The detailed design considers the conceptual design in greater detail and investigates how each component fits into the system. The design process requires numerous iterations as each component effects the overall behavior of the system, and therefore the operating conditions of the other components.

In a large ORC system components would be either fabricated or sized under the assumption that such components are available. With the small scale of this unit it is more practical to research available components that can be used outside their nominal design conditions or can be modified to meet the requirements. This meant that component research was carried out throughout the design process

4.1 Turbine

The system turbine is typically the most costly component in an ORC system as it needs to be precision engineered. The turbine is the most vital element in an ORC system as it allows fluid energy to be extracted via an expansion process. The energy is converted to mechanical shaft energy which is finally converted into electricity by an electrical generator. The turbine needs to be selected carefully to maintain optimum system efficiency and should be specifically designed for the available resource when possible.

The ORC-B only requires a small 1 kW turbine. Turbines that meet the requirements of our proposed system design are not commonly manufactured. As an interim measure a turbo expander was purchased from Air-Squared in the USA. Meanwhile a customized radial turbine is being developed at the University to study the possibility of modifying a turbocharger turbine for use with an ORC system.

4.1.1 Scroll Expander

Positive displacement turbo-machines, such as scroll expanders, are suitable choices for small-scale ORC systems. They are readily available for 1-30 kW applications, with minimal control systems and high volumetric expansion ratios.

A scroll expander uses the expansion of a vapour pocket through two concentric scrolls to produce shaft rotation, as shown in Figure 6. It has a fixed volumetric ratio with two involutes curves orientated in different directions and 180° out of phase. One scroll is fixed while the other scroll orbits. High pressure fluid enters the suction port in Figure 6.1 and expands steadily in the sequence shown in Figure 6.2 to Figure 6.5. The expansion ends when the fluid is discharged at low pressure and temperature as shown in Figure 6.6.

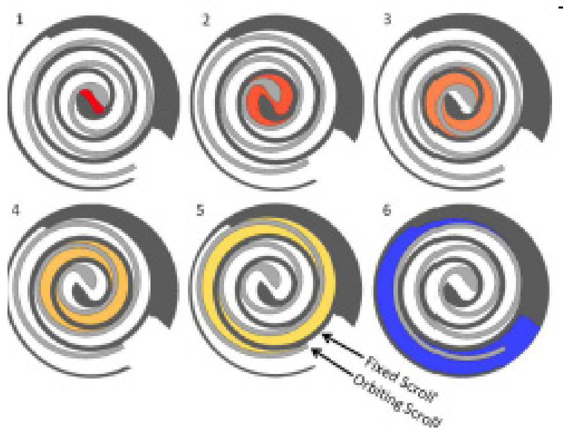

Figure 6 - Expansion through a scroll expander (Oralli, 2010)

Scroll expanders serve adequately but with a trade-off between convenience and efficiency as the efficiency is lower than the corresponding compressor efficiency (Orosz, Mueller, Quoilin, & Hemond, 2009).

A scroll expander was purchased from Air Squared which is custom made for small-scale ORC applications. The expander selected ensures better performance than would be expected from an experimental turbine design and is a readily available solution to facilitate experimental analysis of the heat exchangers and system behavior.

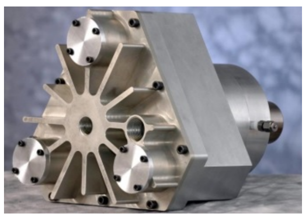

Figure 7 - Air Squared expander

Table 4 - Air Squared expander specifications

Parameters	Value
Expansion Ratio	3.5
Max Rotation Speed	3600 RPM
Max Pressure	13.8 bar
Displacement	12 cm^3/rev
Generator	Magnetic Coupling 240 V, 50 Hz

4.1.2 Radial Turbine

The University of Canterbury is developing a small radial turbine for use with the ORC-B test bed. Turbines are the most difficult and expensive component to acquire in large ORC systems, and so a method has been developed which allows the turbine wheel of an automotive turbocharger to be used as a radial turbine. The pressure ratio of an automotive turbocharger is considerably less than that experienced during operation of an ORC. Despite this the design is similar and the cost significantly less than other radial turbine technology. This approach will make turbines for small scale ORC systems more affordable.

The modification of an automotive turbocharger involves the selection of a turbocharger, evaluation of turbine wheel performance, casing design, development of a lubrication system, and selection of all auxiliary components such as bearings and seals. The proposed design of the ORC turbine, shown in Figure 8, was developed by retrofitting a small turbocharger from a petrol driven vehicle.

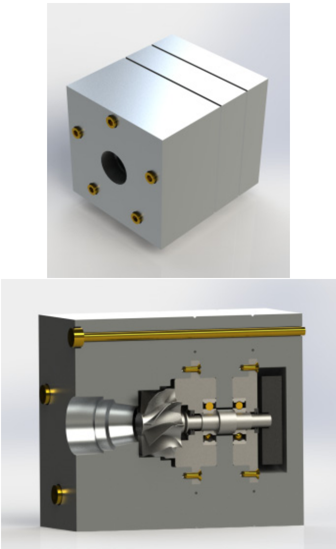

Figure 8 – Radial turbine using an automotive turbocharger

4.2 Heat Exchangers

Two heat exchangers are required within the ORC-B system: an evaporator and condenser. Shell and tube heat exchangers are typically used for full size ORC systems. A scaled down version of these shell and tube heat exchangers is not possible due to manufacturing limitations. Instead plate heat exchangers will be used for both the evaporator and condenser. Plate heat exchangers allow a compact solution for liquid to liquid heat transfer.

4.2.1 Thermodynamic Calculation

The heat exchangers were sized using thermodynamic calculations for heat transfer. These calculations used the NTU-effectiveness method where:

$$NTU = \frac{UA}{\dot{C}_{min}} \qquad (3)$$

(see Shah, R. K. &Sekulić, D. P. Fundamentals of heat exchanger design Wiley Online Library, 2007).

The Number of Thermal Units (NTU) calculated from Equation 3 allows the heat exchanger effectiveness to be estimated for the experimental counter-flow heat exchanger data from Shah (Shah, Sekulic, 2007) as shown in Figure 9. This method allowed the correct selection of the heat exchanger to match the duty from the thermodynamic model.

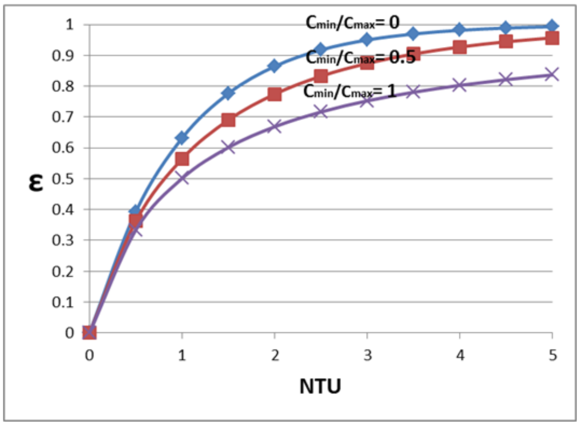

Figure 9 - Effectiveness vs NTU for a counter-flow heat exchanger

$$\dot{Q} = \epsilon \dot{C}_{min}(T_{1_{in}} - T_{2_{in}}) \qquad (4)$$

The process was repeated iteratively to allow the correct model of heat exchanger to be selected that gives the desired transfer of heat. The Kaori brazed plate heat exchangers selected are designed for use as refrigeration evaporators and condensers to allow phase change. They were found to offer a greater selection than competing products and can operate at pressures of up to 45 bar.

Figure 10 - Kaori K205 evaporator

Table 5 - Plate heat exchanger specifications

	Evaporator K205-60	Condenser K095-40
Number of Plates	60	40
Heat Transfer	20.0 kW	19.1 kW
Operating Pressure	7.0 bar	2.1 bar
Channel 1	Evaporating R245fa Inlet 30°C, Outlet 80°C	Condensing R245fa Inlet 30°C, Outlet 80°C
Channel 2	Purity FG Oil Inlet 110°C, Outlet 90°C	Water Inlet 15°C, Outlet 22°C

4.3 Feed Pump

The ORC feed pump is used to provide the evaporator with a constant supply of liquid. Pump selection charts were used to determine the type of pump required. Typically a centrifugal pump would be used in a large scale ORC as the flow and pressure requirements could easily be matched. A small scale ORC requires a much lower flow rate with the same pressure.

4.3.1 Plunger Pump

A reciprocating positive displacement plunger pump was selected to meet the pressure and flow requirements. A small reciprocating pump can meet the pressure requirements of an ORC system while maintaining low flow rates

Table 6 - Feedpump specifications

Parameter	Specification
Pump Model	Cat 2SF22ELS
Displacement	4.8 cc/rev
Duty Flow Rate	5.1 LPM
Max Flow	8 LPM
Duty Pressure	480 kPa
Max Pressure	20.0 MPa

As this is a positive displacement machine the pump will meet the systems pressure requirements at the selected flow rate. The flow rate can also be calculated as a function of the motor RPM. Coupling the pump to a Variable Speed Drive (VSD) allows the pump flow rate to be set via PID control.

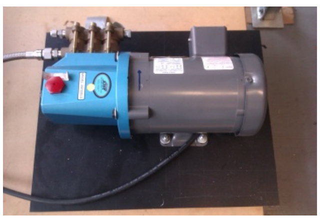

Figure 11 - Cat plunger pump coupled to VSD

4.3.2 Cavitation

Cavitation is a large issue for pumps in ORC systems, as the working fluid leaving the condenser is only slightly sub-cooled and evaporates if the pump suction is too high. Auxiliary sub-coolers can be used to reduce the occurrence of cavitation but compromise the system performance. The pump in the ORC-B has been placed 1.2 m below the condenser to increase the static head at the inlet of the pump and reduce the likelihood of cavitation.

4.4 Buffer Tank

A custom manufactured 1.8 L buffer tank is located before the pump to allow the system to behave dynamically during startup and while testing different operating conditions. The tank also ensures that there is always liquid being provided to the pump as vapour can damage the seals in the pump. A level switch deactivates the pump if vapour is detected within the tank.

Figure 12 - Buffer tank

4.5 Piping and Instrumentation

Once all the components have been selected the piping and instrumentation diagram in Figure 13 was designed to allow instrumentation and control components to be selected.

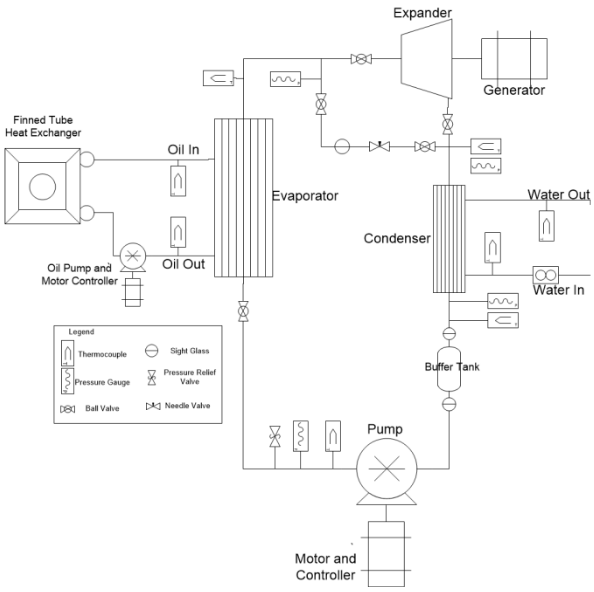

Figure 13 - ORC-B piping and instrumentation diagram

4.5.1 Bypass Line

A bypass line was installed around the expander. This will be used during startup as the installation of a needle valve allows a pressure differential to be reached before the expander is started.

The bypass line is also used as a safety feature in case the expander jams or an emergency shutdown is required.

4.5.2 Piping

Two types of piping are used within the system: stainless steel tube and PTFE flexible tube. The rigid stainless steel is used between the heat exchangers and expander, as these components are fixed. Flexible tubing was installed at the inlet and outlet of the pump to allow the static head at the inlet of the pump to be varied.

Piping losses were considered and appropriate pipe diameter selected to ensure the losses were negligible.

4.5.3 Instrumentation

Temperature, pressure and flow rate data is collected throughout the system to allow the user to operate the system and reach a steady state at the desired operating conditions. A CompacDAQ is used for data acquisition and allows the system performance to be evaluated.

Currently the system will need to be controlled by the user, but as the system dynamics are understood better PID control will be implemented.

5. CONSTRUCTION

Construction of the system was completed throughout the year and the system is now ready for testing. The majority of the components selected were readily available which reduced the fabrication time frame. Extra care was taken throughout the construction and assembly processes to ensure there were no leaks in the system, thus preventing loss of refrigerant.

Figure 14 - ORC-B ready for testing

6. SUMMARY

This paper presents the design process required for the development of a small scale ORC test bed as well as the required component selection considerations. All system components were selected, fabricated and assembled ready for the system to be charged and tested.

7. FUTURE WORK

The immediate future work on this system is to charge the system with the HFC-M1 refrigerant. Once this is complete the system will be extensively tested to allow research into the system behavior. The performance of the unit will then be analyzed and any necessary changes implemented. A custom made radial turbine will be tested within the system. Once all required testing is complete the system will be used to optimize control methods and test different working fluids at a small, manageable scale. This will aid system design as it will allow refrigerant performance to be tested and control systems to be developed to control the dynamic behavior of the ORC-B system.

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