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Combined ORC-HP thermodynamic cycles for DC cooling and waste heat recovery for central heating

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Abstract

Data center is an essential part of modern life that is predicted to increase both in capacity and size over the coming years. This sector consumes a significant proportion of electricity for its operation and for the essential cooling facilities. This consumption is expected to rise with increasing demands which can result in more CO_2 emission. Waste heat is an inevitable by-product of DC operation with a potential of being sustainable, low cost and environmentally friendly heat source. In this paper, we proposed a novel system that integrates three thermodynamic cycles including an ORC, a HP and a Gas burner. The aim of this system is to provide cooling for the DC as well as to utilize the rejected heat to supply hot water for central heating. The results show that this system can maintain the indoor room temperature between 18-25 °C by absorbing 12 kW of heat to increase water temperature from 50 to 80 °C. In addition, the system can achieve an overall fuel-to-heat efficiency of 141.8%. Therefore, utilizing such system can have a great potential of improving DC performance as well as providing usable energy by waste heat recovery.

Keywords: Heat pump cycle; Organic Rankine cyelc, Combined cycle; data centre heat recovery.

1. Introduction

Data centres is a rapidly growing sector and is one of the main consumers of electricity. Worldwide, it is estimated that these centres utilize 1.2 to 1.5 % of the total electricity generated [1]. Such consumption is expected to grow by approximately 20% per year. Electricity is mainly used to power the IT servers and for cooling of these facilities due to high heat generated as a by-product. Cooling process can consume around 40% of the electricity supplied [2]. DC

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waste heat has the potential to be recovered by heat recovery technologies such as Organic Rankine Cycle (ORC), heat pump (HP), combined heat and power cycle (CHP), absorption refrigeration, thermosyphon and the combination between them. Ebrahimi K. *et al* 2015 [3] have proposed a novel configuration to recover wasted heat rejected by servers to power an absorption refrigeration (AR) system to provide cooling for nearby server in the same data centre. They then carried out a thermodynamic and economic analysis of an ORC module for DC waste heat recovery for electricity production [4]. Zhang P. *et al* 2015 [5], investigated the performance and economic feasibility of a heat recovery system based on water cooled integrated air conditioner with thermosyphon. Another proposed system for reuse DC wasted heat to heat water for a nearby swimming pool [6]. Vapor compression heat pump system is proposed to upgrade the temperature of the wasted heat up to 70 ºC which could be convenient for district heating network [1]. Recently Liang Y. *et al* 2018 [7] have revealed the potential of integrating ORC, HP and gas burner for hot water supply. A mixture of fresh air and flue gas is used as a heat source for the HP evaporator. However, the system performance fluctuates with ambient temperature variation. It is reported that the temperature of the DC wasted heat is higher than any other heat source temperature used for modern heat pump systems such as air or ground sources [1]. In this study, similar combined system to Liang Y. *et al* 2018 [7] model has been proposed for DC cooling with waste heat recovery for central heating.

2. Thermodynamic concept and mathematical model

In the proposed system, as shown in Figure (1), waste heat from DC is recovered by HP evaporator to provide cooling effect. A gas driven ORC cycle provides the required mechanical work to run the HP compressor. Waste heat rejected by both cycle condensers is used to heat water for central heating. R134a and R245fa are selected as refrigerant for HP and ORC cycles, respectively. In addition, Methane has been used as a fuel for the gas burner.

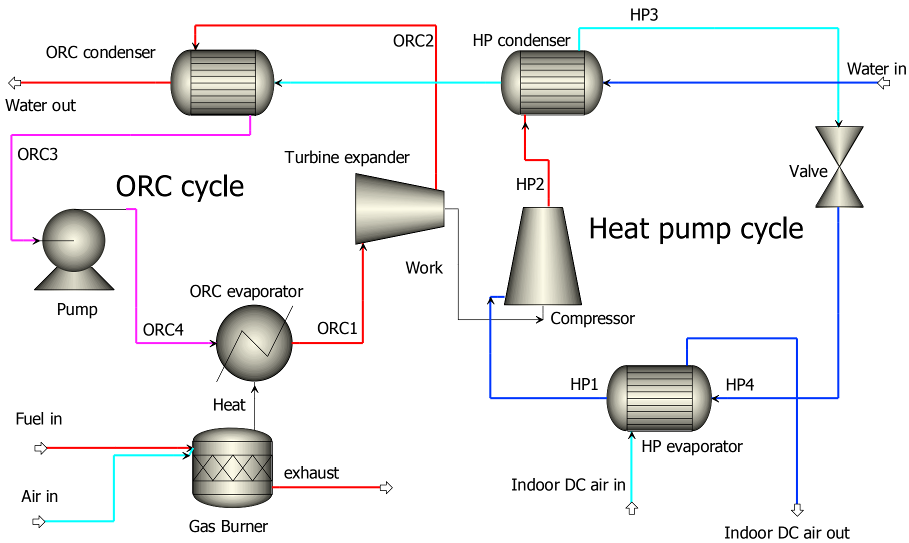

Figure 1 Combined cycle configuration

2.1 Assumptions

1. The heat pump evaporator is proposed to absorb 12 kW wasted thermal energy from the IT equipment of the DC and maintain indoor air temperature between 18-25ºC [2].
2. Evaporator temperature is estimated by maintaining 5 ºC pinch point temperature difference with the outlet air temperature and by assuming the refrigerant is 100% vapor at the evaporator exit.
3. The water temperature is heated from 50 to 80 ºC for central heating application.
4. Isenthalpic expansion process is assumed in the expansion valve ($h_{HP4} = h_{HP3}$). The isentropic efficiency of HP compressor and ORC turbine are assumed to be 70%. Heat and pressure loss is neglected.

5. The heating value of Methane is assumed as 55.5 kJ/kg, with combustion efficiency of 100%.

2.2 Modelling

In-house MATLAB code linked with REFPROP software has been modified to evaluate the energy balance across the combined cycle components. In addition, the steady state results have been converged by ASPEN plus. Cycles efficiencies are calculated using the following equations:

$$COP_c = \frac{Q_{evap}}{W_{comp}} \tag{1}$$

$$COP_h = \frac{Q_{cond}}{W_{comp}} \tag{2}$$

$$\gamma_{ORC} = \frac{\left(W_{turbine_{ORC}} - W_{pump\_ORC}\right)}{Q_{evap_{ORC}}} \tag{3}$$

The total heat released from the gas burner $\dot{Q}_g$ is calculated as

$$\dot{Q}_g = \dot{m}_{fuel} \times \dot{Q}_{HV} \times \eta_{comb} \quad . \tag{4}$$

Fuel to heat efficiency is the ratio of total heat added to water plus heat removed from DC over $\dot{Q}_g$:

$$\eta_{fuel-to-heat} = \frac{\sum \dot{Q}_w}{\dot{Q}_g} = \frac{\dot{Q}_{HP,cond} + \dot{Q}_{ORC,cond} + \dot{Q}_{HP,evap}}{\dot{Q}_g} \tag{5}$$

3. Results and discussion

Figure (2 a and b) shows variation of evaporator and condenser thermal capacities with refrigerant mass flow rate at different condensation temperatures for the HP cycle. For the selected range of condensation temperature, evaporator and condenser cooling and heating duties increases with the rise in R134a mass flow. For each mass flow, increasing condensation temperature will reduce both thermal capacities. In general, thermal capacity is a function of mass flow and enthalpy difference (Delta h). For the evaporator, enthalpy at inlet will increase with the rise in condensation temperature whereas enthalpy at evaporator exit is assumed constant. Hence, evaporator capacity deceases as Delta h increases. For the condoner, both enthalpies at inlet and exit increases with the rise in condoner pressure, however, the increment in the enthalpy at condenser inlet is higher than that at exit resulting in a reduction in Delta h.

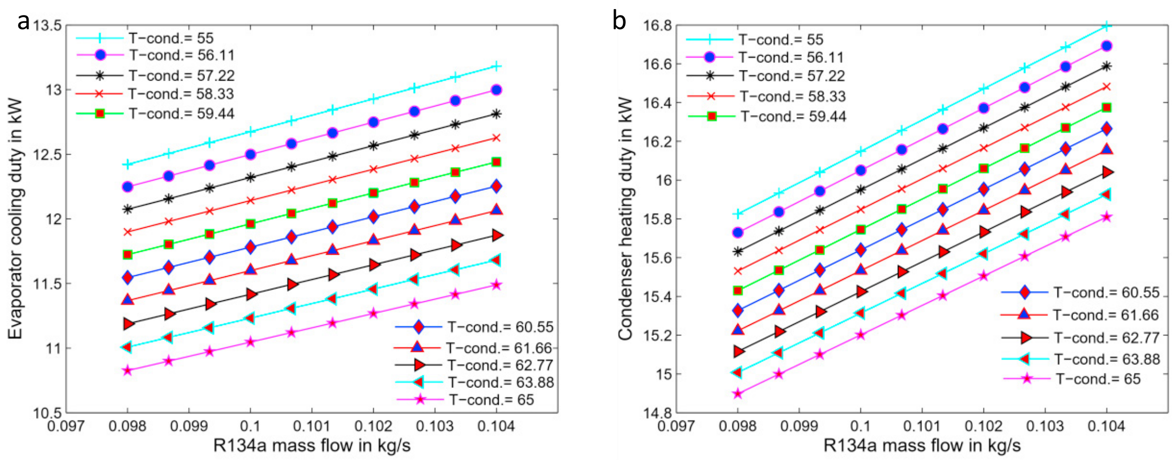

Figure (2) variation in R134a mass and condensation temperature on HP evaporator and condenser capacities.

Figure (3) shows the variation of the compressor network against variation in the condenser pressure and working fluid mass flow. Increasing R134a mass flow and condenser pressure will result in higher compressor network as a result of higher enthalpy at compressor exit.

Figure (4) shows the behaviour of the heating and cooling coefficient of performance when the discharged pressure increases at a constant mass flow. It shows that the efficiency decreases with increasing the condenser pressure. This happened because increasing condenser pressure results in a higher compressor work and a lower heating capacity for the evaporator and condenser as explained in the results above.

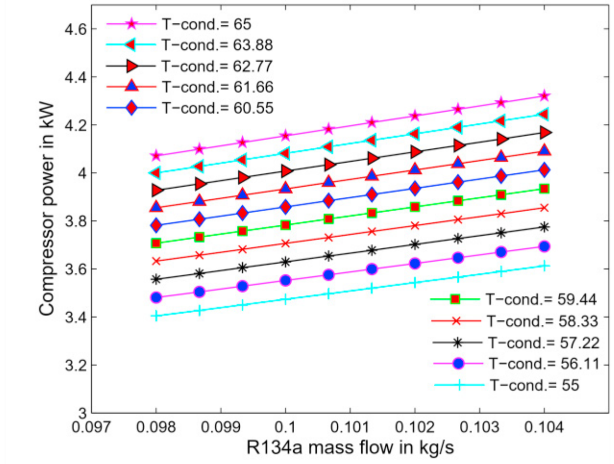

Figure (3) variation of R134a mass and condensation pressure on compressor network.

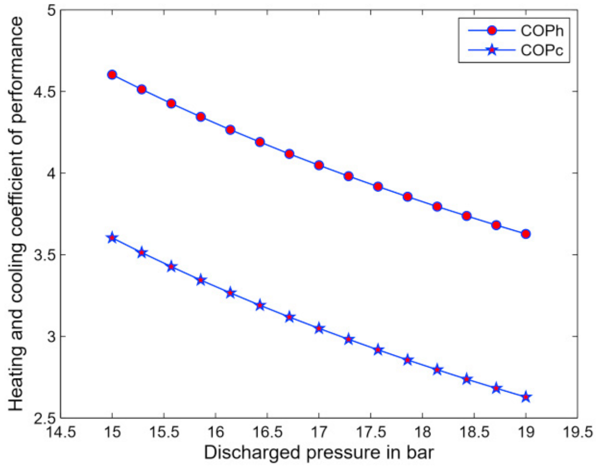

Figure (4) effects of discharged pressure on COP_c and COP_h.

Based on the above results and for the current case study evaluation, the required values of mass flow rate, condenser pressure and temperature for the desired evaporator cooling capacity (12 kW) have been identified. These values are 0.10193 kg/s, 17 bar and 60.4 C respectively. In addition, the required compressor work is 3.937 kW.

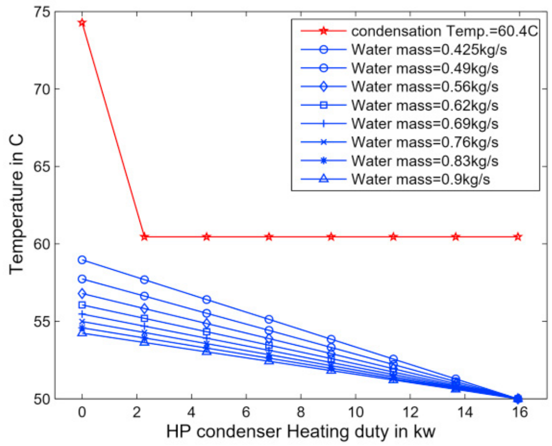

Figure (5) variations of water mass flow in the HP condenser under constant condenser pressure.

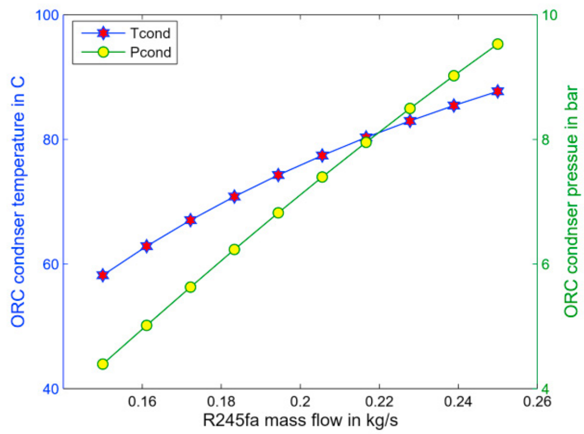

Figure (6) optimization of ORC condenser pressure and temperature.

Figure (5) demonstrates the iteration of the water mass flow at the selected condensation temperature and R134a mass flow. The optimization of this process is based on securing minimum pinch point difference of 3 °C between the two streams. The results show that water mass flow of 0.425 kg/s has satisfied the optimization condition. It also shows that the return water is heated to around 59 °C.

For the ORC cycle, to achieve high thermal efficiency, the evaporator pressure is set to a value around the working fluid critical pressure (36.5 bar). In addition, a temperature of 5 °C have been added to superheat the R245fa to 159 °C before entering the turbine. Some of the working conditions required for the ORC cycle have been recognized from

the output results of the HP cycle such as turbine work, water temperature and mass flow. By adapting these parameters as well as the assumed final water temperature (80 ºC), the condenser heating duty can be directly calculated. To identify the optimum ORC condenser pressure, R245fa mass flow have been iterated at constant ORC evaporator pressure as shown in figure (6). At mass flow of 0.219 kg/s the condensation pressure and temperature are 8 bar and 80.5 ºC, respectively.
Figure (7) shows the TQ curve at variable condensation pressure. From the figure, the optimum condensation pressure has secured the minimum pinch point temperature (3 ºC) between the two flows inside the condenser.

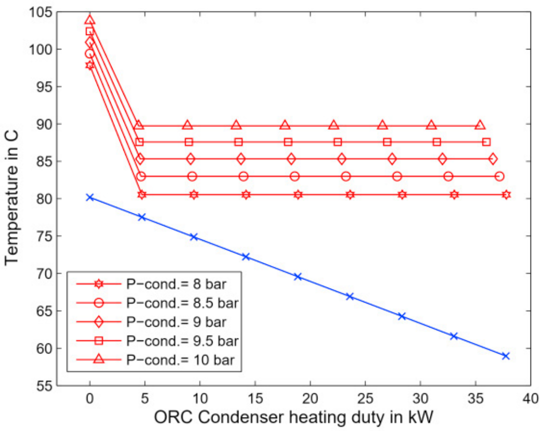

Figure (7) variations of ORC condensation pressure at constant water mass flow.

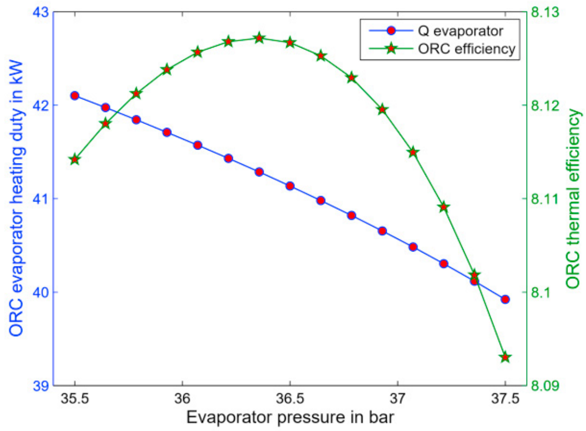

Figure (8) changes in ORC evaporator pressure against capacity and cycle thermal efficiency.

Figure (8) shows effects of ORC evaporator pressure at constant mass flow on the evaporator heating duty and thermal efficiency. The results verified that at evaporator pressure of 36.5 bar, thermal efficiency has reached the maximum value. At this pressure, the evaporator heating duty reached a value of 41 kW.

The steady state results from MATLAB code has been verified by ASPEN plus software, as shown in Tables 1-3.

Table (1) Heat pump and ORC cycle design parameters.

Parameters	Heat pump cycle	ORC cycle
Condenser heating duty, kW	15.936	37.793
Water temperature leaving the cycle, ºC	58.9	80
Evaporator duty, kW	12	41.136
Condensation temperature, ºC	60.4	80.5
Evaporation temperature, ºC	13	159
Condenser pressure, bar	17	8
Evaporator pressure, bar	4.5	36.5
Power produced by the expander of ORC, kW	3.937	
The work of liquid pump, kW		0.594

Table 2 Cycles' performance

Parameters	Heat pump cycle
Heat pump COPc	3
Heat pump COPh	4
ORC thermal efficiency	8.12%
Overall fuel to heat efficiency of whole system	141.8%

Table 3 Gas burner design parameters

Parameters	Heat pump cycle
Mass flow rate of methane, kg/s	0.000835125
Air mass flow, kg/s	0.015867375
Air to fuel ratio	19
Heat production, kW	41.136
Exhaust outlet temperature, ^{o}C	60

Conclusion

Steady state thermodynamic evaluation has been carried out on the combined vapor compression heat pump and ORC power generator cycle to produce cooling and heating effect simultaneously for a data centre. The proposed system is designed to maintain the DC room temperature between 18-25^{o}C and to pump 12 kW of wasted heat to the water. The water is firstly heated in the HP condenser then reaches its designed temperature value at the final stage (ORC condenser). The results show that the HP when work in a steady heat source load and temperature can achieve constant higher system performance with a COPc and COPh of 3 and 4 respectively. In addition, the ORC cycle has achieved a thermal efficiency of 8.12%. Overall, the combined system has achieved high fuel to heat efficiency of around 141.8 % due to the utilization of DC wasted heat by both HP and ORC cycles to heat the water from 50 to 80 ^{o}C.

References

[1] Davies G. *et al*. Using data centres for combined heating and cooling: An investigation for London. Applied Thermal Engineering 94 (2016) 296–304.

[2] Zhou F. *et al*. Development and analysis of a pump-driven loop heat pipe unit for cooling a small data center. Applied Thermal Engineering 124 (2017) 1169–1175.

[3] Ebrahimi K. *et al*. Thermo-economic analysis of steady state waste heat recovery in data centers using absorption refrigeration. Applied Energy 139 (2015) 384–397.

[4] Ebrahimi K. *et al*. The viability of ultra low temperature waste heat recovery using organic Rankine cycle in dual loop data center applications. Applied Thermal Engineering 126 (2017) 393–406.

[5] Zhang P. *et al*. Heat recovery from Internet data centers for space heating based on an integrated air conditioner with thermosyphon. Renewable Energy 80 (2015) 396-406.

[6] Oró E. et al. Design and economic analysis of liquid cooled data centres for waste heat recovery: A case study for an indoor swimming pool. Sustainable Cities and Society 36 (2018) 185–203.

[7] Liang Y. et al. Investigation of a gas-fuelled water heater based on combined power and heat pump cycles. Applied Energy 212 (2018) 1476–1488.

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