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Behaviour of ORC low-temperature power generation with different refrigerants

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This article presents and discusses the behaviour of low-temperature Organic Rankine Cycle for power generation using waste heat. The performance has been compared at low- and medium waste heat temperatures to organic and non-organic fluids. In this article, energy and exergy analysis have also been presented and compared for the behaviour of the different refrigerants. Results showed that the use of refrigerant mixtures maximises the waste heat recovery and produces power from low- and medium waste heat with less exergy destruction compared to other working fluids.

Keywords: Organic Rankine Cycle; refrigerants; behaviour; low temperatures

1. Introduction

It has been established that the method of exergy analysis provides the most accurate and insightful assessment of thermodynamic characteristics of any process and offers a clear quantitative indication of irreversabilities and the degree of matching between resources and the end-use energy flows (Olz *et al*. 2007, Sami 2010).

It is estimated that energy wasted by all the US industrial facilities could produce power equivalent to 20% of US electricity generation capacity without burning any additional fossil fuel, and could help many industries to meet recent global warming regulations. Furthermore, by using the heat recovery and Organic Rankine Cycle (ORC) technologies and tapping into other renewable resources such as solar energy and geothermal energy as well as Ocean Thermal Energy Conversion, our dependence on fossil fuel will be significantly reduced.

Recent problems in electricity production emphasise the urgent need for a renewable approach to support the current electricity system, increase its existing capacity, and, equally important, benefit the environment by reducing the need to build more power plants and utilise environmentally friendly chemicals and heat-recovery technologies such as ORC.

The ORC is a Rankine Cycle that uses a heated chemical instead of steam as found in the conventional Rankine Cycle. Non-organic and organic fluids are used in ORCs.

Cycle include pure refrigerants and refrigerant mixtures (Badr *et al*. 1985, Berger and Berger 1986, Larjola 1995, Verschoor and Brouwer 1995, Klaver and Nouwens 1996, McLinden 1998, Angelino and di Paliano 2000, Sami *et al*. 2000, Rosen 2001, World Energy Assessment 2001, Obernberger *et al*. 2002, Canada *et al*. 2004, Koolwijk 2004, Andersen and Bruno 2005, Rosen *et al*. 2005, Kanoglu *et al*. 2007, Madhawa *et al*. 2007, Ozgener *et al*. 2007, Aoun. 2008, Brasz 2008, Chiles 2009, Sami 2009, Tchnache *et al*. 2009). Organic compounds generally have a higher molecular mass. This gives relatively small volume streams and results in a compact size ORC unit. It also enables high turbine efficiency up to 80% (Klaver and Nouwens 1996, Koolwijk 2004). Unlike steam, organic compounds do not form liquid droplets upon expansion in the turbine. An absence of steam prevents erosion of the turbine blades and enables design flexibility on the heat exchangers (Klaver and Nouwens 1996).

The use of ORC in low-temperature applications under 100°F (37°C) is very limited and depends upon the thermodynamic and thermophysical properties of the working fluid. A cost-effective optimum design criterion for Organic Rankine power cycles utilising low-temperature geothermal heat sources has been presented (Angelino and di Paliano 2000). Evaporation and condensation temperatures, geothermal and cooling water velocities were varied in the optimisation method. The optimum cycle performance was evaluated and compared for working fluids that include

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ammonia, HCFC123, *n*-Pentane and PF5050. Exergy analysis shows that efficiency of the ammonia cycle has been largely compromised in the optimisation process than that of other working fluids. The fluids, HCFC 123 and *n*-Pentane, have better performance than PF 5050.

Theoretical performances as well as thermodynamic and environmental properties of few fluids have been comparatively assessed for use in low-temperature solar ORC systems by Tchnache *et al.* (2009). Efficiencies, volume flow rate, mass flow rate, pressure ratio, toxicity, flammability, ozone depletion potential and global warming potential (GWP) were used for comparison. Of 20 fluids investigated, R134a appears as the most suitable for small-scale solar applications. R152a, R600a, R600 and R290 offer attractive performances but need safety precautions, owing to their flammability.

Most promising working fluids are chlorofluorocarbons (CFCs) and other existing ones are either not environmentally acceptable and or have less-heat recovery efficiency. And in particular, World Energy Assessment IEA (2001) refrigerant mixture's components include hydrochlorofluorocarbon (HCFCs) that are scheduled to be banned by the year 2010. Therefore, a new family of hydrofluorocarbon (HFCs) refrigerant mixtures have been developed (Sami 2009) to overcome the deficiencies of the current working fluids.

This research study has been undertaken to enhance our understanding of the behaviour of ORCs using the different refrigerant and refrigerant mixtures and impact on the ORC performance. Energy and exergy analyses were applied to better understand the benefits of using refrigerant mixtures in various applications.

2. Organic Rankine Cycle

An ORC engine is a standard steam engine that utilises heated vapour to drive a turbine (Sami 2010). In the ORC, a heated organic chemical is used instead of a superheated water steam. The organic chemicals used by an ORC include CFCs and most of the other traditional refrigerants, iso-pentane, CFCs, HFCs, butane, propane and ammonia and recently refrigerant mixtures (Larjola 1995, Verschoor and Brouwer 1995, Klaver and Nouwens 1996, McLinden 1998, Sami *et al.* 2000, Rosen 2001, World Energy Assessment 2001, Canada *et al.* 2004, Koolwijk 2004, Rosen *et al.* 2005, Kanoglu *et al.* 2007, Madhawa *et al.* 2007, Ozgener *et al.* 2007, Aoun. 2008, Tchnache *et al.* 2009, Sami 2009). The traditional refrigerants require a high-temperature heat source.

3. Theoretical considerations

There are four processes in the ORC similar to the steam cycle, each changing the state of the working fluid. These states are identified by number in the diagram. First, the working fluid is pumped from low-to-high pressure by a pump. Pumping requires a power input (e.g. mechanical or electrical). Second, the high-pressure liquid enters a boiler where it is heated at a constant pressure by an external heat source to become a superheated vapour. Common heat sources for power plant systems are coal, natural gas, or nuclear power. Third, the superheated vapour expands through a turbine to generate power output. Ideally, this expansion is isentropic. This decreases the temperature and pressure of the vapour. Fourth, the vapour then enters a condenser where it is cooled to become a saturated liquid. This liquid then re-enters the pump and the cycle repeats (Sami 2010).

4. System equations

4.1. *Energy analysis*

Each of the first four equations is easily derived from the energy and mass balance for a control volume. The fifth equation defines the thermodynamic efficiency of the cycle as the ratio of net power output to heat input.

$$\frac{Q_{\text{in}}}{\dot{m}} = h_3 - h_2 \tag{1}$$

$$\frac{Q_{\text{in}}}{\dot{m}} = h_g - h_1 \tag{2}$$

$$\frac{\dot{W}_{\text{turbine}}}{\dot{m}} = h_3 - h_4 = (h_3 - h_4 s) \times \eta_{\text{turb}} \tag{3}$$

$$\frac{\dot{W}_{\text{pump}}}{\dot{m}} = h_2 - h_1 \approx \frac{\upsilon_1 \Delta p}{\eta_{\text{pump}}} \approx \frac{\upsilon_1 (p_2 - p_1)}{\eta_{\text{pump}}} \tag{4}$$

$$\eta_{\text{therm}} = \frac{\dot{W}_{\text{turbine}} - \dot{W}_{\text{pump}}}{\dot{Q}_{\text{in}}} \approx \frac{\dot{W}_{\text{turbine}}}{\dot{Q}_{\text{in}}} \tag{5}$$

$$\text{NHR} = \frac{\text{Q}_{\text{in}}}{\text{W}_{\text{turbine}}} \tag{6}$$

In a real ORC, the pumping and expansion processes are non-reversible and entropy is increased during the two processes. Modern practices are implemented such as reheat and regenerative cycles. In the regenerative ORC, the working fluid is heated by steam tapped from the hot portion of the cycle. This increases the average temperature of heat addition, which in turn increases the cycle efficiency.

4.2. *Exergy and energy efficiency*

The use of exergy in assessing the power cycles such as ORC is highly beneficial. The efficiency of the ORC based upon exergy, as the ratio of total exergy output to exergy input:

$$\eta_{ex} = \frac{Ex_{out}}{Ex_{input}} = \frac{(W_{net} + Ex_{heat})}{Ex_{input}} \tag{7}$$

and can be equal:

$$= 1 - Ex_{dest}/Ex_{input} \tag{8}$$

where Ex_{heat} represents the rate of exergy transfer associated with transfer of heat, Ex_{dest} is the rate of exergy destruction and W_{net} represents the net work.

In this analysis, the thermal exergy rate is expressed in terms of the decrease of the hot fluid:

$$\begin{aligned} Ex_{heat} &= -\Delta Ex_{heat} - hot \\ &= \dot{m}[h_i - h_e - T_o(s_e - s_i)] \end{aligned} \tag{9}$$

The subscripts i and e refer to the inlet and exit states of the fluid in the heat exchanger and $\dot{m}$ is the mass flow rate of the fluid circulating in the ORC.

Finally, the ORC efficiency based upon the rate of exergy destruction is

$$\eta_{ex} = \frac{\left\{ \begin{array}{l} (W_{net,out} + \dot{m}[h_i - h_e \\ \quad - 00A0T_o(s_i - s_e)]) \end{array} \right\}}{Ex_{input}} \tag{10}$$

and the rate of exergy input is

$$Ex_{input} = \dot{m}[h_e - h_i - T_o(s_e - s_i)] \tag{11}$$

In the particular case of heat recovery across a waste heat boiler,

$$Ex_{input} = \dot{m}[C_p(T_e - T_i - T_o(s_e - s_i)] \tag{12}$$

and the entropy change of flue gases is

$$(s_e - s_i) = C_p \mathrm{Ln}(T_e/T_i) \tag{13}$$

Furthermore, the second law efficiency can be given as follows:

$$\eta_{II} = \frac{Ex_{output}}{Ex_{input}} \tag{14}$$

$$Ex_{output} = (\Delta h - T_o(\Delta s))_{turbine,net} \tag{15}$$

Thermodynamic and thermophysical properties are determined using the well-known NIST REPROP.8 program (McLinden 1998). In addition to these properties, the conservation are solved for each control volume using finite difference convergence technique to obtain the thermal behaviour for each component. Each component was represented by a finite control volume.

5. Discussion and analysis

In order to analyse the ORC cycle using various refrigerant and refrigerant mixtures, the set of equations referred to have been coupled with the equation of state calculated by the REFPROP program (McLinden 1998) to obtain the thermodynamic and thermophysical properties of the fluids in question which were solved using the finite difference control volumes.

5.1. *Selection of a working fluid*

The refrigerants selected for this comparative study fulfil the following thermodynamic and thermophysical requirements; critical temperature must be higher than the highest operating temperature of the cycle to minimise the irreversibilities generated by heat transfer through the waste heat boiler, the working fluid condensing pressure should be higher than the atmospheric pressure to avoid leakage issues, the excessive superheat at the exit of the turbine should be avoided to reduce an exergetic loss, a high ratio of the latent heat of vapourisation to the liquid heat capacity, finally a high heat capacity of the liquid can lead to a higher recovered energy from the heat source and enhance the total efficiency of the cycle.

Other properties are also desirable such as large enthalpy variation in the turbine, high-working fluid density at the inlet of the turbine, higher convective heat coefficient and high-thermal conductivity and low viscosity to reduce pressure drop and maximise heat transfer. Working fluids should be non-corrosive and thermally and chemically stable, non-toxic, non-flammable and have lower GWP. Figure 1 depicts some of the working fluids candidates selected for this study, where total Rankine efficiency is plotted against boiling temperatures. It can be seen that at the low temperature range ($<140°C$) only a handful of refrigerant can be used since their heat recovery efficiency is 10%. It is also quite clear from the comparison displayed in Figure 1 that water and methanol have

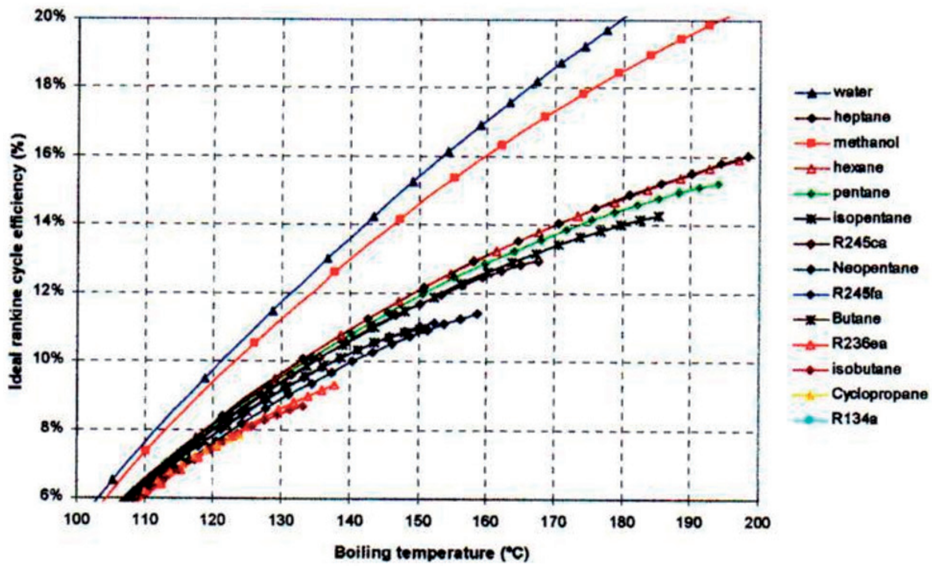

Figure 1. Ideal Rankine cycle efficiency (Aoun 2008).

high-energy performance and R-245fa has the lowest efficiency but remains a good candidate for ORCs. However, isobutene has more potential than R245fa in ORC applications despite the fact it is a highly flammable refrigerant.

List of the possible working fluids potential considered for the ORC as possible candidates that have been used in operational Rankine engines such as CFC, HFC, HCFC and Hydrocarbons. HFC refrigerant mixtures proposed include the following components HFC 125, HFC 134a, HFC 245fa, HFC 152a, HFC 236ea and HFC 245ca mixed up in different combinations at various concentrations (World Energy Assessment IEA 2001).

The comparative study presented here was conducted at the following operating conditions: boiling pressure under 300 psi (2000 kPa) and condensing temperature under 12°C (55°F).

A comparative study has been made between the behaviour of a refrigerant mixture (Sami 2009) and other refrigerants reported in the literature of similar applications. The system simulation of the various refrigerants: R-11, R-114, R-245fa as well as (Sami *et al*. 2010) and proposed mixtures by Sami (2009) under operating conditions; 112°C (235°F) and 230 psi (1585 kPa) at the waste heat boiler exit and 29°C (85°F) and 10 psi (68 kPa) at the condenser inlet. System capacity is 125 kW. Our ORC system is retrofitted to a combined heat and power (CHP) system. The CHP system is a gas turbine system with a steam generator. Typically, the temperature of the flue gases at the gas turbine exit varies between 426°C (800°F) and 537°C (1000°F). In addition, the flue gas temperature after the steam generator is around 148°C (300°F) to 204°C (400°F). At this temperature range, we can recover heat and produce power at significantly higher efficiency using the proposed mixture compared than other fluids. The results of the comparative study have been plotted in Figure 2. It is evident from this figure that using our refrigerant mixture results in lower values of net heat rate (NHR) and more power production at the same heat input at the waste heat boiler. This is mainly due to the lower boiling temperature of the mixture and higher latent heat of evaporation compared to the refrigerants under investigation.

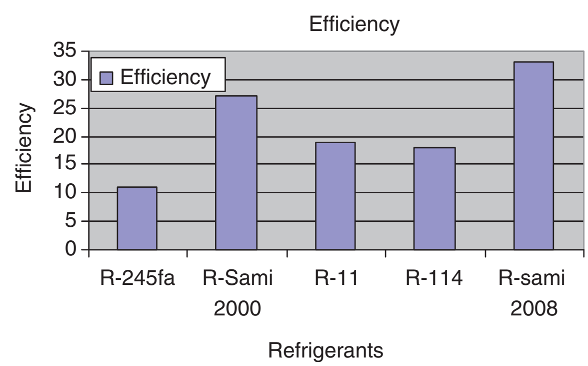

Figure 2. Efficiency for different refrigerants.

Figure 2 displays the system efficiency using the proposed refrigerant mixture (Sami 2009) compared to the other refrigerants under investigation. It is

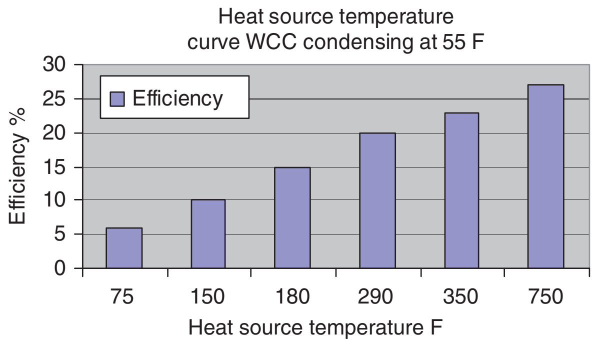

Figure 3. Cycle efficiency using Sami (2008) refrigerant mixtures.

apparent that the refrigerant mixture has superior cycle efficiency. This is due to the increase in work produced at the same heat source. This comparison is significant since it compares the refrigerant mixture efficiency to that of R-245fa which is currently used and considered as alternative to the CFCs R-11 and R-114 in chillers and ORC applications. This is also due to the high heat transfer ratio between the thermal energy and kinetic energy at the turbine side as well as the pressure ratio.

The impact of using (Sami 2009) refrigerant mixtures on the ORC efficiency at various heat source temperatures is shown in Figure 3, where a water cooled condenser is used at condensing temperature of 12.8°C (55°F). The data clearly shows the higher the flue gas temperature the more power produced at the turbine side. This result is expected since increasing the flue gas temperature increases the thermal energy dissipated at the turbine and converted to kinetic energy. The data displayed in this figure clearly shows that retrofitting our proposed ORC will significantly enhance the efficiency and reduce the NHR and will also have a positive impact on the environment by cooling down the flue gases, reducing emission and global warming.

Another comparative study has been conducted to simulate an ORC heat recovery process using the refrigerant mixture and isobutene as working fluid under waste heat source input and output.

Input and output temperatures to the waste heat boiler are 260°C (500°F) and 150°C (302°F), respectively, and that of the condenser cooling conditions are 25°C (77°F) and 36°C (97°F), respectively. The results of this particular simulation are displayed in Figures 4 and 5.

It is quite evident from the simulation results presented in these two figures that the high heat capacity of the refrigerant mixture liquid resulted in higher recovered energy from the heat source and then increases the total efficiency of the cycle. Furthermore, the mixture enthalpy variation in the turbine is larger than the isobutene, and therefore increased the efficiency of the thermodynamic cycle. Another factor to be considered in this comparison is that the mixture density at the inlet of the turbine is higher and will result in a reduced turbine size. In addition, the convective heat coefficient of the mixture is significantly higher than the isobutene and has higher thermal conductivity and low viscosity, which is attributed to lower frictional pressure drops and enhancement of the convective heat transfer coefficients.

An ORC plant in a Biomass based cogeneration system with split type ORCs with two loops of heat source, where input and outlet conditions of the high-temperature loop to the waste heat boiler are 310°C (590°F) and 250°C (482°F), respectively. The low-temperature loop has the input and out conditions of 310°C (482°F) and 130°C (266°F), respectively. The working fluid for the Biomass ORC is isobutene. A water cooled condenser was used and conditions at the inlet and outlet of the condenser are 60°C (140°F) and 80°C (176°F), respectively. Figure 6 depicts the comparison between the output gross power of the ORC using isobutene and the blend. It is evident that under the same heat source conditions, the blend is capable of recovering more heat from the heat source than the isobutene. In addition, the graph also shows that the blend has higher high heat to power efficiency on the average 28.6% compared to 19% for the isobutene.

In most solar and geothermal applications heat transfer fluids circulating in liquid heating flat tube and or evacuated collectors and wells are at temperatures between 95°C (203°F) and 115°C (234°F). ORC coupled with geothermal wells or solar collectors for solar thermal power production mostly use isobutene and or R245fa and yield low-heat recovery efficiency under 7% depending upon the heat transfer fluid temperature (Mills 2008). Data displayed in Figures 4 and 7 demonstrates the added value using the refrigerant mixture under investigation and particularly at temperatures lower than the 82.2°C (180°F). In addition, current ORCs based on solar and geothermal failed to generate electricity with economically viable solutions because of the thermodynamic characteristics of the working fluids used. However, the use of the proposed refrigerant mixtures enables us to produce electricity at the aforementioned temperature range efficiently and with less complication and less expense since ORC can be connected directly to the well water.

Figures 8 and 9 have been constructed to present the energy and the second law thermal efficiencies analysis and exergy performance results of the various

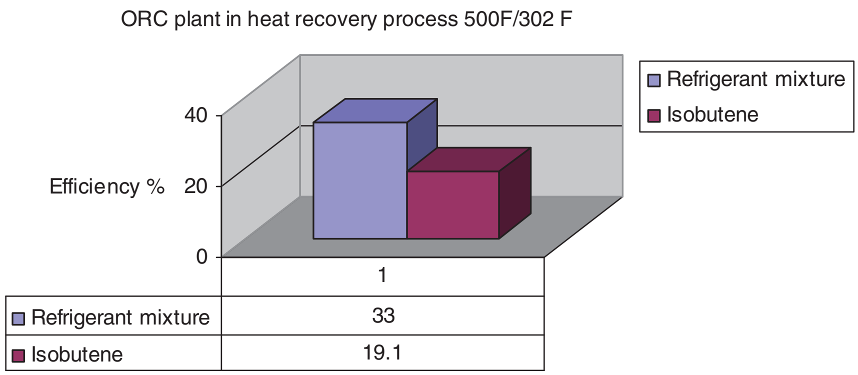

Figure 4. ORC plant heat-recovery process.

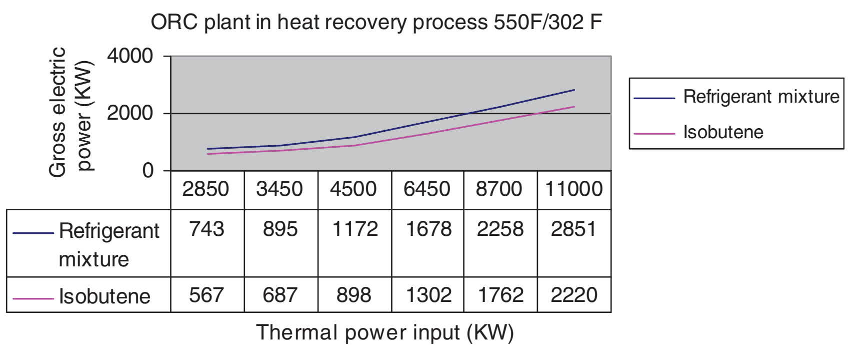

Figure 5. Comparison between refrigerant mixture and isobutene.

fluids used in this study; namely R245fa, R-114, R-11, Sami *et al.* (2000) and the quaternary refrigerant mixture (Sami 2009). To facilitate the comparisons of the refrigerants under questions, the same heat source and sink conditions were used; 4.5 MW, 112°C (235°F) and 29°C (85°F), respectively. Operating parameters were selected to yield the same flow rate in each case.

Clearly ORC operating with a refrigerant mixture (Sami 2009) has the highest energy efficiency compared to the other fluids. Examining the exergy efficiencies displayed in Figures 8–10 shows that the quaternary refrigerant mixture has the higher thermal efficiency compared to the other refrigerants under study. R-245fa has the lowest thermal efficiency because of its low-boiling point compared to the quaternary refrigerant mixture as well as others.

Similar results can be observed from Figure 8 when examining the second law efficiencies. Furthermore, the results presented in Figure 9 clearly show that the exergy destruction is much lower in conditions, where the quaternary refrigerant mixture (Sami 2009) is used compared to other refrigerants. In addition, the results displayed in this figure also show that R-245fa has the highest exergy destruction among the refrigerants under investigation in this study. It is believed that the highest exergy destruction was observed for R245fa because of its low-boiling point.

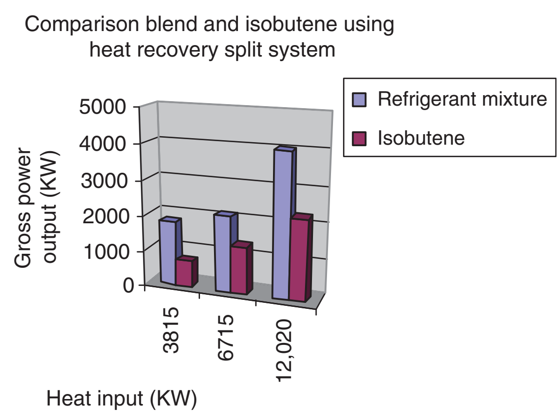

Figure 6. Comparison of ORC plant in biomass based cogeneration.

It is worthwhile mentioning that when mixtures are use, the composition of the refrigerant mixture can be

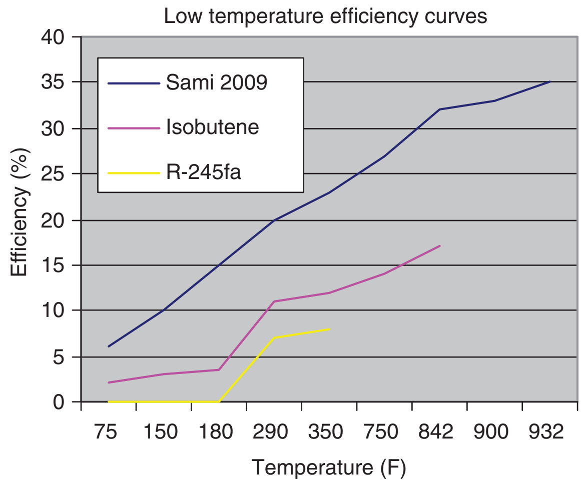

Figure 7. Advantages of proposed refrigerant mixture in low-temperature applications.

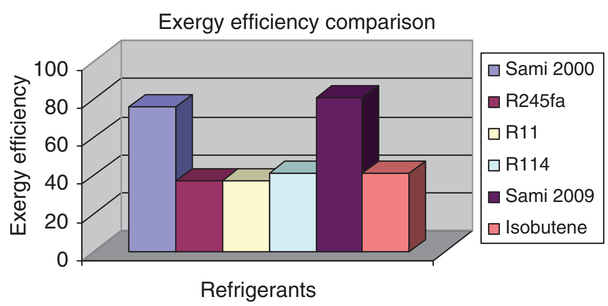

Figure 8. Exergy efficiency comparison of refrigerants.

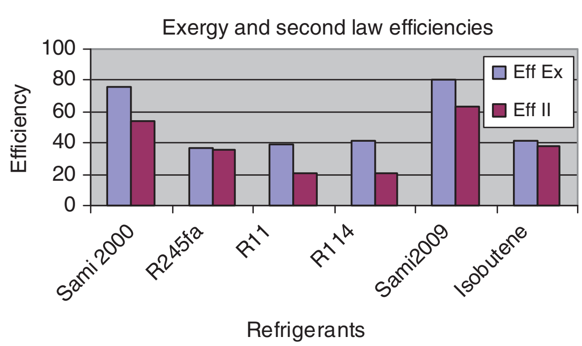

Figure 9. Comparisons between exergy and second law efficiencies.

adjusted to boil the mixture and generate power at a wide range of temperatures. However in Sami (2009) several refrigerant mixtures have been developed to match the heat transfer profile of the heat source at various temperatures and in particular the low-temperature application range to reduce losses,

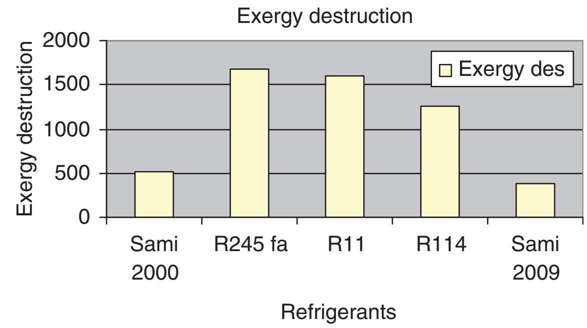

Figure 10. Exergy destruction.

minimise the irreversibilities generated by heat transfer through the waste heat boiler, reduce an exergetic loss, increase ratio of the latent heat of vapourisation to the liquid heat capacity, and heat capacity of the liquid can lead to a higher recovered energy from the heat source and enhanced the total efficiency of the cycle.

Furthermore, the use of quaternary refrigerant mixtures in the ORC reduces emissions and global warming. Compared to typical fossil fuel, using the new HFCs refrigerant mixtures reduces NOx by over than 4 tons per year and significantly reduces CO_2 emission.

6. Conclusions

In this analytical study, behaviour of ORC using various refrigerants has been analysed and discussed. The study clearly showed that the use of mixtures effectively capture heat at a wider range of temperatures compared to other refrigerants.

Various comparative studies have also been presented using energy, exergy analysis to demonstrate the superior performance of refrigerant mixture over other working fluids and produce power from low- and medium-waste heat with less exergy destruction.

Nomenclature

C_p	specific heat (kJ/kg K)
Ex	exergy rate (kJ/kg/s)
η_{ex}	exergy efficiency
η_{pump}	pump efficiency
η_{turb}	turbine efficiency
η	cycle efficiency
$\dot{Q}_{in}$	heat input rate (kJ/s)
Q	heat transfer (kJ/kg)
$\dot{m}$	mass flow rate (kg/s)
$\dot{W}$	mechanical power used by or provided to the system (kJ/s)

NHR thermodynamic efficiency of the process (power used for turbine per heat input, net heat rate) (Btu/kWh)

h_1, h_2, h_3, h_4 these are the specific enthalpies at indicated points on the T–S diagram (kJ/kg)

ΔP pressure drop (kPa)

S entropy (kJ/kg K)

T temperature (K)

v specific volume (m^3/kg)

W work (kJ/kg)

Subscripts

3 turbine inlet and saturated vapour

3′ turbine inlet superheated vapour

4 turbine outlet and wet vapour

4′ condenser inlet and saturated vapour

2 waste heat boiler inlet

1 inlet to the pump

in input to waste heat boiler

i input

e output

des destruction

o ambiant

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