

Low Temperature Difference Electricity Generation Using ORC

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Many geothermal sites in Australia have low temperature water as an available heat source. Often this temperature is under 75°C, in conditions where ambient temperatures can climb into the mid 30's on a regular basis. These heat sources have often been considered to have an insufficient temperature difference, with respect to the atmospheric heat sink, to be worthwhile for driving a heat engine. As the temperature difference reduces, the maximum achievable efficiency must also reduce as demonstrated by classical thermodynamics. Thus, as the temperature difference drops, it is a matter of improving component efficiencies while reducing costs to make such systems viable. Lower temperature systems frequently use an Organic Rankine Cycle (ORC) heat engine. As the temperatures drop, the key parameters in the design of such systems become cost and efficiency. Of prime importance is the cost of the expander, efficiency of the heat exchangers, and the precision of the control system. The main investigations presented are:

- The reduction in system cost through selection of alternatives to turbine expanders.
- The selection of an optimised working fluid for low temperature operation.

Keywords: Organic Rankine Cycle Heat Engine, low grade heat, low temperature difference electricity generation, hydrothermal energy, low temperature geothermal.

Low Grade Heat Sources in Australia

Low grade heat (LGH) sources, here defined as below 100°C, are available throughout Australia. The utilisation of renewable energy sources (such as geothermal and solar thermal power) has been a topic in vogue recently and these sources lend themselves cost effectively to providing LGH that can then be transformed into other forms of energy. Low grade waste heat sourced from industrial processes can also be converted into electricity and therefore improve the efficiency of these industrial processes and consequently advance the pursuit of cost reduction and sustainability in industry. LGH is widely available from these industrial sources and includes the heat energy not utilised in absorption chillers, process condensate and cooling water from a range of industrial processes.

Australian hydrothermal resources

Australia has hydrothermal resources at a number of locations throughout the continent. Notable areas where hydrothermal energy is easily

accessible include the Otway and Gippsland Basins (King et al 1985; Sinclair Knight Mertz 2005), the Perth Basin (Ghori 2008) as well as the Great Artesian Basin (Burns et al. 1995; Pirlo 2004). Direct heating applications have been applied to the heated waters of the Gippsland and Otway Basins (Burns et al. 2000) and the Great Artesian Basin has been used to produce electricity on a small to moderate scale with the Mulka Station (now closed) and Birdsville ORC installations (Sawyer 1991; Burns et al. 1995; Burns et al. 2000). The Great Artesian Basin geothermal resource, used as the heat source for the Birdsville plant, has the highest temperature of the resources described. The power station utilises a 1230m deep bore delivering 27l/s of 98°C water to provide heat energy to the plant (EPA Qld 2005). The other three hydrothermal resources mentioned previously in this section have considerably lower temperatures associated with them. The reason for exploiting these low temperature hydrothermal resources is to capitalise on the relative ease in accessing them and thus save on drilling costs.

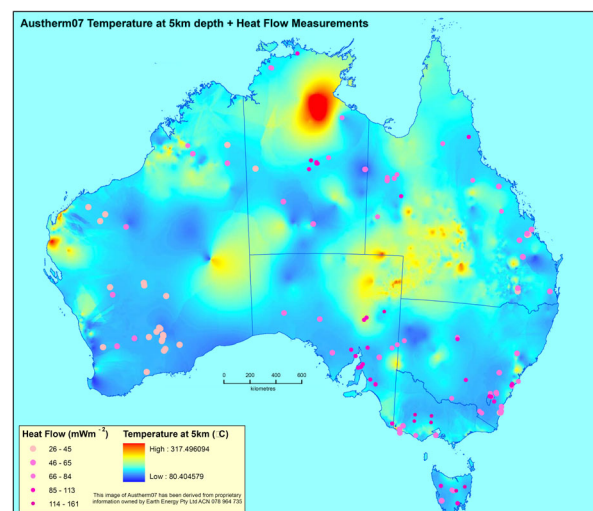


Figure 1: Map of projected temperatures at 5km depth and heat flows for Australia. The areas of darkest blue indicate a temperature of 80°C at 5km depth. The image of Austherm07 from proprietary information owned by Earth Energy Pty Ltd (sourced from Geoscience Australia 2007).

Figure 1 is a diagram that many people are familiar with and is a projection of the temperatures at a depth of 5km. It can be observed from the figure that for most of the continent, the projected temperature at that depth

is around 80°C. It can also be seen that these low temperature areas cover the national population centres and the higher projected temperatures are a considerable distance from the major cities.

In 1999 Kanoglu and Cengel calculated that the cost of drilling to a depth of 5km could cost 2 ½ times the cost of the power plant itself. Ruge (1999) contended that to drill 3.5 km into the Dilwyn Aquifer in the vicinity of Portland Victoria would cost A\$1.98 million. For current projects, drilling to the depths to utilise 'hot dry rock' energy in Australia would be between A\$15 million and A\$37 million according to Goldstein et al (2008). This costing is backed up by a report prepared for the Australian Geothermal Energy Association by the consultants McLennan Magasanik Associates Pty Ltd (2008). One proponent of the study conducted by McLennan Magasanik Associates Pty Ltd (2008) suggested 'the cost of drilling to 4,500 metres was four times the cost of drilling to 3,000 metres'. One way to avoid the high costs of deep drilling is to avoid going to such depths as is required with Australia's geology to source high grade heat. Instead, it may be more cost effective to instead utilise lower grade heat.

Heat Engines for Electricity Generation from Low Grade Heat Sources

Currently, the lowest cost method of converting high grade heat into electricity is through the use of a heat engine. This statement holds true for lower temperature heat sources as well, but as heat source temperatures drop, the units have to be scaled up to achieve the same output. This is due to the compounding effects of 1) the amount of energy contained in lower temperature streams being less and 2) the efficiency of conversion being lower. The result is increased capital costs per unit output. The way to improve the utilisation of lower temperature heat sources, and thereby make use of underexploited resources or reclaim wasted energy, is to reduce the costs of employing heat engines for converting lower grade heat into a higher form of energy such as electricity. Investigations have been conducted into the cost drivers for ORCHEs by those such as Schuster et al. (2009) and Leibowitz et al. (2006), albeit for source temperatures around 100°C and above. These studies showed the possibility of designing commercially viable heat engines for those temperatures. Care has to be taken with the design of the employed cycle and the selection of components for use within these heat engines as the capital cost makes up the vast majority of the lifecycle cost of the equipment.

Thermodynamic cycle selection

One of the more fundamental design decisions to be made when developing a heat engine for use with LGH is the choice of thermodynamic cycle around which the machine will be based. Considerable amounts of research have been conducted looking at various cycles that are applicable to LGH and some of the more recent investigations include examinations of transcritical cycles (Cayer et al 2009), the trilateral flash cycle (Bryson 2007) and the Rankine cycle (Mago et al. 2008). Trials of various working fluids for use in low temperature heat engines have also been conducted and include Tchanche et al. (2009) investigating pure fluids, and Wang et al. (2009) studying zeotropic mixtures. The available research suggests that the Rankine cycle, using a low boiling point working fluid, provides the lowest price (per unit output of electricity).

Working fluid selection

The use of less traditional working fluids, either pure or in mixtures, is another topic that has seen a considerable amount of research being conducted over a number of years. Recently, Tchanche et al. (2009) have examined a range of fluids for use with an ORC utilising LGH and concluded (as have many before them) the ideal working fluid would be one which exhibits high cycle and heat exchange efficiencies, low turbine outlet volume flow rate, manageable high side and low side pressures, develops a favourable pressure ratio upon evaporation, no ozone depletion potential, low global warming potential and is non-flammable, non-toxic and should have no material compatibility problems. Also to be considered are: the specific volume of the liquid working fluid, the thermal conductivity, the viscosity, the surface tension, the thermal stability as well as the cost and availability of the fluid. Availability of the working fluid has become more of a concern in recent times due to the restrictions put on ozone depleting substances and the restrictions on the use of working fluids with a high global warming potential.

When selecting the working fluid for use within a particular heat engine that operates on a chosen cycle, the most critical variables upon which to base the choice are the heat source and sink temperatures. The heat source is usually fixed by what is available at the source of the heat energy, although, in the case of geothermal energy, there is the option of expending more on the drilling costs and accessing a higher temperature. In the case of the low temperature heat sink that must be used to make the engine function, this is usually dictated by the site of the plant (usually a function of a wet or dry bulb temperature or an available watercourse temperature). The working fluid choice, amongst other things, then dictates (for a given heat exchanger effectiveness) the pressures experienced by the high pressure side

and the low pressure side of the engine, the resulting pressure and volumetric ratios, the pumping power required to circulate the fluid and the efficiency of the conversion of heat energy into mechanical energy and then ultimately into electricity.

Figure 2 is a plot of the gross thermal efficiency of a regenerative Rankine cycle with refrigerant top temperatures between 50°C and 100°C. The model presented, developed by Air International Thermal Systems (AITS), has been run with a stipulation of 0°C expander entry superheat and a 20°C condensing temperature. In order to keep the unit from falling below atmospheric pressure in the condenser, a minimum condenser pressure of 130 kPa (absolute) was set. If, at 20°C condensing temperature, the pressure in the condenser (for the fluid being trialled) resulted in an absolute pressure below the set minimum of 130kPa, then the condensing temperature used was the saturation temperature of that of fluid at that minimum pressure. The requirement that the heat engine not fall below the minimum pressure is to avoid the possibility of air being drawn into the machine and consequently prevent the cycle attaining the design pressures. This is a problem experienced by conventional steam plants and various strategies are employed to allow for the removal of the trapped non-condensable air.

It can easily be seen from Figure 2 that, within the temperature range examined, that neopentane gives the highest cycle efficiency over the entire range. The fluids with the next highest efficiency

(within 1% of neopentane at 75°C) are in descending order: R114, n-butane, RC318, isobutane, R236ea, R236fa, R227ea, R124, R245fa and R142b. The Carnot efficiency has been included on the plot to give a benchmark.

In accordance with the list of working fluid selection criteria mentioned previously, several of the working fluids examined can be immediately discounted from anything but academic consideration. This list includes the fluids controlled under the Montreal Protocol due to their ozone depletion potential (R114, R124 and R142b). Consideration must also be given to the long term availability of working fluids with high global warming potential (the hydrofluorocarbons RC318, R227ea, R236ea, R236fa and R245fa)) as these fluids are coming under more and more stringent regulatory control. This elimination leaves the naturally occurring hydrocarbons, with their lack of ozone depletion potential and no direct global warming potential, as the remaining choices from this process. There is the concern about the flammability of these substances and also a heat engine must be designed with their cost and availability in mind.

Rankine cycle heat engine components

A simple schematic diagram of a heat engine using a regenerative Rankine cycle can be seen in Figure 3. As can be seen from the diagram, such a device is made up from six major components. The largest expense for a single item is the expander for most systems.

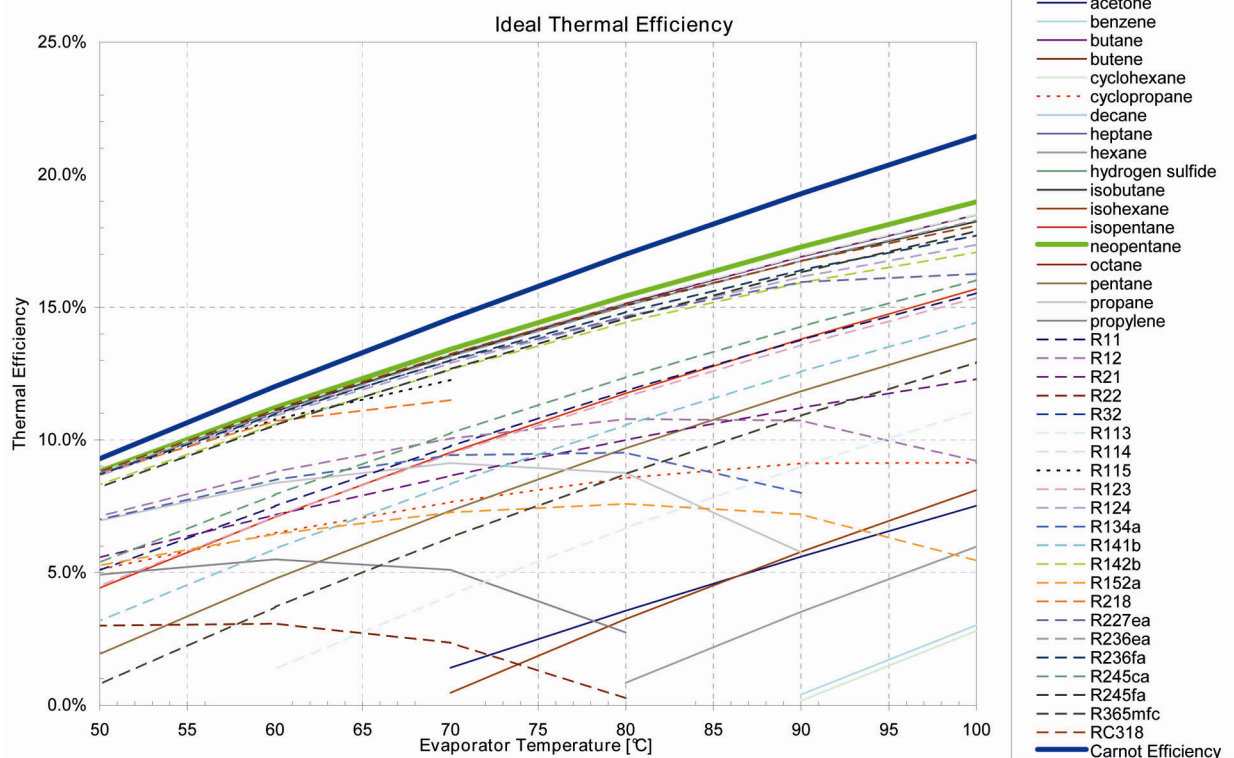


Figure 2: Efficiency figures for an ideal Rankine cycle with a 20°C condensing temperature (or a 130 kPa-abs condensing pressure if the saturation temperature is below this at 20°C) for a range of potential working fluids neglecting any parasitic loads.

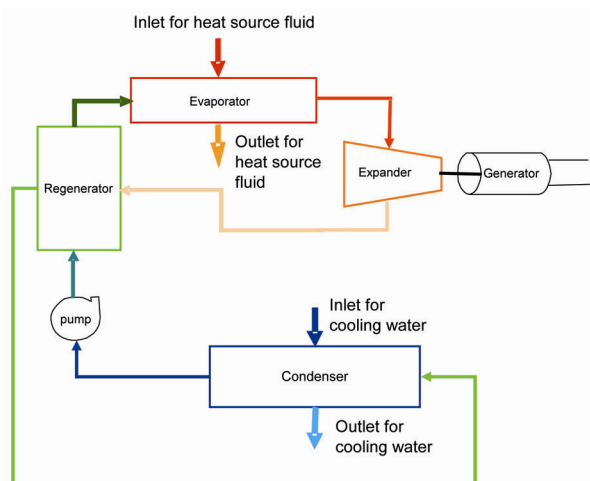


Figure 3: Simple schematic diagram of components used in a regenerative Rankine cycle heat engine.

From the diagram it can also be seen that in a regenerative Rankine cycle, there are three components responsible for heat exchange these being the evaporator, condenser and the regenerator. It is obvious that there is a requirement for the expander and the three heat exchangers to be sourced cost effectively, whilst still performing to an acceptable level, in order for the low temperature heat engine to be a commercially viable option for converting heat into power.

The goal for the design of a heat engine is to produce electricity at the lower price possible per unit of energy sent out. This leads to the two competing design drivers being the maximising of efficiency of energy conversion while minimising the capital outlay. It is usual to find that a more efficient or effective a component is, the more expensive it is. Therefore an optimal balance has to be sought between component efficiency and cost.

Expanders for use with organic Rankine cycle heat engines

As mentioned previously, the single component that has the greatest bearing on the viability of a low temperature heat engine is the expander. Depending upon the configuration, design and materials chosen the cost of sourcing a state of the art expander can easily cost more than the rest of the components combined. Most commercially available turbines developed for power production purposes were designed for service with steam power plants and are now in mass production making them relatively inexpensive to purchase. These units are not, however, suitable for use with many low boiling point working fluids such as hydrocarbons. This is due to the significantly higher molecular mass of the low boiling point working fluids requiring the shafts to carry significantly higher torque at

substantially lower rotational speeds. While it is possible to source such a turbine with a very high efficiency, the relatively low numbers in production cause such a unit to be sold at a price high enough to make the ORC unit economically unviable.

Scroll and screw type compressors lend themselves well to operating in reverse as ORC expanders. They are both mass produced leading to their cost effective application to low temperature ORC units. Scroll compressors are widely used in the refrigeration and air conditioning industries and from experience gained in those industries, some concerns have been raised about the durability of the scroll end "tips". There is not the same concern regarding screw compressors. In addition to this, experience has shown that the largest size of scroll compressor able to be sourced cost effectively on a commercial scale is 500kW. This is significantly smaller than that required for most commercial power production applications. The use of low temperature heat sources allows the selection of screw machines whose normal duty would be compressing gases such as air. There are many of these units produced each year and so their selection means that they are a cost effective part for use in ORC engines.

ORC Experimental apparatus

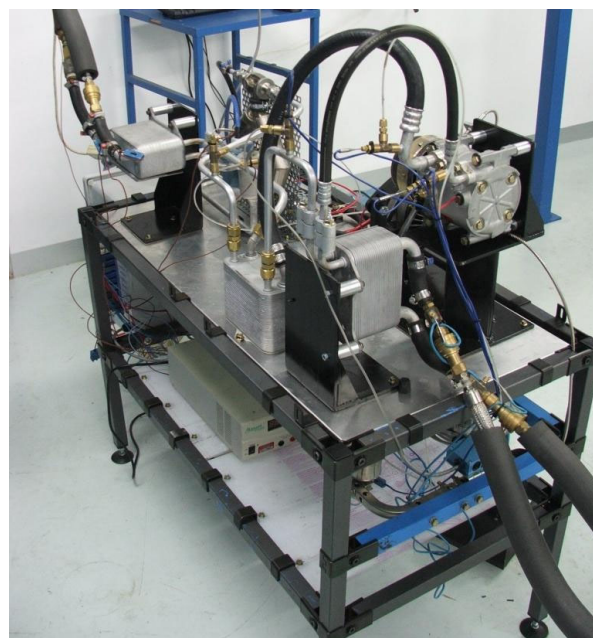


Figure 4: Photograph of the AITS ORC test apparatus.

In order to conduct a parametric experimental investigation of a small scale ORC, AITS has built and instrumented a unit by adapting parts manufactured for other purposes. A photograph of the experimental unit, as tested by RMIT and

AITs, appears in Figure 4. The heat engine, constructed almost entirely from automotive parts either built or supplied by AITs, includes a range of pressure and temperature probes and is part of a closed loop automated system that ensures that the required test parameters are controlled to within the testing tolerance specification. The system uses an air conditioning scroll compressor, operating in reverse, as an expander. A range of tests have been performed and include (amongst others) a charge determination procedure and an evaporator exit superheat sensitivity trial.

Results from the AITs experimental ORC rig

For the purposes of exploring the effect of varying the heating source fluid temperature upon this specific unit, experiments were conducted where the heat source fluid temperature was varied while holding the cooling source fluid temperature at $14.3^{\circ}\text{C} \pm 1^{\circ}\text{C}$ and the superheat of the working fluid was maintained at $10^{\circ}\text{C} \pm 1^{\circ}\text{C}$. The amount of working fluid superheat was controlled to within its tolerance band by the computerised control system via varying the flow rate of the working fluid pump. It should also be noted that the fluid flow rates of the heat source and sink were held constant for all of the tests. Figure 5 shows a plot of the power output as the heat source fluid is varied. The output power is given a normalised percentage of the output power of the apparatus when the heat source fluid temperature was 75°C .

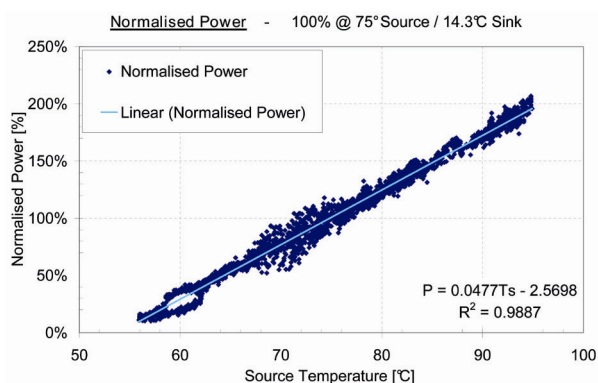


Figure 5: Normalised power output as a function of the heating source fluid temperature for a constant cooling source temperature.

Figure 5 indicates that there is a high degree of linearity in the relationship between the normalised power and the temperature of the heating source. The figure also shows that, for the experimental apparatus examined, the power output at 95°C source temperature is approximately double that at 75°C , while at a source temperature of 64°C is approximately half that produced at 75°C .

Tests trialling the sensitivity of the power output of the apparatus to the temperature of the cooling

source fluid were also undertaken. The experiments were conducted with a $75^{\circ}\text{C} \pm 1^{\circ}\text{C}$ heating source temperature throughout the tests and the superheat of the working fluid was maintained at $10^{\circ}\text{C} \pm 1^{\circ}\text{C}$. As with the tests examining the sensitivity of the unit to the heat source temperature, the amount of working fluid superheat was controlled in the same manner as for the heat source sensitivity tests. Figure 6 shows a plot of the power output as the cooling fluid temperature is varied. The power out is presented as a normalised percentage of the output power of the apparatus when the cooling source fluid temperature was 14.3°C .

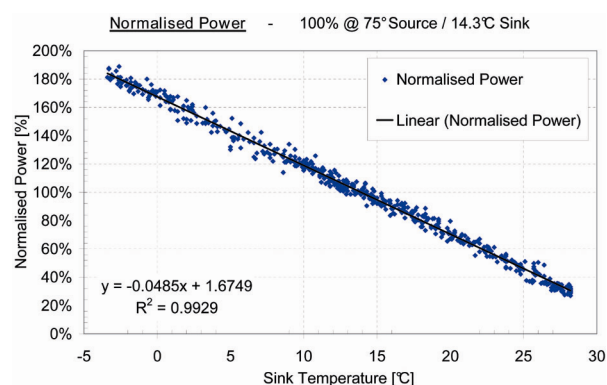


Figure 6: Normalised power output as a function of the cooling source fluid temperature for a constant heat source temperature.

Figure 6 also shows a high degree of linearity in the relationship between cooling fluid temperature and the output power of the rig. At a cooling fluid sink temperature of -3°C , the output of the rig is approximately 1.8 times that of the 14.3°C benchmark trials. At the top cooling fluid temperature of 28°C the output is approximately 30% of the nominal power output.

Conclusions

The available literature on the cost effective production of electricity using ORC heat engines that employ low temperature heat sources is limited. Although the preliminary results of this investigation have been concerned with the performance of the ORC, the indications are that it is cost effective to recover and convert low grade heat to electricity. For heat source temperatures of 75°C and below, the selection of the expander is a key concern due to the trade off between efficiency and capital cost. The selection of a working fluid that achieves a favourable ideal cycle efficiency is also of great impact upon the viability of these machines that utilise low grade heat. Results from the experiments jointly conducted by Air International Thermal Systems and RMIT University show a direct proportionality between achieved power output and heat source temperature within the range examined. It was

also found that, within the range examined, there was also a direct proportionality between the heat sink temperature and the achieved power output. Also notable is the fact that power was able to be produced by the experimental unit from around a 55°C heat source temperature while using a 14.3°C heat sink.

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