A COMBINED ORGANIC RANKINE CYCLE-HEAT PUMP SYSTEM FOR DOMESTIC HOT WATER APPLICATION

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ABSTRACT

This paper investigates a novel system to improve the efficiency of using natural gas for domestic heating. The exhaust from a gas burner powers a small-scale Organic Rankine Cycle (ORC) system using hexane as the working fluid, which is used to directly drive the compressor of a heat pump, using R134a as the working fluid. Water is heated from ambient by passing it through three heat exchangers, the condenser of the Heat Pump, the condenser of the ORC, and the secondary heat exchanger that is heated by the hot flue gas from the burner after it transfers the heat to the evaporator of the ORC subsystem. By using the heat generated from the burning of gas in a burner in this way, a fuel-to-usable-heat efficiency of up to 160% is projected, outperforming the other technologies discussed, giving it the potential to significantly reduce energy demand and carbon emissions. This paper investigates the effect of varying ambient conditions upon the cycle, namely the temperature of ambient air, which has a strong effect on the performance of the heat pump.

INTRODUCTION

The UK has set an ambitious target to cut its greenhouse gas emissions by at least 80% by 2050, relative to 1990 levels [1]. Currently, heat accounts for nearly half of the energy consumption in the UK and about a third of the nation's carbon emissions [2]. Around 80% of heat is used in homes and other buildings, and the remaining 20% is used in various industrial sectors. The majority of the UK houses are heated by gas boilers. There are currently around 27 million houses in the UK so the residential heating approximately accounts for 92.4 MtCO2 per year [3]. In order to achieve the UK's carbon reduction target, the residential heating sector has to be substantially decarbonised.

There is a wide range of technologies at different stages of development for domestic heating applications, including electrical resistive heaters, gas boilers, heat pumps (HPs), micro-CHP (combined heat and power) systems. A comparison between different heating technologies, based on a fuel-to-heat efficiency, defined as the ratio of the useful heat production to the energy contained in the consumed fossil fuel, is shown in Figure 1.

Since 2005, most gas boilers fitted in the UK, as required by government policy, are condensing boilers which have a fuel-to-heat efficiency of around 90% by recovering some heat from flue gas. There are still a large number of non-condensing gas boilers in operation, which have a fuel-to-heat efficiency around 80% [4]. Electrical resistive heaters have near 100% efficiency to convert electricity to heat, but their fuel-to-heat efficiency is around 32-40% since electricity is largely generated from fossil fuels.

An electrically-driven HP is an ideal alternative technology for domestic heating if low carbon electricity is supplied. The Coefficient of Performance (COP) of a HP is defined as the ratio of the useful heat production to the consumed electric power. To achieve any saving of carbon emissions relative to gas boilers, the HP would require a COP of over 2.5 with the current UK electricity supply mix [5]. Both air- and ground-sourced HPs installed in the UK to date have COP in the range of 1.2 to 3.3 [6]. This is consistent with theoretical research into the performance of heat pumps [7]. Fossil-fuelled power plants have efficiency around 34-42%, and the loss in electricity transmission and distribution is about 6% [1]. Assuming an average COP of 2.5, the overall fuel-to-heat efficiency of an electrically-powered HP would be around 80-100%.

Micro-CHP comprises a group of emerging technologies that can generate some electricity (e.g. 1-2 kW) from the high temperature combustion gas of boilers via a miniature power generator (e.g. engine or fuel cell), while the rejected heat is utilised for residential heating [3]. However, varying and conflicting demands for heat and electricity minimise the overall carbon emission savings. Other technologies such as biomass fuelled heating systems and solar thermal devices are available for water heating, but they are not practical for space heating in the UK [3].

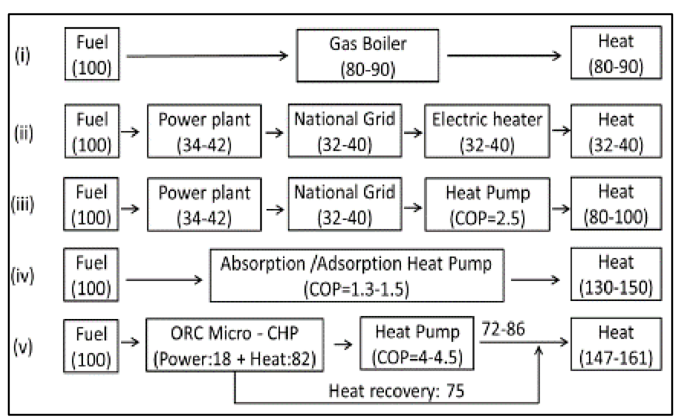

Figure 1 Comparison of different strategies for domestic heating. (i)=Conventional gas boiler, (ii)=Electric heater, (iii)=Electricity-driven heat pump. (iv)=Adsorption/Absorption heat pump, (v)=System Proposed in this paper.

Gas-driven absorption [5] and adsorption [8] HPs have potential to reduce carbon emissions. Gas-driven absorption HPs have been successfully used in large scale district heating applications. Most recently, domestic scale gas-driven absorption HPs have been launched to the European market claiming fuel-to-heat efficiency around 130-150% [9]. Gas-powered adsorption HPs have also been researched recently, achieving fuel-to-heat efficiency about 130% [9]. However, the high capital cost hinders uptake.

Internal combustion engine-driven HPs have been commercialised by a few Japanese manufacturers, targeting large scale district heating applications [10], but high engine noise is a big obstacle for their domestic application.

Heat pumps themselves perform better when the step-up in temperature is lower. This means that they will operate at a higher COP either when the ambient temperature is higher, or the required water outlet temperature is lower [11] [12] [13].

This paper proposes and models a combined ORC-HP system, which works as follows. Natural gas burns in the combustion heat exchanger and transfers heat to the evaporator of the ORC. The ORC directly drives the compressor of the Heat Pump via a common driveshaft. The condensers of the Heat Pump and ORC both reject heat to a stream of water, bringing it up to about 35°C after passing through the Heat Pump, and 55°C after the ORC. Finally, the exhausted hot gas from the combustion heat exchanger is used to bring the final temperature of the water up to the 60°C required for domestic applications [14]. This process is shown in Figure 2. The combined cycle means that the Heat Pump can operate at a higher Coefficient of Performance than a heat pump operating in isolation due to the lower temperature of water required to exit the heat pump's condenser [11].

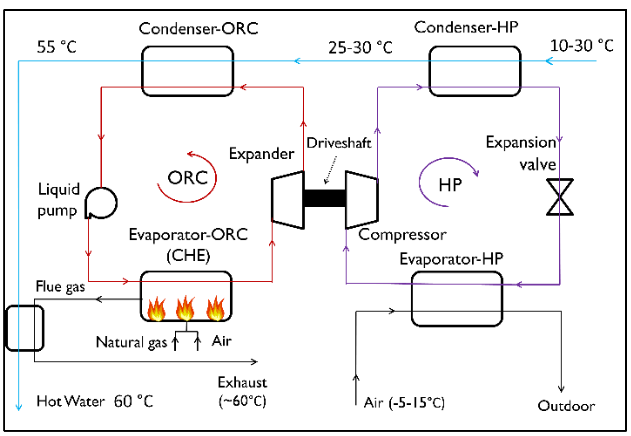

Figure 2 Diagram of the proposed combined cycle

The idea of combining power generation cycles and heat pump cycles has not been widely investigated in the past. Demierre et al. proposed a double Rankine cycle system using a shaft to connect a turbine and a compressor [15], and the results of their prototype shows that such a system has great potential for domestic heating application. However, an extensive literature review has revealed no other investigations of this cycle configuration, which maximises the heat pump's coefficient of performance, and minimises transmission losses and complexity by directly linking the ORC expander and the HP compressor. Cho et al [16] performed a conceptual study of a separate heat pump, driven by a separate CHP system, both producing hot water in parallel. This configuration has a relatively low theoretical heat pump COP of 3.6-4 due to the high temperature of water required. Kang et al [17] also analysed a parallel system, in which a gas turbine powers a heat pump while its waste heat provides hot water, but this again suffers from a lower heat pump efficiency due to the requirement for a high condenser temperature. Schimpf and Span [18] simulated a heat pump system capable of running in reverse as an Organic Rankine Cycle by diverting the working fluid between an expansion valve and pump, depending on flow direction. However, the ORC does not directly power the heat pump, and the waste heat from the ORC is used to recharge a geothermal heat source, not directly provide heating itself. Liu et at [19] analysed a system in which an internal combustion engine directly drove a heat pump, but in which the waste heat from the engine was used to drive an ORC instead of heating water.

STEADY STATE NUMERICAL MODEL

The numerical model developed in MATLAB consists of three separate sections: the heat pump analysis, the ORC analysis, and the analysis of the exhaust from the gas burners. All working fluid properties are provided by REFPROP 9.1 [20].

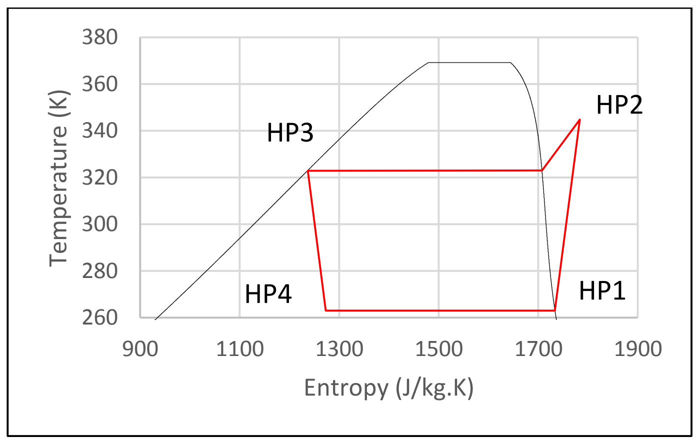

Figure 3 T-s Diagram of heat pump cycle, showing numbering convention

The heat pump is a vapour compression cycle using R134a as the working fluid. It is assumed that there are no pressure losses in pipework, or heat exchange with the surroundings except in the heat exchangers. The issue of frost accumulation in the evaporator is not considered in this research, as a comparable heat pump operating on its own would suffer from identical problems. It is also assumed that the working fluid enters the compressor as a saturated vapour, and that the temperature at this point is 5°C below the temperature of the ambient air to facilitate heat transfer.

$$T_{1,HP} = T_{ambient} - 5 \tag{1}$$

$$P_{1,HP} = P_{sat}\ at\ T_{1,HP} \tag{2}$$

The pressure $P_{2,HP}$ was calculated by REFPROP so as to give a condensing temperature of 50°C. This was judged to be safely above the temperature of the water flow (see Figure 2). The isentropic efficiency of the compressor $\eta_{compressor,HP}$ was taken to be 70%. Assuming isentropic compression, $S_{2,HP}=S_{1,HP}$, which allows $H_{2s,HP}$ to be calculated using REFPROP, and the true value $H_{2,HP}$ to be calculated according to the equation

$$\eta_{compressor} = \frac{(h_{2s,HP} - h_{1,HP})}{(h_{2,HP} - h_{1,HP})} \quad (3)$$

The fluid is assumed to exit the condenser at state 3, a saturated liquid. Knowing this, and that the pressure $P_{3,HP} = P_{2,HP}$, $T_{3,HP}$ can easily be calculated using REFPROP.

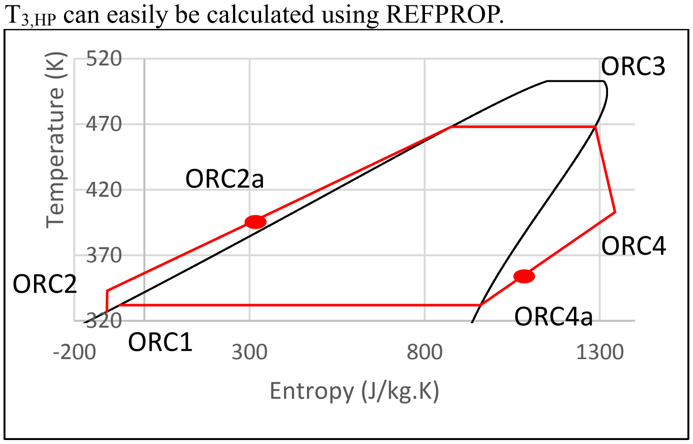

Figure 4 Diagram of ORC, showing numbering convention

Assuming an isenthalpic throttling process, $h_{4,HP}=h_{3,HP}$, and $P_{4,HP}=P_{1,HP}$, due to the assumption of zero pressure losses in heat exchangers. This allows the performance of the heat pump to be determined according to the following parameters

$$\Delta h_{evaporator,HP} = h_{1,HP} - h_{4,HP} \quad (4)$$

$$\Delta h_{condenser,HP} = h_{2,HP} - h_{3,HP} \quad (5)$$

$$\Delta h_{compressor,HP} = h_{2,HP} - h_{1,HP} \quad (6)$$

$$COP_{HP} = \frac{\Delta h_{condenser,HP}}{\Delta h_{compressor,HP}} \quad (7)$$

The compressor power is taken to be 1kW to simply the simulation, so the flow rate of the system can be calculated as follows

$$\dot{m}_{HP} = \frac{1}{\Delta h_{compressor,HP}} \quad (8)$$

And the condenser heat transfer rate as

$$Q_{condenser,HP} = \dot{m}_{HP} * \Delta h_{condenser,HP} \quad (9)$$

The ORC is a regenerative cycle using hexane as the working fluid. The condenser pressure is determined by the desired water outlet temperature, in this case 55°C. A pinch point temperature difference is set as 5°C, which is consistent with literature [21]. This allows us to determine the condensation temperature $Tcond_{,ORC}$ as 60°C. REFPROP can then be used to find the saturation pressure at this temperature, giving the condenser pressure $P_{cond,ORC}$. The fluid must be subcooled at the pump inlet to ensure no cavitation occurs, which gives

$$P_{1,ORC} = P_{cond,ORC} \quad (10)$$

$$T_{1,ORC} = T_{cond,ORC} - 5 \quad (11)$$

Knowing these two values, the entropy and enthalpy at the state 1 can be calculated using REFPROP.

The expander inlet temperature $T_{3,ORC}$ was fixed at 200°C. To avoid superheat at the expander inlet, which reduces cycle efficiency, the evaporator pressure $P_{evap,ORC}$ was set to the saturation pressure at this temperature. This could be calculated using REFPROP. Knowing $T_{3,ORC}$ and $P_{3,ORC}$, $h_{3,ORC}$ and $S_{3,ORC}$ could also be calculated.

The pump outlet pressure is equal to the evaporator pressure, and the expander outlet pressure is equal to the condenser pressure, so

$$P_{2,ORC} = P_{evap,ORC} \quad (12)$$

$$P_{4,ORC} = P_{cond,ORC} \quad (13)$$

The pump and the expander were taken to have isentropic efficiencies of 90% and 70%, respectively. If isentropic pumping and expansion are assumed, $S_{2,ORC}=S_{1,ORC}$ and $S_{4,ORC}=S_{3,ORC}$, allowing, $h_{2s,ORC}$ and $h_{4s,ORC}$ to be obtained from REFPROP, and used to calculate the actual values, using the equations.

$$\eta_{pump,ORC} = \frac{(h_{2s,ORC} - h_{1,ORC})}{(h_{2,ORC} - h_{1,ORC})} \quad (14)$$

$$\eta_{expander,ORC} = \frac{(h_{3,ORC} - h_{4,ORC})}{(h_{3,ORC} - h_{4s,ORC})} \quad (15)$$

Once these are known, $T_{2,ORC}$ and $T_{4,ORC}$ can be calculated using REFPROP.

The amount of heat transferred in the regenerator was calculated as follows. Initially, it assumed zero heat transfer, which gave a pinch point temperature difference of $T_{4,ORC}$-$T_{1,ORC}$. The program then increased the amount of heat transfer, which introduced an overlap between hot and cold lines, closing the pinch point. It monitored the temperature difference between the hot and cold flow at 100 different points until the minimum temperature difference reached the pinch point value of 5K. The value of heat transfer that gave this pinch point value could then be designated as $\Delta h_{regenerator}$, and the conditions at the regenerator inlet calculated as follows.

$$P_{2a,ORC} = P_{evap,ORC} \tag{16}$$

$$h_{2a,ORC} = h_{2,ORC} + \Delta h_{regenerator} \tag{17}$$

$$P_{4a,ORC} = P_{cond,ORC} \tag{18}$$

$$h_{4a,ORC} = h_{4,ORC} - \Delta h_{regenerator} \tag{19}$$

Similar with the analysis of heat pump, this allows several key parameters of the cycle to be determined.

$$\Delta h_{evaporatorORC} = h_{3ORC} - h_{2aORC} \tag{20}$$

$$\Delta h_{condenser,ORC} = h_{4aORC} - h_{1ORC} \tag{21}$$

$$\Delta h_{pump,ORC} = h_{2ORC} - h_{1ORC} \tag{22}$$

$$\eta_{ORC} = \frac{\Delta h_{expander,ORC} - \Delta h_{pump,ORC}}{\Delta h_{evaporator,ORC}} \tag{23}$$

Because of the direct linkage between the ORC expander and the Heat Pump compressor, the expander power must be the same 1kW as the compressor power, neglecting any mechanical losses, so

$$\dot{m}_{ORC} = \frac{1}{\Delta h_{expander,ORC}} \tag{24}$$

This means that the evaporator and condenser loadings can also be calculated.

$$Q_{condenser,ORC} = \dot{m}_{ORC} * \Delta h_{condenser,ORC} \tag{25}$$

$$Q_{evaporator,ORC} = \dot{m}_{ORC} * \Delta h_{evaporator,ORC} \tag{26}$$

The cold water stream was then analysed, and the data plotted in Figure 5. The temperature rise of the water can be given by

$$Q_{total} = \dot{m}_{water} * C_{p,water} * \Delta T. \tag{27}$$

Knowing that Q_{total} is the sum of the heat rejected in the condensers of the heat pump and ORC, and that ΔT is the initial temperature of the water subtracted from the final temperature, both of which are known, the required flow rate of water can be calculated.

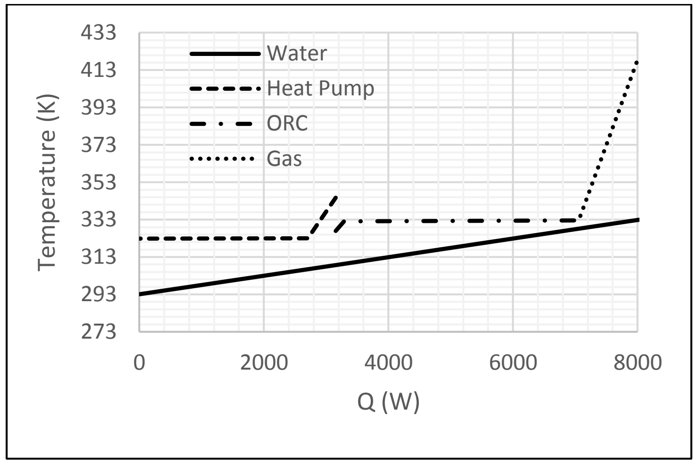

Figure 5 Heating of the water by the heat pump, ORC and gas burner exhaust

For the gas burner, the mixture of gases was determined assuming lean-burning methane. Methane reacts with oxygen according to the equation.

$$CH_4 + O_2 \rightarrow CO_2 + 2H_2O \tag{28}$$

This means that the combustion products will be 33% CO_2, 67% water vapour by volume, or 55% CO_2 by mass. The actual thermal fluid on the hot side of the heat exchanger will also have a proportion of air, depending on how lean the combustion process is. The amount of air in the final thermal fluid can be calculated from its temperature according to the formula

$$T_{1,gas} = \left(T_{flame} * M_{flame}\right) + \left(T_{air} * M_{air}\right) \tag{29}$$

Where M represents the mass fraction of a component, T_{flame} is the flame temperature, taken to be 2000°C, and T_{air} is the ambient temperature of the air. By trial and error, a thermal fluid temperature of 600°C was found to transfer 90% of its energy to the ORC and water, which is close to the claimed thermal efficiency of commercially available condensing boilers, so this value was taken for the model. This implies an air to fuel ratio of approximately 7:3.

The heat lost by the gas is assumed to all be transferred to the ORC and water in the post-heater, as shown in

Figure 6. The pinch point occurs at the entrance of the post-heater, at which point the water is at 55°C, giving the thermal fluid a temperature of 60°C. The enthalpy at the inlet and outlet can then be calculated using REFPROP, and the formula below used to calculate the necessary mass flow rate of the gas

$$\dot{m}_{gas} = \frac{Q_{gas}}{\Delta h_{gas}} \tag{30}$$

With all of this information, the fuel to useable heat efficiency of the cycle can be calculated.

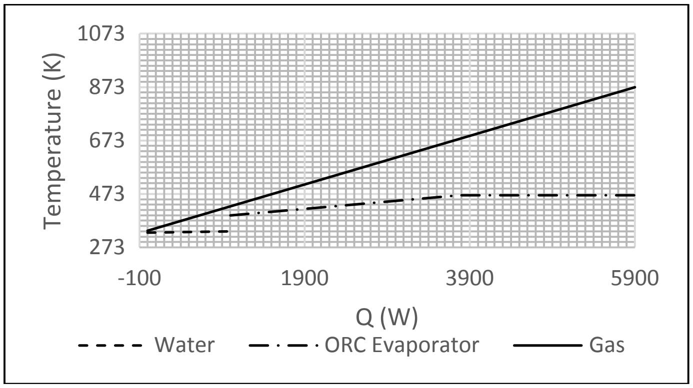

Figure 6 T-Q diagram of the gas flow, post heater and ORC evaporator

RESULTS

This paper examined the effect on the cycle of varying ambient temperature on the combined heat pump/ORC system. A typical breakdown of the energy transferred is shown in Figure 7. It can be seen that the heat pump and the ORC have almost equal shares, while the post-heater's share is considerably smaller. This is primarily down to the setup of the cycle, with the heat pump's COP of ~5 and the ORC's efficiency of ~20% being very similar by coincidence. The exact ratio of heat transferred in each component is an area for further optimisation.

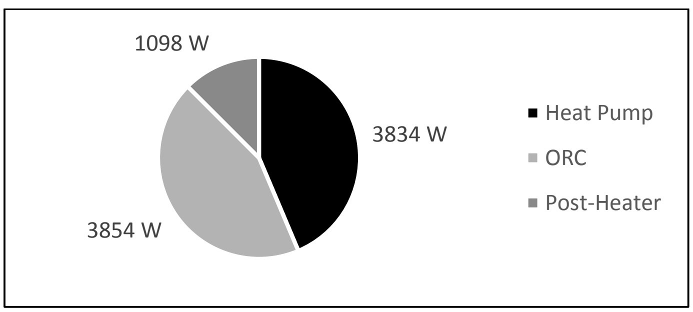

Figure 7 Pie chart of energy sources used to heat the water for an ambient temperature of 5°C

Figure 8 shows how the fuel to heat efficiency of the cycle varies with ambient temperature. It can be seen that increasing temperature causes an increase in this efficiency, from 136% at an ambient temperature of -5°C, to 164% at an ambient temperature of 15°C. The increased ambient air temperature allows the evaporator temperature to be raised, and reduces the necessary compressor work to raise the working fluid to the condenser temperature, thereby tending to increase the coefficient of performance of the heat pump.

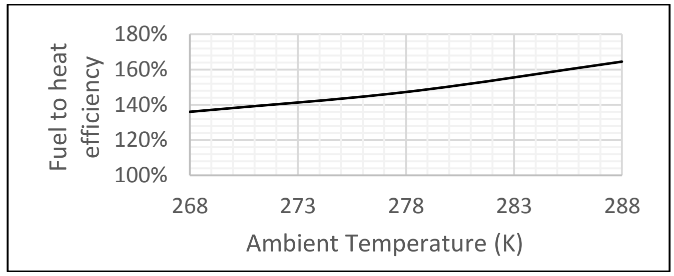

Figure 8 Fuel-to-heat efficiency against ambient temperature

This is borne out by the data plotted in Figure 9. The coefficient of performance of the heat pump increases steadily with increasing ambient air temperature.

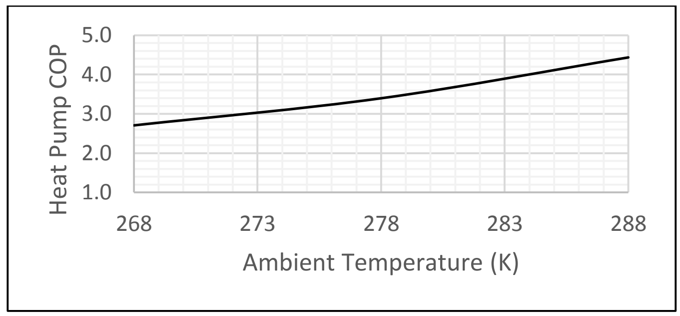

Figure 9 Variation in heat pump COP with varying ambient temperature

The ORC efficiency remained constant over the entire range of ambient temperatures, as the condensing temperature is high enough to be unaffected by variations in the water flow.

The energy transferred into the cycle from the hot gas increased, as the increased COP of the heat pump resulted in more heat being transferred from the ambient air to the water. To maintain the same temperature rise with this increase in energy flux, the mass flow rate of water needed to be increased as shown in Figure 10. The increase in the heat flux from the gas can be seen clearly, as can the corresponding increase in hot water flow.

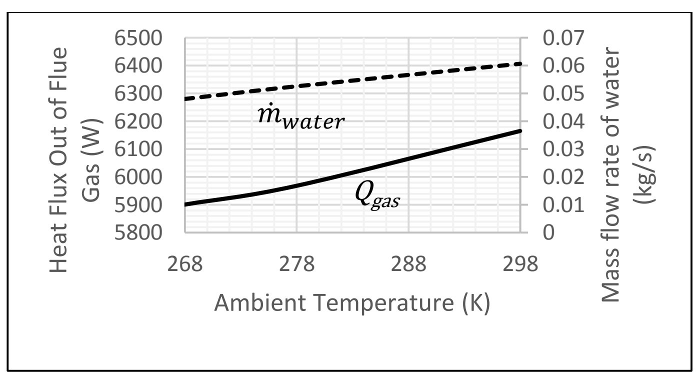

Figure 10 Heat flux from hot gas against ambient temperature

The reason for the observed increase in the heat flux from the flue gas is shown in Figure 11, which plots the heat flux in the post-heater, which provides the final stage of heating from 55°C to 60°C. The increased COP of the heat pump increases the amount of energy available to heat the water from its initial inlet condition to 55°C, allowing a higher water flow rate. This increased flow rate does, however, necessitate a greater amount of heat in the post-heater to increase the temperature from 55°C to 60°C.

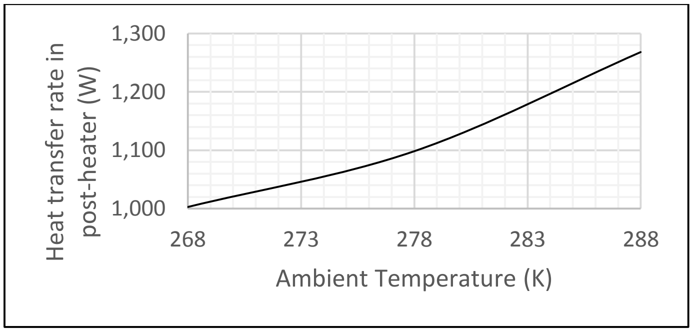

Figure 11 Post-heater heat transfer against ambient temperature

CONCLUSIONS

The combined heat pump/ORC system considered in this paper shows a strong dependency on the ambient temperature of air outside the system. This is to be expected, as the heat pump is responsible for almost half of the energy transferred to the water, and it uses the ambient air as its heat source, making its coefficient of performance highly sensitive to variations in ambient air temperature.

The overall fuel-to-heat efficiency of the cycle varies from 136% when the ambient temperature is -5°C to 164% when the ambient temperature is 15°C. This compares very favourably with competing heat generation technologies, such as electric heaters, condensing boilers, heat pumps run from the national grid, and adsorption heat pumps. The proposed cycle benefits from the fact that the higher-temperature heating is done in the post-heater and the ORC condenser, allowing the heat pump to run at a higher coefficient of performance than if it were required to take on the entire heating process itself.

These results show that the proposed combined cycle is technically competitive with existing technology, and an economic comparison of such cycles is an interesting area for further research, along with the optimization of the cycle parameters to achieve its maximum efficiency.

ACKNOWLEDGEMENT

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NOMENCLATURE

CHP		Combined Heat and Power
COP		Coefficient of Performance
HP		Heat Pump
ORC		Organic Rankine Cycle
T	[K]	Temperature
P	[bar]	Pressure
h	[J/kg]	Enthalpy
S	[J/kg.K]	Entropy
Q	[W]	Thermal Energy Transfer
W	[W]	Mechanical Power
$\dot{m}$	[kg/s]	Mass flow rate
Special characters		
η	[%]	First Law Efficiency
Subscripts		
1,HP		Heat Pump Compressor Inlet
2,HP		Heat Pump Condenser Inlet
3,HP		Heat Pump Expansion Valve Inlet
4,HP		Heat Pump Evaporator Inlet
1,ORC		ORC Pump Inlet
2,ORC		ORC Regenerator Inlet, Cold
2a,ORC		ORC Evaporator Inlet
3,ORC		ORC Expander Inlet
4,ORC		ORC Regenerator Inlet, Hot
4a,ORC		ORC Condenser Inlet
Flame		Relating to gas in combustion chamber
Air		Relating to ambient air
Evap		Evaporator
Cond		Condenser
Sat		Saturation
S		Isentropic

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