Performance analysis of diesel engine heat pump incorporated with heat recovery

N.N. Shah[1,*], M.J. Huang[*], N.J. Hewitt[*]

**Centre for Sustainable Technologies, Ulster University, Newtownabbey, Co. Antrim, BT37 0QB, UK*

Abstracts

This paper presents experimental study of diesel engine heat pump (DEHP) system to find potential as retrofit technology in off-gas or weak electricity network area to replace existing gas/oil/electric heating system in domestic sector. Test set-up of diesel engine driven water-to-water heat pump system was built which included heat recovery arrangement from the engine coolant & exhaust gas. The system was designed to meet typical house heating demand in Northern Ireland. Performance of DEHP was evaluated to meet house-heating demand at different flow temperature (35, 45, 55 & 65 °C), a typical requirement of underfloor space heating, medium/high temperature radiators and domestic hot water. The performance was evaluated against four-evaporator water inlet temperature (0, 5, 10 & 15 °C) and at three different engine speed 1600, 2000 & 2400 rpm. Experiment results were analysed in terms of heating/cooling capacity, heat recovery, total heat output, primary energy ratio (PER), isentropic efficiency etc. Test results showed that DEHP is able to meet house-heating demand with help of heat recovery with reduced system size. Heat recovery contributed in a range of 22 to 39% in total heat output. It is possible to achieve high flow temperature in a range of 74 °C with help of heat recovery. Overall system PER varied in a range of 0.93 to 1.33. Speed increment and flow temperature has significant impact on heat recovery, total heat output and PER. A case scenario with different flow temperature to match house-heating demand has been presented to show working potential with different heat distribution system. In addition, DEHP shows good potential to save primary energy consumption and CO_2 emissions, a helpful technology to achieve national emission reduction target.

Key words: Heat pump, DEHP, Retrofit, Water source, Heat recovery, Diesel engine

Highlights

- Diesel engine heat pump with heat recovery
- Water-to-water source heat pump based on R134a
- Possible retrofit application in off-gas or weak electricity network area
- Potential to diversify use of fossil fuel, primary energy and CO_2 emission savings

[1] Corresponding author: Tel: +44 (0) 2890366122; Fax: +44 (0) 2890368239.
Email: n.shah@ulster.ac.uk (N.N. Shah)

1 Introduction

Future challenges of secure fuel supply, fuel prices, climate change and CO_2 emissions requires collective efforts towards use of more efficient technology along with increased share of renewable energy. In the UK, domestic sector is responsible for 64.2 $MtCO_2e$ emissions by consuming 38.2 Mtoe of energy in 2014 [1] [2]. The domestic heat demand exists mainly due to space heating and hot water production mostly met by natural gas. Hence, there is a great potential to diversify use of fossil fuels, improve energy efficiency, reduce primary energy consumption and emissions in domestic sector.

Vapour compression heat pumps are such mature and efficient technology to provide space heating and hot water for various applications. Different application of heat pumps with other renewable technology and energy storage has shown promising results [3] [4]. However, the most heat pumps are driven by electric motor that uses electricity called electric heat pumps (EHPs). If major deployments of EHPs (10-20% penetration) were carried out then it would require attention to existing electricity distribution network [5] [6]. In addition, poor insulated UK housing stock and single-phase electricity supply adds additional cost to components/sizing of EHPs [7]. To take advantage of oil/gas boiler replacement in existing housing stocks, EHPs faces several challenges such as oversized radiator, flow temperature requirement (above 60˚C) etc. as a retrofit application [8] [9] [10].

Engine driven heat pumps (ENHPs) is such potential technology to overcome issues of EHPs due to waste heat recovery and speed modulation [11]. Extensive literature search on web of science gave output of around 110 articles from year 1980 to 2016 on engine heat pump. A literature search shows four focus area of ENHP articles: 1.) Application/experimental work: commercial, industrial, transport (e.g. green house heating, drying etc.); 2.) Theoretical studies: reviews, energy/exergy analysis etc.; 3.) Novel application: hybrid system, tri-generation, use with renewable technology or technology combination; 4.) Simulation/modelling, controller, control strategy for optimum operation, cost and emission savings.

For example, Hepbasli et al. (2009) [12] have presented a review on gas engine heat pump (GEHP) application in residential and industrial sector showing benefits of GEHP over conventional system. In addition, there are opportunities to use various fuel based on renewable sources such as biogas, biofuels etc. with use of waste heat from the engine [13] [14]. Additionally, many authors presented advantages of ENHPs over EHPs mainly due to waste heat recovery and engine speed modulation for various applications [15] [16] [17] [18] [19] [20]. There are various literatures available on ENHPs application for commercial and industrial applications [21]. Most literature focuses on system parameters influence mainly on primary energy ratio in heating, cooling and hot water application [22] [23] [24] [25]. Few other investigations focused on GEHP's simulation and/or experiment in heating or hot water application showing heat recovery contribution (30%) in total heat output [26] and 37% emission reduction compared to gas boiler [27]. Mostly, all investigations focused on gas engine and air-to-water or air-to-air system for heating/cooling/hot water application. There is very little investigation on ENHPs domestic application with water-to-water heat pump and/or use of diesel engine. Lian et al. (2005) showed benefits water-to-water based engine heat pump with reduced payback period [28] but still no application for domestic sector. Additionally, there is not any small engine heat pump system (e.g 10kW) commercially available in Europe as the main market players of engine heat pump system are from ASIA (mainly Japan) with capacity from 14 to 175 kW [29].

Hence, this work presents novel domestic application with <10kW size heat pump (water-to-water) driven by diesel engine. The performance analysis of system aims to address retrofit challenges in domestic sector. It provides information for various flow temperature required for conventional radiator (high temperature), medium temperature and underfloor heating system. The study on high temperature application (65˚C flow temperature) for retrofit application has been shown by author's previous work [30]. This paper presents detailed work related to flow temperature, engine speeds and their comparative analysis at respective evaporation temperature conditions.

2 DEHP test set-up development and test procedures

2.1 Design criteria

Diesel engine driven heat pump (DEHP) system component size, selection and design mainly depends on average/peak heating demand. It is also important to estimate heat recovery percentage that helps to reduce heat pump system components size and hence, the cost. The domestic heat demand mainly occurs due to space heating and hot water that varies with dwelling types, occupancy, season etc. For the test set-up development, heat demand (Figure 1) of a typical three bedroom 105m^2 test-houses in Carrickfergus, Northern Ireland was considered [31]. The house heating demand varies between 4.2 kW to 8.5 kW at air temperature of -10˚C to 20˚C. DEHP system was designed to meet 7.1 kW heat demand at 0˚C ambient temperature to provide flow temperature in a range 55˚C to 65˚C (above 60˚C to avoid legionella) with the help of heat recovery. Based on the heating demand other system components were designed, selected and assembled together.

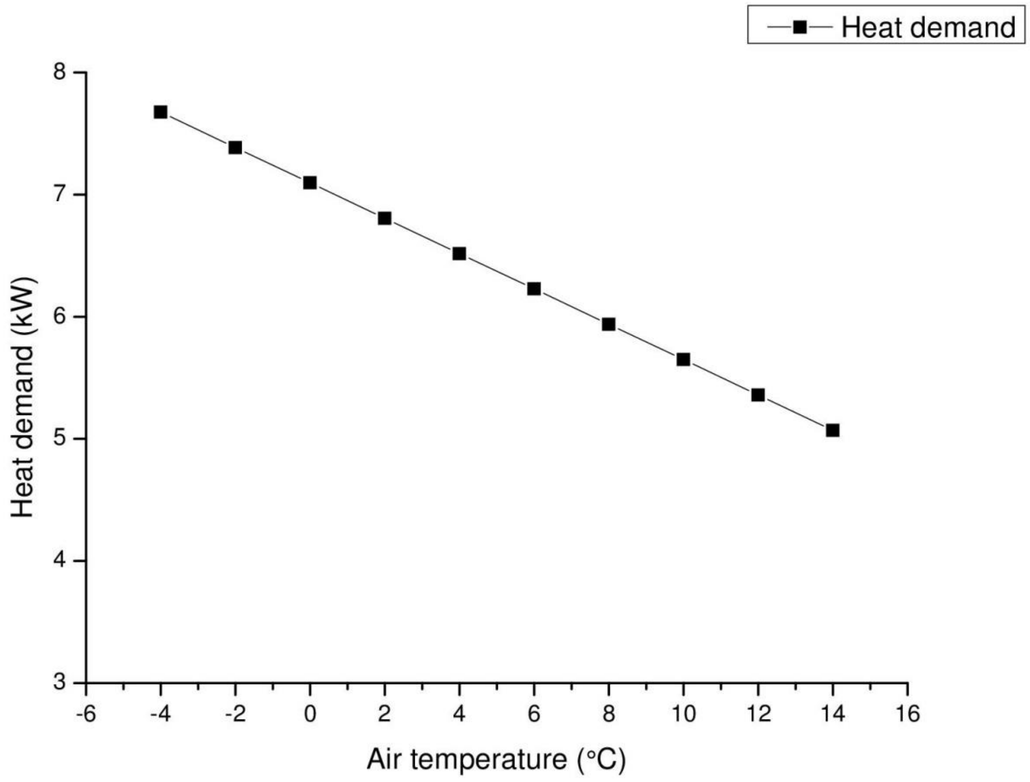

Figure 1 Typical house heating demand (incl. DHW demand) [31]

2.2 System components

DEHP system has two major components; the diesel engine and open reciprocating compressor. Both components were selected/checked simultaneously to match their torque and speed requirement. Based on initial analysis, commercially available diesel engine Kubota EB300-E [32] was selected to drive open reciprocating compressor with help of flexible coupling. R134a was used as a refrigerant due to higher temperature limit and lower torque requirement for the compressor/engine. The engine gives 4.14 kW continuous power at maximum speed of 3000 rpm whereas compressor has a speed range of 750 to 3000 rpm.

The engine came with thermosiphon water-cooled system accompanied by radiator fan. Therefore, the engine was modified to accommodate coolant heat recovery. A low temperature thermostatic valve, pump, and plate heat exchanger replaced fan & radiators arrangements. This enabled coolant heat recovery when temperature of coolant rises above 60 °C.

Heat pump circuit consisted a brazed plate heat exchanger as condenser/evaporator, thermostatic expansion valve (TEX), filter/drier and liquid receiver. TEX converts high pressure liquid into low-pressure vapour-liquid mixture at outlet whereas liquid receiver helps to store and provide refrigerant during load change and pump down conditions. Heat recovery circuit has plate heat recovery for coolant heat recovery, exhaust gas heat recovery heat exchanger (shell-tube type), pumps and three-way valve. Figure 2 presents schematics of DEHP set-up where it shows the arrangement of various components and instrumentations for DEHP system. Primary circuit represents refrigeration cycle and heat pump related components/instruments. In secondary circuit, water temperature at inlet of evaporator and outlet of condenser maintained constant with the help of PID controller. This PID controller operates motorised 3-way valve in third circuit. In this arrangement, heat gained from condenser side is given to evaporator side and excess heat is removed by fan coil to atmosphere. The main variables such as temperature, pressure, flow rate and speed were measured on heat pump side; engine side and heat recover side. All measured data were logged in data acquisition system. Measurement accuracy of temperature, pressure, flow rate and speed was in the range of ± 0.15 K, ± 1%, ± 1% and ±1% respectively. Data were logged every 15s and stored for data analysis purpose with data acquisition system.

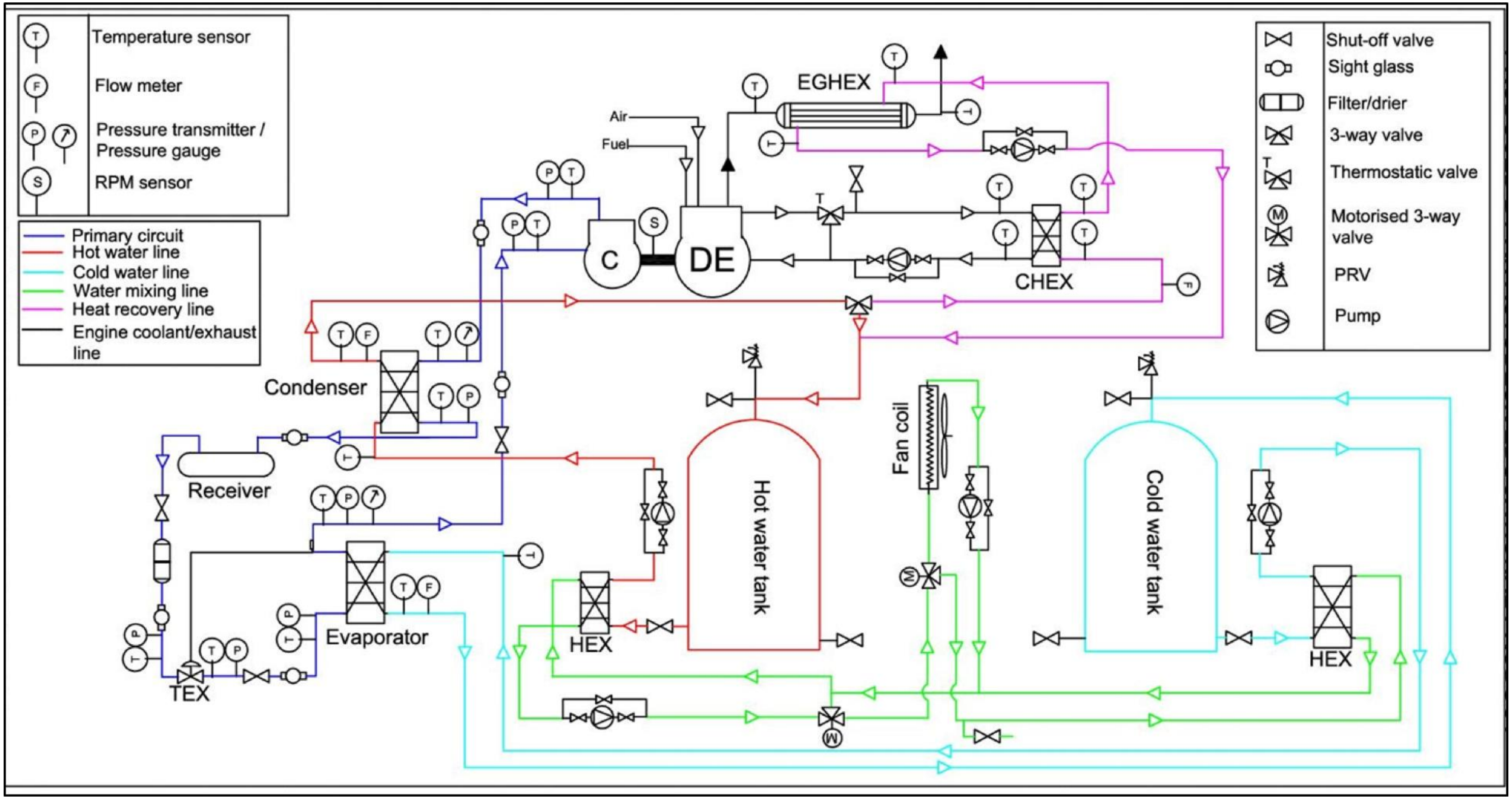

Figure 2 DEHP test setup with heat recovery arrangement [30]

2.3 Experimental methods and procedure

This work is aimed to understand the effect of various flow temperatures required for different heating system. DEHP experiments carried out at 35°C, 45°C, 55°C and 65°C flow temperature (condenser water outlet). At each flow temperature, DEHP system performance was measured against three different engine speeds (1600 rpm, 2000 rpm & 2400 rpm) and four evaporator water inlet temperature (0°C, 5°C, 10°C and 15°C). This evaporation water temperature ran also provided good information for ground source heat pump. As there is no current European test standard that covers water-to-water based DEHP system for small scale, British standard EN 14511 [33] was taken as a reference for DEHP testing.

As per standards, initial test were carried out at fixed engine speed of 1600 rpm to obtain desired flow rate and temperature conditions. Afterwards, flow rate was set constant while changing the speed and evaporation temperature for same flow temperature conditions. Similar steps were repeated for other flow temperature conditions.

The parameters such as pressure, temperature, flow rate and speed were monitored/logged and used to calculate performance of DEHP system using thermodynamic properties of refrigerant and heat transfer fluid. Following equations are used to calculate various performance parameters of DEHP system.

2.3.1 Condensing/heating capacity

Condensing capacity (Q_c) or heat output from heat pump mainly depends on mass flow rate of water (m_w) and temperature difference (outlet (t_{cout}) & inlet (t_{cin}) temperature) of water on secondary side of condenser. Primary side heat output by refrigerant can be given by mass flow rate refrigerant (m_{ref}) and enthalpy difference at inlet (h_{cin}) and outlet (h_{cout}) of condenser. Hence, this can be given as

$$Q_c = m_w\, C\, (t_{cout} - t_{cin}) = m_{ref}(h_{cin} - h_{cout}) \qquad 1$$

2.3.2 Cooling capacity

Cooling capacity (Q_e) is the heat gained by heat pump from water and delivered it to refrigerant. This can be calculated in a similar way to condenser for both primary and secondary side. On secondary side of evaporator, mixture of ethylene glycol and water was used instead of pure water. Hence, mass flow rate of heat transfer fluid (m_{hf}) and temperature difference across evaporator (t_{ein} –t_{eout}) can be given as:

$$Q_e = m_{hf}\, C\, (t_{ein} - t_{eout}) = m_{ref}(h_{eout} - h_{ein}) \qquad 2$$

2.3.3 Power consumption

For DEHP system, power consumption (P_c) was not measured directly due to vibration & safety issues. However, it was calculated indirectly from data obtained from experiments that gives good indication of power consumed by refrigeration cycle and helps to calculate COP too. Power consumption across compressor is dependent on mass flow rate of refrigerant and enthalpy difference at outlet (h_{pout}) and inlet (h_{pin}) of compressor. This can be written as:

$$P_c = m_{ref}\left(h_{pout} - h_{pin}\right) \quad 3$$

2.3.4 Coefficient of performance

For engine driven system, primary energy ratio is more important to have comparison with other technologies. However, for analysis purpose, COP of DEHP system has been calculated using condensing capacity and compressor power consumption that can be given as

$$COP = \frac{Q_c}{P_c} = \frac{m_{ref}\,(h_{cin} - h_{cout})}{m_{ref}\,(h_{pout} - h_{pin})} \quad 4$$

2.3.5 Isentropic efficiency of compressor

Isentropic efficiency (η_i) of compressor gives good indication on compressor performance and it can be obtained with the help of suction enthalpy at compressor (h_{pin}), actual discharge enthalpy of compressor (h_{pout}) and isentropic enthalpy at discharge (h_{pouti}). This can be calculated as:

$$\eta_i = \frac{h_{pouti} - h_{pin}}{h_{pout} - h_{pin}} \quad 5$$

2.3.6 Diesel fuel combustion heat

Diesel fuel is primary source of energy for DEHP system. Diesel burns inside the engine and generates the heat. The heat generated by the fuel (Q_f) can be calculated using HHV of diesel fuel and mass flow rate of diesel fuel (m_f) at the engine:

$$Q_f = m_f\, H.H.V. \quad 6$$

2.3.7 Heat recovery from exhaust gas and coolant

The exhaust gas heat was recovered using shell and tube heat exchanger where water passes through the shell and gas on tube side. Heat gained by water from exhaust gas (Q_{ex}) can be given by mass flow rate of water (m_w) and water temperature difference (t_{exout} –t_{exin}) across heat exchanger.

$$Q_{ex} = m_w\, C\, (t_{exout} - t_{exin}) \quad 7$$

Similarly, coolant heat was recovered using plate heat exchanger and heat recovered by water from coolant (Q_{cl}) can be given using water mass flow rate and water temperature different at outlet an inlet of heat exchanger(t_{clout} –t_{clin}).

$$Q_{cl} = m_w\, C\, (t_{clout} - t_{clin}) \quad 8$$

2.3.8 *Total heat output from DEHP system*

Total heat output (Q_t) from DEHP system includes condenser heat output, heat gained from exhaust gas and coolant. This can be given as

$$Q_t = Q_c + Q_{ex} + Q_{cl}$$

$$Q_t = m_w C(t_{cout} - t_{cin}) + m_w C(t_{exout} - t_{exin}) + m_w C(t_{clout} - t_{clin}) \quad 9$$

2.3.9 *Diesel engine efficiency*

Diesel engine efficiency (η_e) gives good indication for overall energy balance of diesel engine that includes losses from the engine and heat lost to the exhaust gas and coolant. Diesel engine efficiency can be calculated using power consumed at compressor and diesel fuel combustion heat. Therefore,

$$\eta_e = \frac{P_c}{Q_f} = \frac{m_{ref}(h_{pout} - h_{pin})}{m_f HHV} \quad 10$$

2.3.10 *Primary energy ratio*

Primary energy ratio (PER) gives true indication for DEHP system performance and used to compare performance of EHPs and conventional heating system. PER can be represented with the help of total heat output from DEHP to total heat supplied by diesel fuel. Therefore,

$$PER = \frac{Q_t}{Q_f} = \frac{Q_c + Q_{ex} + Q_{cl}}{m_f\, H.H.V.} \quad 11$$

Using equations 1 to 11, performance of DEHP system has been calculated for various conditions based on experimental data.

3 Results and discussion

DEHP system has three-sub system namely; heat pump side, engine side and heat recovery side. For ease of understanding, results and discussion are presented according to sub systems. The system was designed to meet house-heating demand by total heat output. Hence, whenever total heat output from DEHP system was higher than required house heat demand, test was terminated as such excess demand is not going to occur in real life conditions. For example, few tests for 35˚C flow temperature at 2000 rpm and 2400 rpm were not executed due to higher heat output than house heat demand. Hence, it is not shown in graph but just mentioned in text. However, performance was simulated at those conditions based on selection software along with other practical assumption based on experiment results.

3.1 Heat pump performance

This section includes performance parameters related to heat pump. Heat pump performance is mainly given by condensing capacity or heat output from the condenser during heating and hot water application whereas cooling capacity is a good indicator for cooling/cold water application. In addition, other important parameters such as discharge temperature, isentropic efficiency and COP have been discussed too.

3.1.1 Condensing or heating capacity

DEHP system heat output from condenser has been evaluated to provide flow temperature in a range of 35˚C to 65˚C at three different speeds and four evaporation temperatures. Figure 3 shows variation in condensing capacity with respect to evaporation temperature at engine speed of 1600 rpm, 2000 rpm and 2400 rpm. This graph does not shows variation in condensing capacity at 35˚C flow temperature due to excess heat output than house heating demand at higher speed (above 1600 rpm). Hence, at 35˚C flow temperature experiments were carried out only at 1600 rpm and it is not shown on graph to maintain uniformity.

Test results shows that condensing capacity varied between 3.64 kW to 9.75 kW during all test conditions. For 35˚C flow temperature, heat output remains between 4.61 kW to 8.67 kW at 1600 rpm. For all flow temperature conditions, heating capacity increases by 14% with each step speed increment (e.g. changing from 1600 rpm to 2000 or 2000 to 2400 rpm). Similarly, heating capacity increases by 22% with increasing evaporation temperature. However, heating capacity decreases by 10% with increasing flow temperature by 10K.

It is also evident that heat output from condenser only would not be able to meet house heat demand (7.1 kW at 0˚C) and heat pump would be undersized for given application. In such scenario, heat recovery form the engine is crucial to meet house heat demand at respective flow temperature. Additionally, experimental data were verified with compressor selection software for the same test conditions. The test data gave 17% lower heat output compared to software simulation mainly due to heat loss, pressure drop and condenser effectiveness.

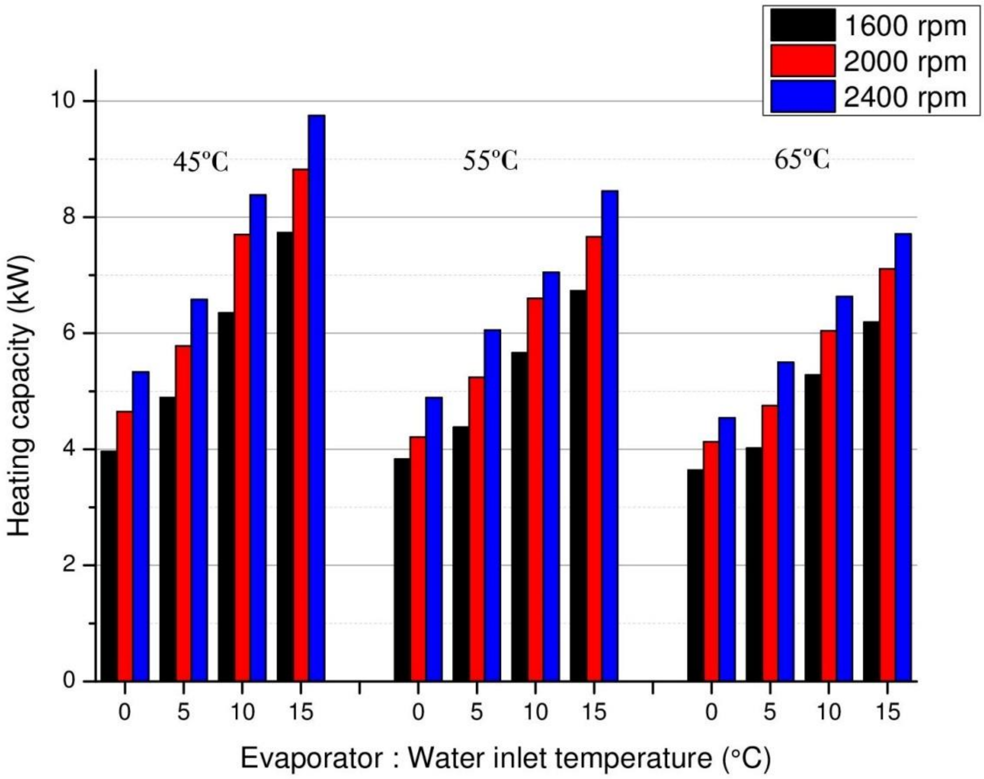

Figure 3 Heating (condensing) capacity variation with respect to flow temperature and engine speed

3.1.2 Cooling capacity

Cooling capacity is an important function where heat pumps are used for cooling or cold-water application. Cooling capacity is calculated using parameters on secondary side of evaporator. Analysis on primary (refrigerant) and secondary side of evaporator also helps to understand effectiveness of evaporator. Figure 4 shows variation in cooling capacity with respect to engine speed and evaporation temperature at various flow temperature conditions. Cooling capacity varies between 2 kW to 7 kW during entire test conditions. For 35˚C flow temperature, cooling capacity varies between 3.3 kW to 7 kW at 1600 rpm. Cooling capacity increases by 11% with increasing speed by each step whereas cooling capacity increases by 26% with increasing evaporation temperature. However, cooling capacity decreases by 15% increasing flow temperature by 10K. Therefore, it is beneficial to run system at higher engine speed, evaporation temperature and low flow temperature to get higher cooling and heating capacity due to thermodynamic nature of the refrigeration cycle.

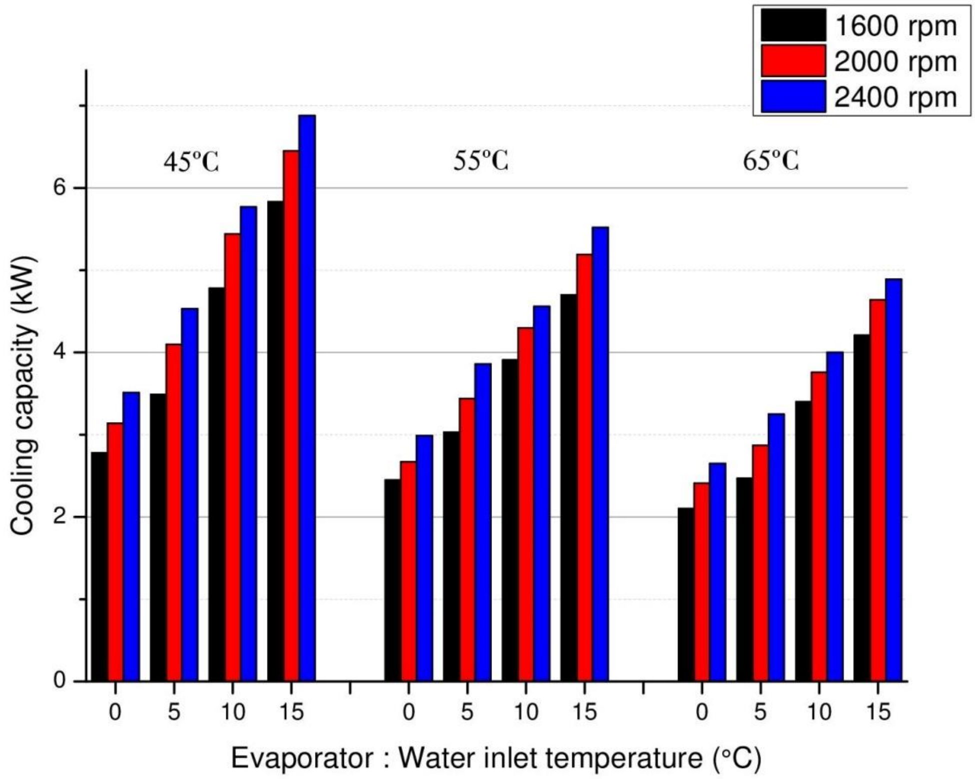

Figure 4 Cooling capacity variation with respect flow temperature and engine speed

3.1.3 Other heat pump parameters

During DEHP experiments, variation in discharge temperature and isentropic efficiency was observed to understand limitation and potential as a high temperature heat pump application.

Figure 5 shows variation in discharge temperature with respect to speed and evaporation temperature. Discharge temperature varies between 58˚C to 96˚C during all experiments. Speed increment shows noticeable impact on discharge temperature as discharge temperature increases with increasing engine speed whereas influence of evaporation temperature increment shows reverse trend and discharge temperature decreases with increasing evaporation temperature. The experiments results show promising higher limits of discharge temperature, a potential for high speed/flow temperature, suitable for high temperature applications.

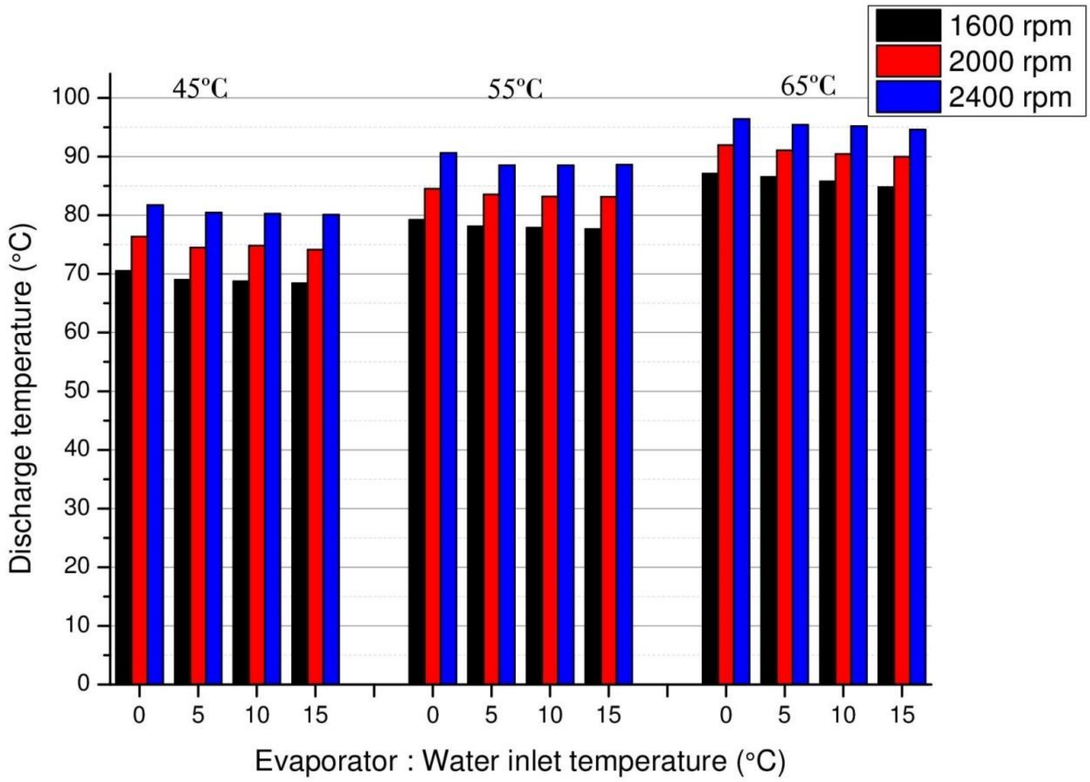

Figure 5 Discharge temperature variation with respect to flow temperature and engine speed

Similarly, isentropic efficiency on compressor side gives information on compressor performance and potential improvement. Figure 6 shows variation in isentropic efficiency (IE) for different flow temperature. IE remains in a range of 58% to 81% during all test conditions. For 35˚C flow temperature, IE remains around 65% at 1600 rpm. IE decreases by 6% with each speed increment whereas IE increases by 8% with increasing flow temperature. Evaporation temperature does not show any clear influence on IE; however, in 43% cases IE decreases with increasing evaporation temperature whereas in other cases it remains same or increases slightly. Thus, it is better to run heat pump system at lower possible speed and at higher flow temperature conditions in order to achieve higher isentropic efficiency.

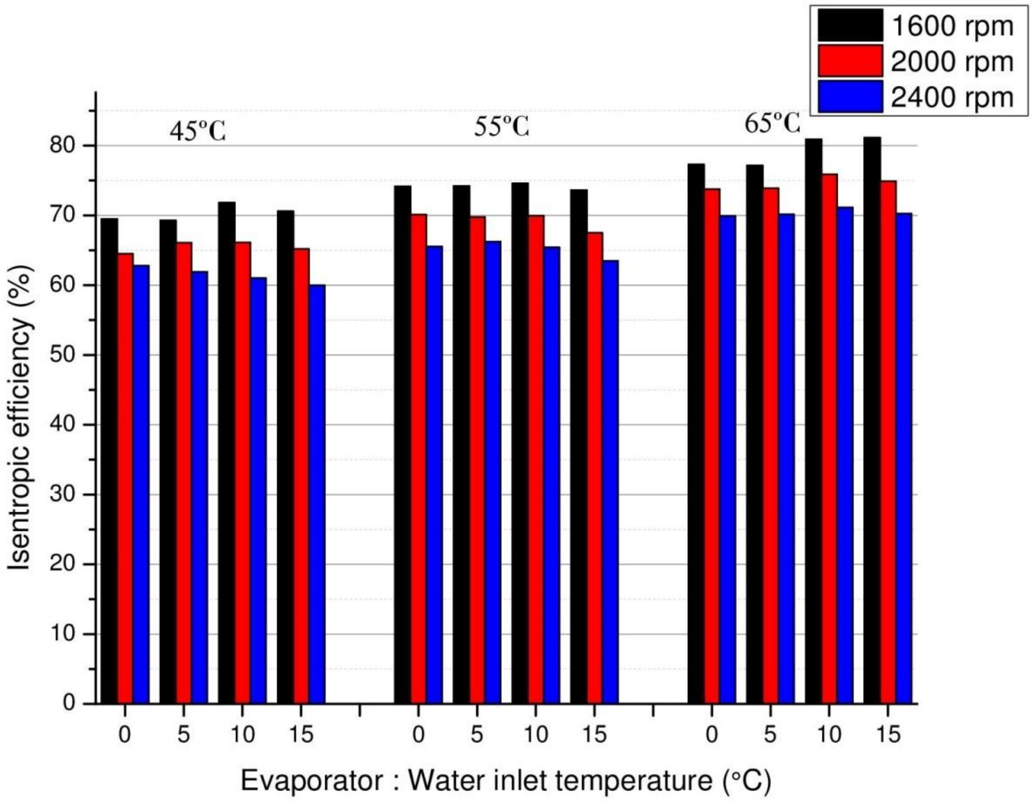

Figure 6 Isentropic efficiency variation with respect to flow temperature and engine speed

Other parameters such as refrigerant mass flow rate, power consumption and COP of heat pump were calculated from heating/cooling capacity and enthalpy difference across evaporator, condenser and compressor. From calculation, it is evident that refrigerant mass flow rate increases with engine speed and evaporation temperature whereas it decreases with flow temperature. Similarly, compressor power consumption increases with increasing speed and evaporation temperature due to increased pressure ratio and refrigerant mass flow rate that also reflects in COP outcome. COP analysis shows that COP varies in a range of 2.36 to 5.22. COP increases with evaporation temperature whereas it decreases with increasing engine speed which shows similar trend compared to other EHPs. However, for DEHP, primary energy ratio (PER) is appropriate measure than COP, hence, PER variation with engine speed and evaporation temperature has been discussed in more detail in later section.

3.2 Diesel engine performance including heat recovery

3.2.1 Diesel fuel consumption variation

Diesel engine performance is mainly dependent on temperature lift on heat pump side and speed of the engine. The compressor power demand decides load, fuel consumption and heat recovery potential from the engine. In addition, airflow rate and fuel flow rate are decided by engine geometry based on load/speed conditions. Diesel fuel flow rate with higher heating value gives fuel energy input for the DEHP system. Figure 7 shows variation in fuel energy input. Fuel energy input varies from 4.7 kW to 9.4 kW during all test conditions. For 35˚C flow temperature, fuel heat input remains in a range of 4.7 kW to 5.5 kW at 1600 rpm. Fuel consumption or fuel energy input increases with engine speed by 21% whereas it increases by 3% with flow temperature increment. Evaporation temperature increment also impacts fuel consumption and it increases by 7% with evaporation temperature. Hence, it is ideal to run DEHP system at possible lower speed, flow temperature and evaporation temperature to keep fuel consumption minimum. However, it affects other parameter such as heat recovery or thermal comfort that requires right balance point or strategy for operation.

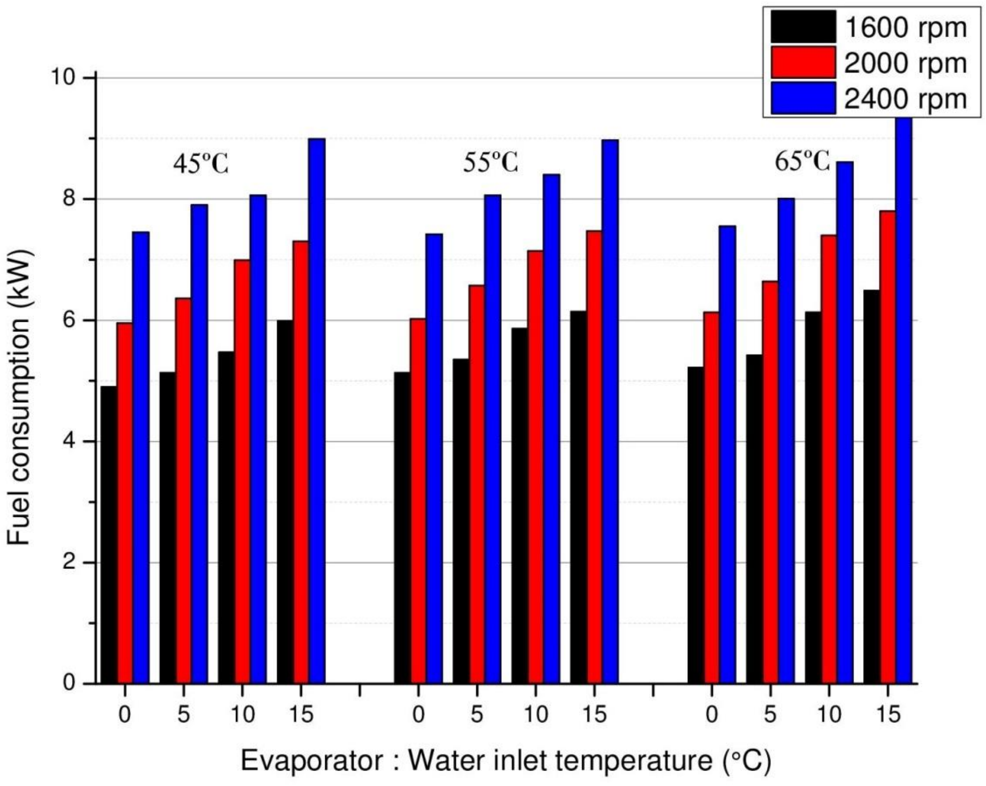

Figure 7 Diesel fuel: energy input variation with respect to flow temperature and engine speed

3.2.2 Coolant heat recovery and exhaust gas heat recovery

The coolant heat recovery (CHR) and exhaust gas heat recovery (EGHR) mainly depends on the engine load, speed and fuel consumption. Temperature lift on heat pump side influences the heat recovery from the engine. Figure 8 shows variation in CHR with speed, flow temperature and evaporation temperature. CHR varies from 1.2 kW to 2.6 kW during all test conditions. For 35˚C flow temperature, CHR remains in a range of 1.4 kW to 1.7 kW. CHR increases by 19% as speed increases whereas CHR increases by 10% with increment in evaporation temperature. However, influence of flow temperature is not very clear on CHR but in 60% test conditions, CHR increases with increasing flow temperature. Hence, CHR favours high engine speed and evaporation temperature.

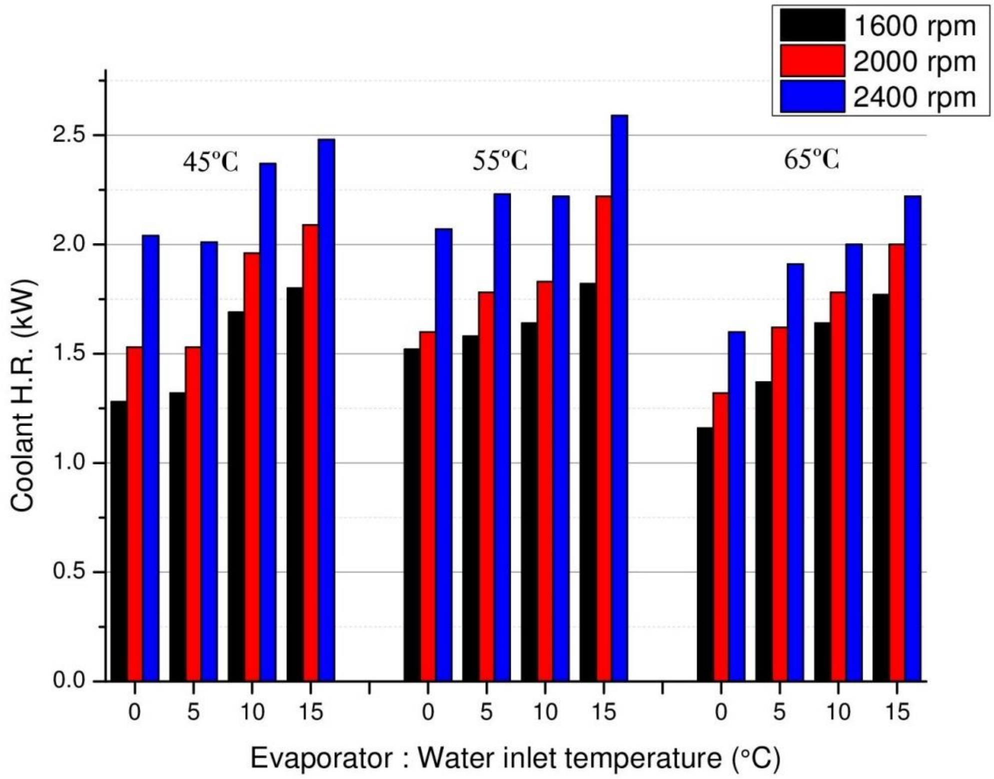

Figure 8 Coolant heat recovery variation with respect to flow temperature and engine speed

Similarly, EGHR also depends on many engine parameters. EGHR variation with respect to engine speed, flow temperature and evaporation temperature has been shown in Figure 9. EGHR varies between 0.5 kW to 1.6 kW during all test conditions. For 35˚C flow temperature, it remains between 0.66 kW to 0.73 kW at 1600 rpm. Speed increment has huge influence on EGHR and it increase by 39% with increasing speed. Evaporation temperature has less influence on EGHR and some cases it does not follow liner increment. However, in general, EGHR increases by 11% with increasing evaporation temperature whereas EGHR decrease by 3% with increasing flow temperature.

For DEHP system, total heat recovery varies in a range of 1.7 kW to 4.2 kW, a significant share to meet house heat demand. If total heat recovery variation is considered, for 35˚C flow temperature, it varies between 2 kW to 2.4 kW at 1600 rpm. Total HR increases by 25% and 11% with engine speed and evaporation temperature respectively. However, it decreases by 3% with increasing flow temperature. Hence, total heat recovery favours higher engine speed and evaporation temperature.

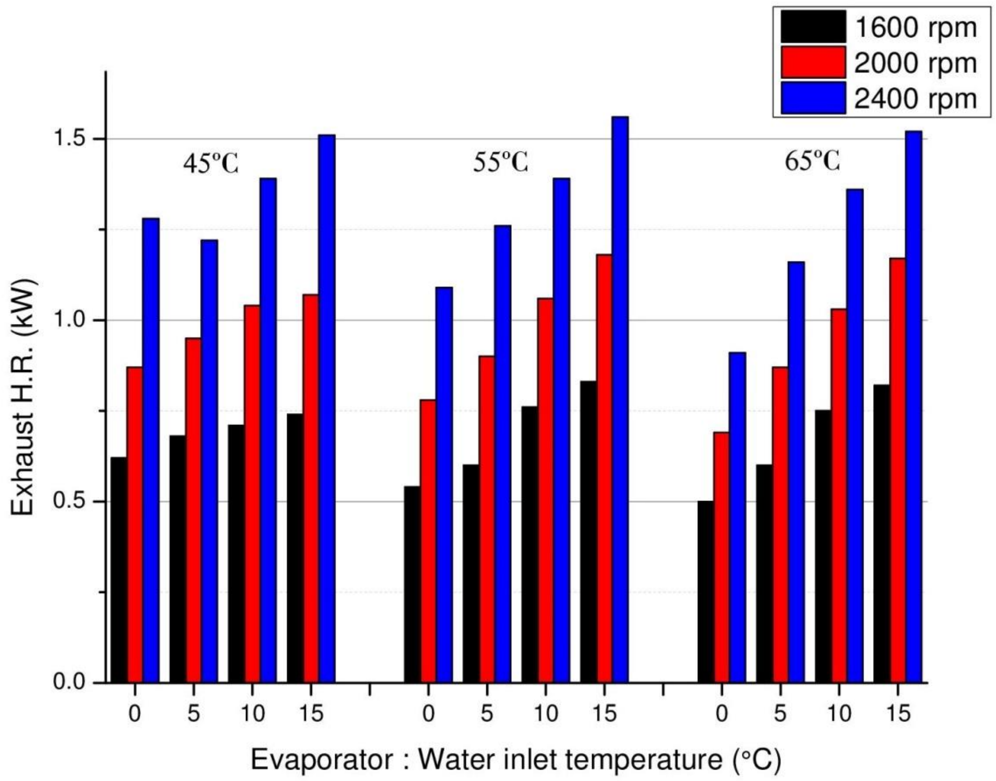

Figure 9 Exhaust gas heat recovery variation with respect to flow temperature and engine speed

3.2.3 Diesel engine energy distribution

Diesel engine energy balance is calculated based on fuel input energy, heat recovery, load on the engine and losses. Heat recovery and fuel energy calculated from experimental data whereas engine efficiency is calculate indirectly from power consumption. Heat losses from the engine were balanced as reminder from fuel input energy. Figure 10 show the diesel engine energy balance based on experimental data and calculation. It is evident that heat losses from the engine decreases as evaporation temperature increases due to higher engine efficiency and heat recovery. However, speed increment does not show any clear influence on the losses from the engine but in most cases heat losses decreases or remains same with increasing engine speed. In most cases, engine efficiency also increases with evaporation temperature and flow temperature due to increased load on the engine.

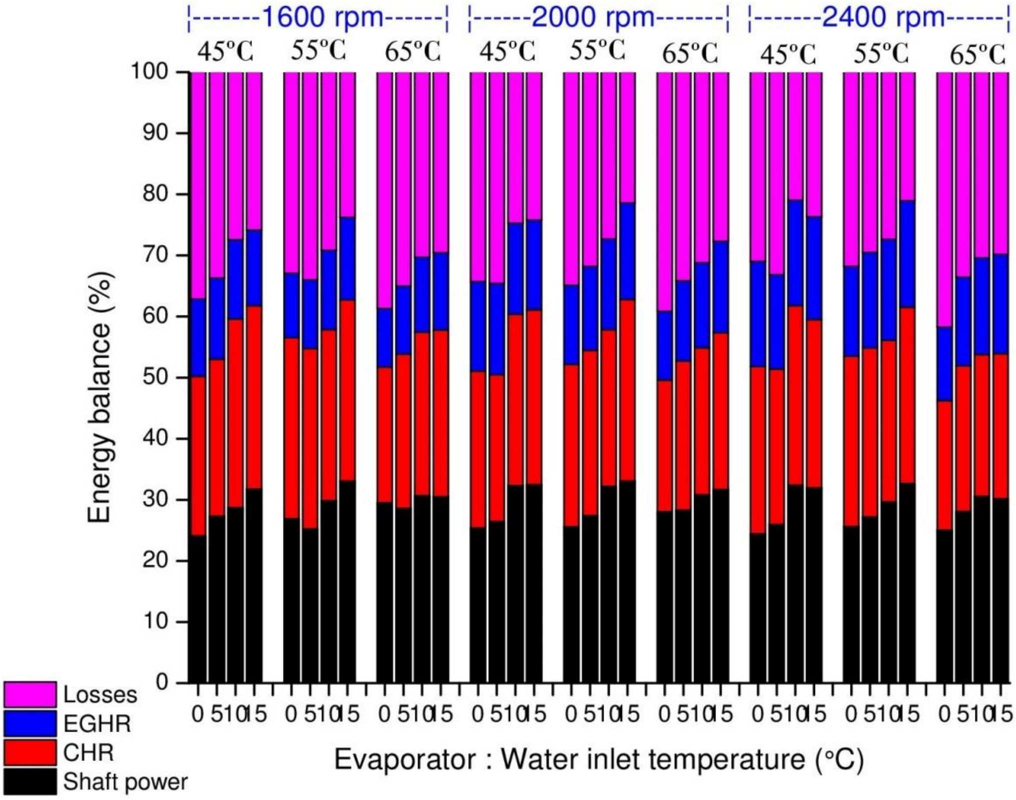

Figure 10 Diesel engine energy balance

3.3 Overall DEHP system performance

Overall DEHP performance analysis includes total heat output from the system, heat recovery share in total heat output and primary energy ratio where PER gives overall efficiency of the DEHP system.

3.3.1 Total heat output from DEHP system

Total heat output from DEHP system includes heat output from condenser, coolant heat recovery and exhaust gas heat recovery. Figure 11 shows variation in total heat output of DEHP system with respect to engine speed and flow temperature. Total heat output from DEHP system varies between 5.3 kW to 13.7 kW where DEHP heat output is 7.05 kW at 0˚C evaporation water inlet temperature at 2400 rpm. Hence, DEHP system is able to meet designed house demand with the help of heat recovery from the engine. Any heat output above the house heat demand would require reduction in DEHP speed or stop the system. Additionally, engine speed, evaporation temperature and flow temperature influences total heat output. Total heat output increases by 17% and 18% with increasing speed and evaporation temperature respectively whereas it decreases by 8% with increasing flow temperatures. The increment in total heat output is mainly due to increment in condensing capacity at higher evaporation temperature and speed. Hence, it is better to operate DEHP system at possible higher speed and at higher evaporation temperature to obtain higher total heat output from the system.

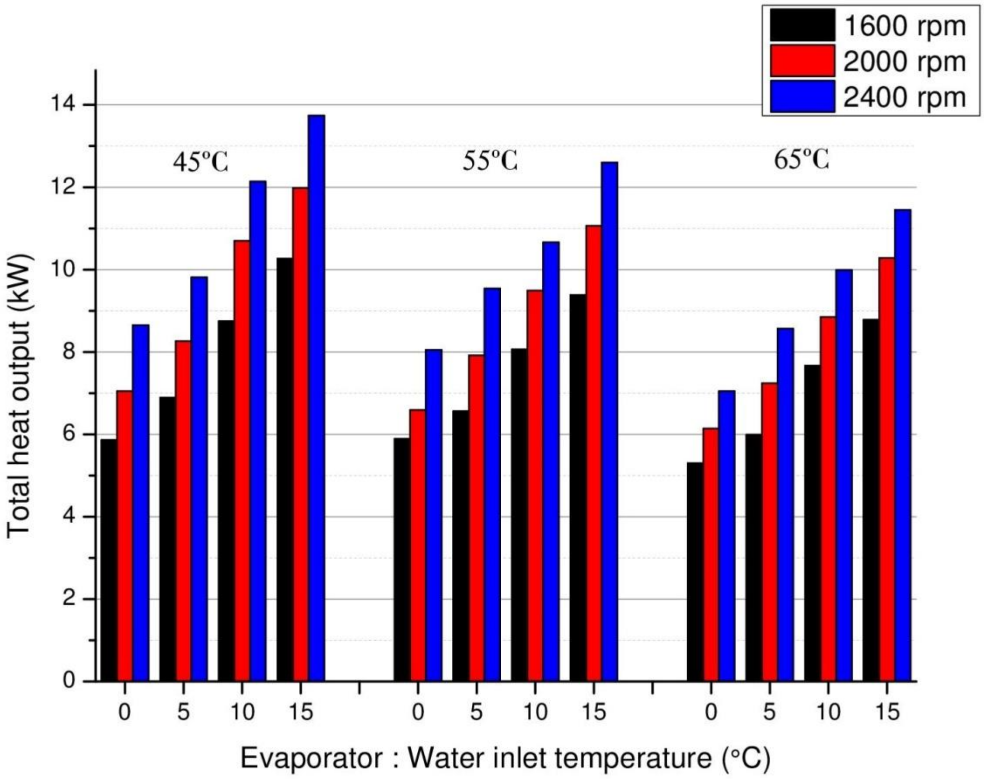

Figure 11 Total heat output variation with respect to speed and flow temperature

3.3.2 Heat recovery percentage in total heat output

Total heat output from DEHP system gives ability to meet house heat demand. However, it is important to understand heat recovery percentage in final heat output and variation in order to design larger system component and control system for efficient demand side management. Figure 12 shows variation in heat recovery percentage in total heat output for DEHP system. Heat recovery percentage varies from 22% to 39% in total heat output during all test conditions. For 35˚C flow temperature, it remains in a range 22 % to 31% at 1600 rpm. Heat recovery percentage increases by 7% with increasing speed whereas it decreases with evaporation temperature by 6% due to increased condensing capacity at higher speed. Flow temperature influence on heat recovery percentage is not very clear as other parameters but in 75 % cases heat recovery percentage increases with flow temperature. Hence, heat recovery percentage in total heat output support high speed, evaporation temperature and flow temperature conditions.

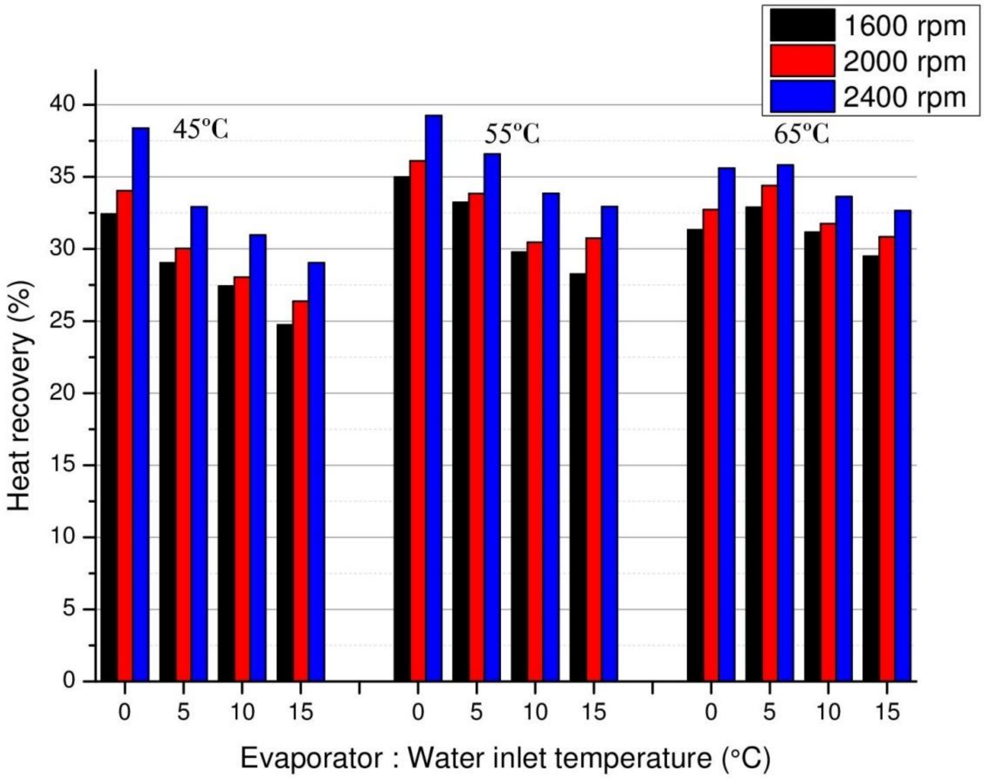

Figure 12 Heat recovery percentage in total heat output

3.3.3 Primary energy ratio

Primary energy ratio is a true indicator for DEHP system performance. PER is dependent on engine efficiency, heat recovery and heat pump performance. Figure 13 shows variation of PER for DEHP system. Primary energy ratio varies from 0.93 to 1.33 for all test conditions, which is still higher compared to efficient gas boiler system with 90% efficiency. For 35˚C flow temperature, PER remains in a range of 1.43 to 2, a promising results for underfloor heating system. PER decreases by 4% and 10% while increasing speed and flow temperature respectively. However, increasing evaporation temperature supports PER and PER increases by 11% with evaporation temperature. PER is lower at higher speed due to increased fuel input energy compared to total heat output. On contrary, PER increases with increasing evaporation temperature due to increased heat output from heat pump and load on the engine which gives higher total heat output compared to fuel energy input. Thus, in order to get higher PER value, it is better to run system at best possible lower speed. However, at lower speed DEHP system may not be able to meet heat demand. Hence, it is important to find balance between speed and heat demand in order to run system in the most efficient way.

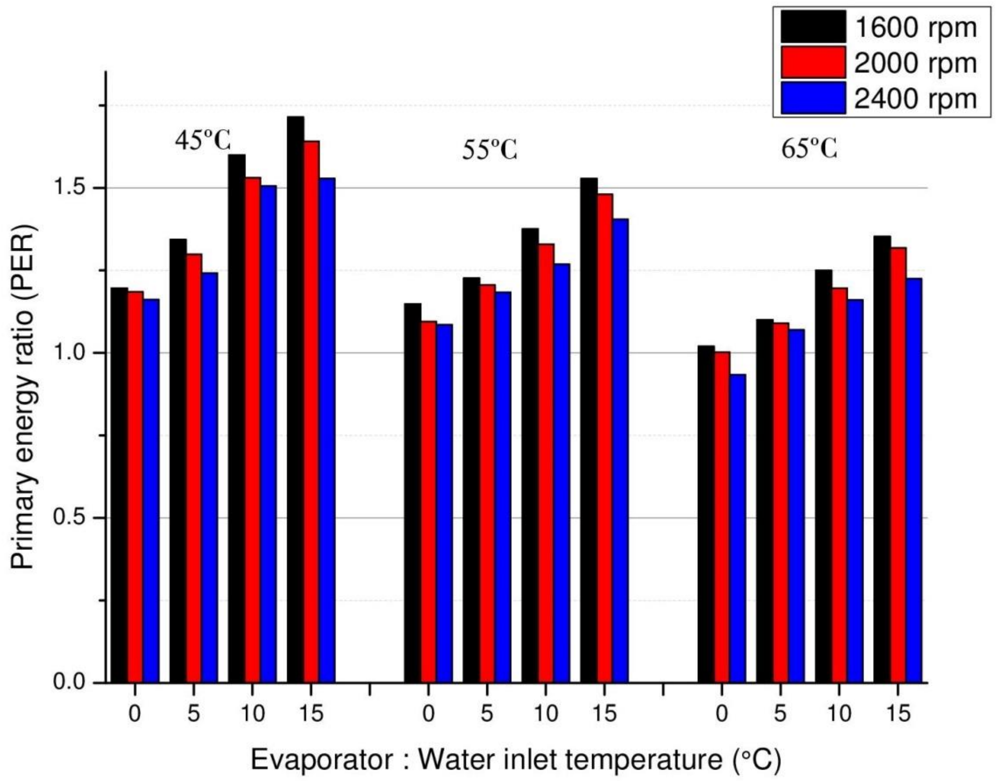

Figure 13 Primary energy ratio variation for DEHP system

In addition, DEHP system PER was compared with similar kind of EHP. In a case of EHP system, it is assumed that the diesel engine is replaced by an electric motor for same kind of system and power generation efficiency is taken into consideration for PER calculation. PER was compared for DEHP without heat recovery, DEHP with heat recovery and EHP system for same test conditions and system. Figure 14 shows PER comparison at 2400 rpm for DEHP and EHP system. It is evident that DEHP system does not show any benefits over EHP without heat recovery. However, due to heat recovery, DEHP gives 36% higher PER compared to EHP system where PER increment varies from 21 to 52% for given flow temperature conditions.

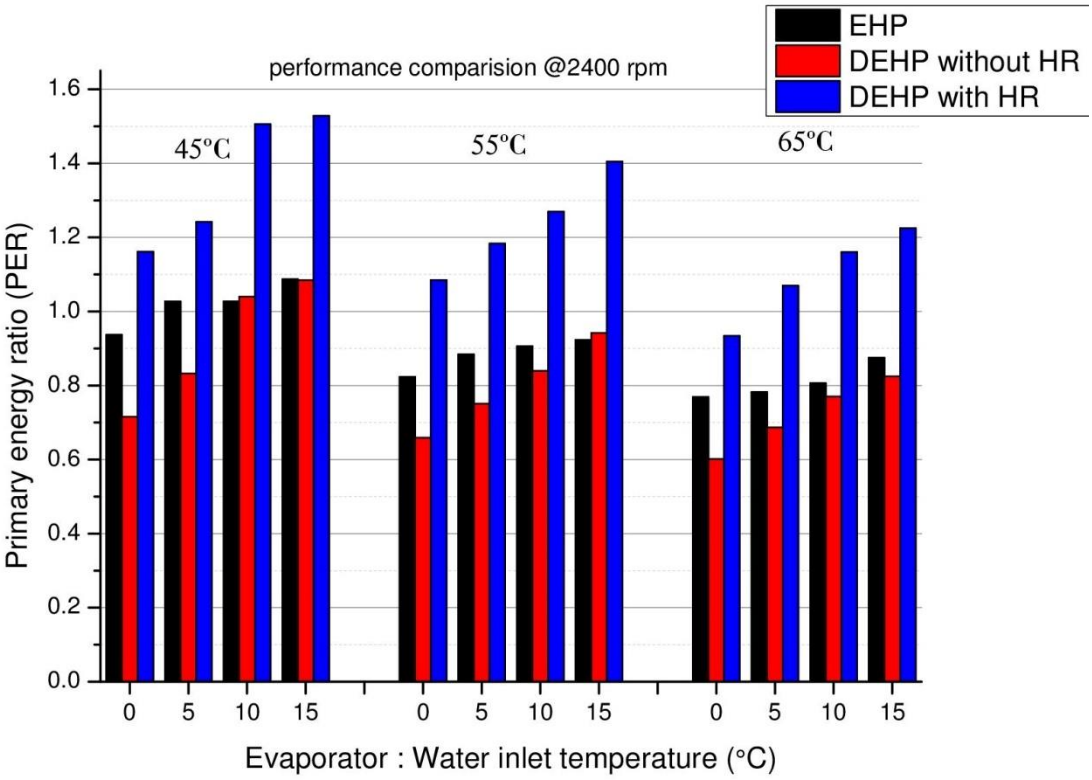

Figure 14 PER comparison at 2400 rpm

3.4 Achievable flow temperature and DEHP optimisation

3.4.1 Achievable flow temperature and flow rate

Experimental result shows clear benefits of heat recovery to meet house-heating demand. Due to that, it is possible to achieve higher flow temperature compared EHPs, a common requirement for various heat distribution system. During experiments, water outlet from condenser was directed to heat recovery system with help of three-way valve while maintaining water temperature difference across condenser as per standards. Hence, only part of condenser water flows through heat recovery system that increases water temperature further. Figure 15 & Figure 16 shows achievable final water temperature and water flow rate from DEHP system after heat recovery system. For 35˚C flow temperature, water temperature could be achieved between 38˚C to 43˚C at 1600 rpm. It is possible to achieve water temperature up to 74˚C from DEHP system for 65˚C flow temperature conditions that is similar to gas boiler. A black line in Figure 15 shows set point for flow temperature whereas bar graph above line shows final water temperature from DEHP system at respective speed and evaporation temperature. Final flow rate from heat recovery system remains between 3.5 l/min to 11.5 l/min. In fact, at higher flow temperature almost 90-97% condenser water flows through heat recovery system. Hence, higher flow temperature supports higher heat recovery flow rate and final temperature.

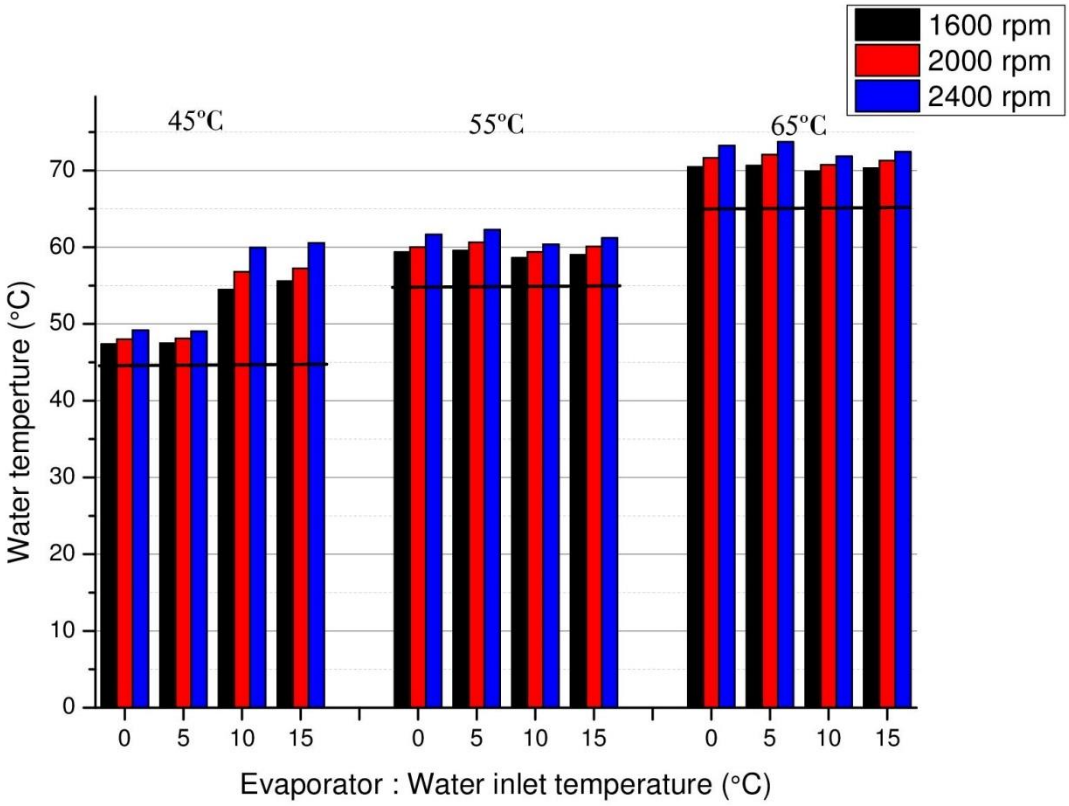

Figure 15 Achievable final water temperature

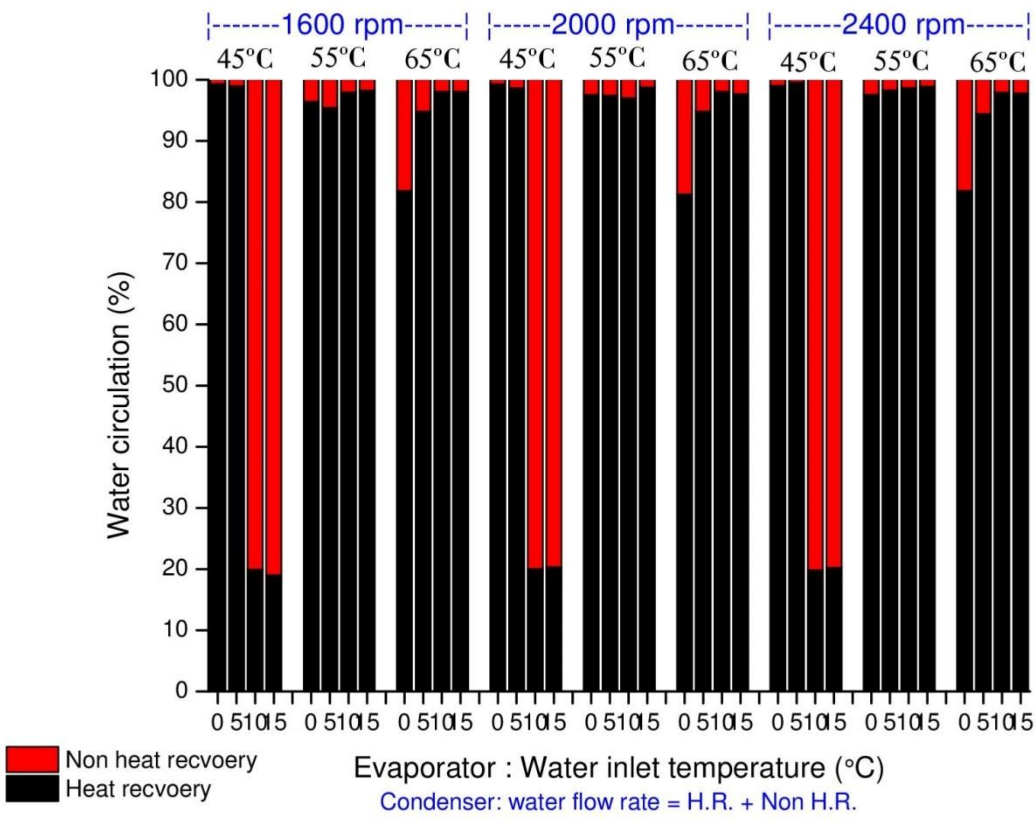

Figure 16 Achievable water flow rate from DEHP system due to H.R.

3.4.2 DEHP optimisation and comparison

DEHP has a good potential to provide higher water temperature and flow rate required for domestic application. Mostly, new heat distribution works in a temperature range of 35˚C to 65˚C. If DEHP is used to for such heat distribution system then it could provide better capacity control and thermal comfort due to heat recovery and engine speed modulation compared to fixed speed EHPs. Additionally, capacity modulation also plays important role to meet house heat demand during summer and winter.

In order to assess DEHP performance for better capacity control, experiments data were matched with house heating demand at respective temperature. A case for retrofit application at 65˚C flow temperature has been already presented by Shah et al. [30]. Figure 17 shows DEHP optimisation for underfloor heating (35˚C), medium temperature radiator (45˚C & 55˚C) and high temperature/DHW (65˚C) heating conditions. This case scenario includes total house demand at respective air temperature. Analysis shows that the system balance varies with flow temperature. At 35˚C, 45˚C, 55˚C and 65˚C flow temperature, DEHP system obtains balance point at just above 1600 rpm, 2000 rpm, 2200 rpm and 2400 rpm respectively. In summer month, with 35C flow temperature, almost all-heating demand can be met by heat recovery. This provides opportunity to use electric generator integration to produce electricity for a house that can act as a combined heat and power unit. Hence, DEHP system shows good potential with flexible speed variation to achieve better demand side management and thermal comfort.

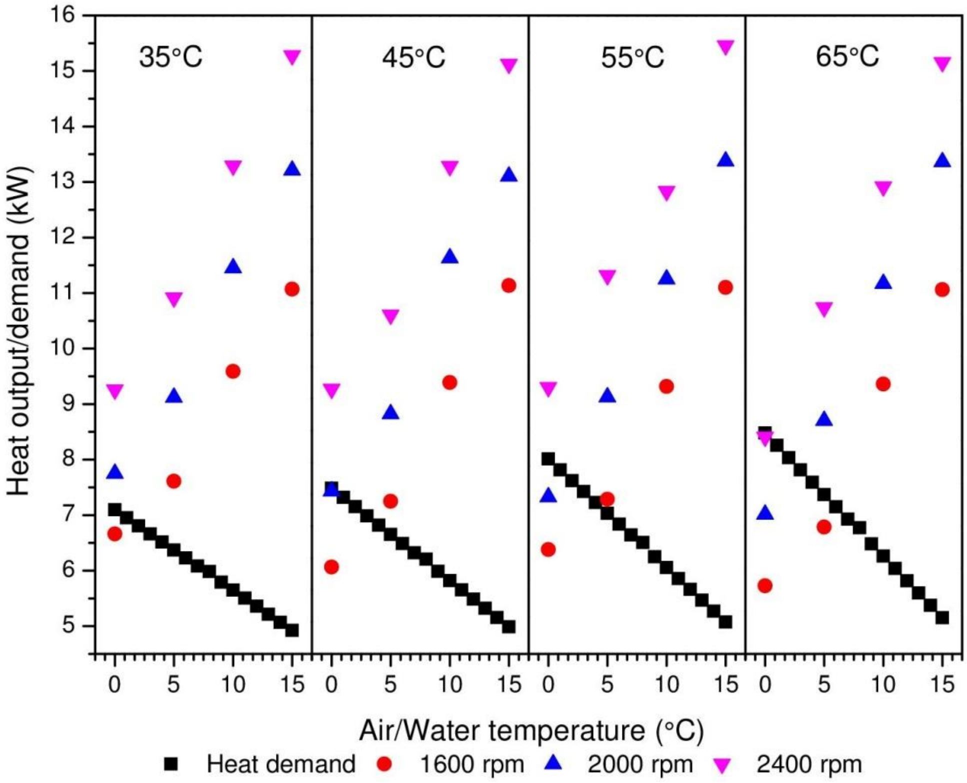

Figure 17 DEHP optimisation to meet space heating and DHW demand for different flow temperature

In addition, DEHP performance is compared with conventional and renewable energy technologies. For this comparison, annual heating demand of 3000 kWh is considered and based on system efficiency assumption primary energy input is calculated. Based on primary energy calculation, CO_2 emission has been calculated using Greenhouse gas calculator [34]. Oil boiler performance is taken as a reference in order to calculate CO_2 and primary energy savings. Potential of biodiesel driven DEHP system is also evaluated with other renewable technologies. Table 1 shows comparison of different heating system with DEHP. Results showed that electric heating is the most inefficient way to meet house demand in terms of primary energy consumption and emissions. Ground source heat pump and high COP air source heat pump also shows good potential. DEHP driven by 100% biodiesel shows very good potential to save primary energy and emission. Apart from electric heating, all other heating technologies are able to save money. Taking a case scenario of DEHP system efficiency of 80% (includes electrical and thermal efficiency) gives higher annual fuel cost saving compared to gas boiler. This efficiency is still lower compared to commercially available CHP unit from Honda that has declared efficiency of 85% [35].

Table 1 Comparison of different heating system with DEHP

Type of technology	**Sources of energy**	**System Efficiency**	**Total primary energy input (kWh)**	**Annual energy use (kWh)**	**Annual CO_2 Emission (kg)**	**Primary energy consumption savings (%)***	**CO_2 Emission savings (%)***
Electric heating	Electricity	100%	8824	3000	1851	-173%	-96%
Biomass boiler	wood chips, log , pellets	90%	-	3333	130	100%	86%
Oil boiler	Oil	93%	3226	3226	944	-	-
Gas boiler	Natural gas	90%	3333	3333	677	-3%	28%
Air source heat pump	electricity	COP =3	2941	1000	617	9%	35%
Ground source heat pump	electricity	COP = 4	2206	750	463	32%	51%
Gas engine driven heat pump	Gas	75%	2400	2400	488	26%	48%
Diesel engine driven heat pump	Diesel	75%	2400	2400	722	26%	24%
Biodiesel	Bio diesel	75%		2400	376	100%	60%

engine driven heat pump	(100%)						
Biodiesel (50%) driven heat pump	50% bio diesel blend	75%	1200	2400	574	63%	39%
Solar assisted hot water system	Sun	57%	-	3000		100%	100%

4 Conclusion

DEHP experimental results shows clear influence of engine speed, evaporation temperature and flow temperature on heat output, heat recovery and PER. DEHP test results is concluded as followings:

- Heating capacity and cooling capacity both increases with engine speed and evaporation temperature.
- Isentropic efficiency remains between 58 to 81% and it supports higher flow temperature.
- Fuel consumption increases with all three parameters due to increased load.
- Total heat recovery from the DEHP system remains around 1.7 kW to 4.2 kW and it is highly influenced by engine speed and evaporation temperature. Coolant heat recovery has higher contribution than exhaust gas heat recovery in total heat recovery.
- DEHP system offer heat output between 5.3 kW to 13.7 kW with help of heat recovery.
- Heat recovery contribution in total heat output remains around 22 % to 39%.
- DEHP system overall performance is measured by PER which remains around 0.93 to 1.33.
- DEHP system does not show much benefit without heat recovery. However, DEHP system with heat recovery is able to meet designed house heat demand with reduced system components size.
- In addition, waste heat recovery helps to obtain high water temperature up to 74°C, suitable for retrofit application

DEHP performance analysis shows great potential for domestic, commercial and industrial application due to engine waste heat recovery and speed modulation. During summer time, with the help of speed modulation, heat recovery alone can meet DHW demand giving potential to generate electricity, a good solution as a combined heat and power unit for remote area. An opportunity to use various biofuel with help of recovery makes it attractive as a renewable source of energy in rural area or off-gas/electricity area. In addition, comparative analysis shows that DEHP provides higher heat output and higher PER compared to EHP system. DEHP system has good potential to save primary energy, CO_2 emissions and annual fuel cost compared to oil boiler. Main issues of high initial cost can be solved by government support and mass production of advanced engine/compressor unit which can lower the engine cost in the range of 50£/kW in order to provide better power to heat ratio making it more attractive as co-generation unit for domestic application.

DEHP shows promising future as a transition technology for domestic retrofit application along with better capacity control and thermal comfort. Thermal and electrical energy storage integration with DEHP system can provide good solution to shift peak electricity/gas demand by efficient demand side strategy.

5 Acknowledgement

The author would like to thank Science Foundation Ireland - The Charles Parsons Energy Research Award for their financial support

6 References

[1] DECC, “Energy Consumption in the UK,” Department of Energy & Climate Change, London, 2015.

[2] DECC, “UK Greenhouse gas emissions, Final Figures,” Department of Energy & Climate Change, London , 2016.

[3] M. Esen and T. Yuksel, “Experimental evaluation of using various renewable energy sources for heating a greenhouse,” *Energy and Buildings,* vol. 65, pp. 340-351, 2013.

[4] M. Esen, “Thermal performance of a solar-aided latent heat store used for space heating by heat pump,” *Solar Energy,* vol. 69, no. 1, pp. 15-25, 2000.

[5] M. Akmal, B. Fox, D. Morrow and T. Littler, “Impact of high penetration of heat pumps on low voltage distribution networks,” in *PowerTech, 2011 IEEE,* Trondheim, 2011.

[6] P. Mancarella, C. Gan and G. Strabac, “Evaluation of the impact of electric heat pumps and distributed CHP on LV networks,” in *PowerTech, 2011 IEEE* , Trondheim, 2011.

[7] H. Singh, A. Muetze and P. Eames, “Factors influencing the uptake of heat pump technology by the UK domestic sector,” *Renewable Energy,* vol. 35, no. 4, pp. 873-878, 2010.

[8] J. Arteconi, N. Hewitt and F. Polonara, “Domestic demand-side management (DSM): Role of heat pumps and thermal energy storage (TES) systems,” *Applied Thermal Engineering,* vol. 51, no. 1-2, pp. 155-165, 2013.

[9] N. Hewitt, “Heat pumps and energy storage – The challenges of implementation,”

Applied Energy, vol. 89, no. 1, pp. 37-44, 2012.

[10] N. Hewitt, M. J. Huang, M. Anderson and M. Quinn, “Advanced air soruce heat pump for UK and European domestic bulidings,” *Applied Thermal Engineering,* vol. 31, no. 17-18, pp. 3713-3719, 2011.

[11] D. D. Colosimo, “Introduction to engine-driven heat pumps- Concepts, Approach and Economics,” *ASHRAE Transactions,* vol. 3, no. 1, pp. 987-996, 1987.

[12] A. Hepbasli, Z. Erbay, F. Icier, N. Colak and E. Hancioglu, “A review of gas engine driven heat pumps (GEHPs) for residentail and industrial applications,” *Renewable and Sustainable EnergyReviews,* vol. 13, pp. 85-99, 2009.

[13] P. Pradhan, H. Raheman and D. Padhee, “Combustion and performance of a diesel engine with preheated Jatropha curcas oil using waste heat from exhaust gas,” *Fuel,* vol. 115, pp. 527-533, 2014.

[14] A. Hossain and P. Davies, “Performance, emission and combustion characteristics of an indirect injection (IDI) multi-cylinder compression ignition (CI) engine operating on neat jatropha and karanj oils preheated by jacket water,” *Biomass and Bioenergy,* vol. 46, pp. 332-342, 2012.

[15] J. Parise and W. Cartwright, “Experimental Analysis of a Diesel Engine Driven Water-to-Water Heat Pump,” *Heat Recovery Systems & CHP,* vol. 8, no. 2, pp. 75-85, 1988.

[16] M. D'Accadia, M. Sasso and S. Sibilio, “Field Test Of A Small Size Gas Engine Driven Heat Pump In An Office Application: First Results,” *International Journal of Ambient Energy,* vol. 16, no. 4, pp. 183-191, 1995.

[17] F. Cascetta, M. Sasso and S. Sibilio, “A Metrological Analysis Of The In-Situ Evaluation Of The Performance Of A Gas Engine Driven Heat Pump,” *Measurements,* vol. 16, no. 4, pp. 209-217, 1995.

[18] J. Brenn, P. Soltic and C. Bach, “Comparison of Natural Gas Driven Heat Pumps and Electrically Driven Heat Pumps with Conventional Systems for Building Heating Purposes,” *Energy and Buildings,* vol. 42, no. 6, pp. 904-908, 2010.

[19] A. Zaltash, P. Geoghegan, E. Vineyard, R. Wetherington, R. Linkous and I. Mahderekal, “Laboratory Evaluation: Performance of a 10 RT GAs-Engine-Driven Heat Pump (GHP),” *ASHRAE Transactions,* vol. 114, no. 2, pp. 224-230, 2008.

[20] G. Nowakowski, M. Inada and M. Dearing, “Development And Field Testing Of A High-Efficiency Engine-Driven Gas Heat Pump For Light Commercial Applications,” *ASHARAE Transactions,* vol. 98, no. 1, pp. 994-1000, 1992.

[21] R. Kamal, A. Narasimhan, C. Wickramaratne, A. Bhardwaj, Y. Goswami, E. Stefankos and H. Ingley, “Field performance of gas-engine driven heat pumps in a commercial building,” *International Journal of Refrigeration,* pp. In Press, doi:10.1016/j.ijrefrig.2016.04.019, 2016.

[22] E. Elgendy, J. Schmidt, A. Khalil and M. Fatouh, “Performance of a Gas Engine Driven

Heat Pump for Hot Water Supply Systems," *Energy,* vol. 36, no. 5, pp. 2883-2889, 2011.

[23] E. Elgendy, J. Schmidt, A. Khalil and M. Fatouh, "Performance of a Gas Engine Heat Pump (GEHP) using R410A for Heating and Cooling Applications," *Energy,* vol. 35, no. 12, pp. 4941-4948, 2010.

[24] E. Elgendy and J. Schmidt, "Experimental Study of Gas Engine Driven Air to Water Heat Pump in Cooling Mode," *Energy,* vol. 35, no. 6, pp. 2461-2467, 2010.

[25] W. Zhang, T. Wang, S. Zheng, X. Peng and X. Wang, "Experimental Study of the Gas Engine Driven Heat Pump with Engine Heat Recovery," *Mathematical Problems in Engineering,* vol. 2015, no. Article ID 417432, p. 10, 2015.

[26] R. Zhang, X. Lu, S. Li, W. Lin and A. Gu, "Analysis on the Heating Performance of a Gas Engine Driven Air to WAter Heat Pump Based on A Steady State Model," *Energy Conversion and Management,* vol. 46, no. 11-12, pp. 1714-1730, 2005.

[27] Z. Yang, W. Wang and X. Wu, "Thermal modeling and operating tests for a gas-engine driven heat pump working as a water heater in winter," *ENERGY AND BUILDINGS,* vol. 58, pp. 219-226, 2013.

[28] Z. Lian, S. Park, W. Huang, Y. Baik and Y. Yao, "Conception of Combination of Gas Engine-Driven Heat Pump and Water-Loop Heat Pump System," *International Journal of Refrigeration,* vol. 28, no. 6, pp. 810-819, 2005.

[29] PHP, "Promotion of efficient heat pumps for heating (ProHeatPump)," 2009. [Online]. Available: http://www.proheatpump.eu/Downloads/. [Accessed 5 12 2011].

[30] N. Shah, M. Huang and N. Hewitt, "Experimental study of a diesel engine heat pump in heating mode for domestic retrofit application," *Applied Thermal Engineering,* vol. 98, pp. 522-531, 2015.

[31] M. Huang, N. Hewitt and N. Minh, "Field testing of an economised vapour injection heat pump," in *22nd IIR International Congress of Refrigeration*, Beijing, China, 2007.

[32] Kubota, "Kubota.co.uk," Engine, 2014. [Online]. Available: http://www.kubota.co.uk/product-range/engines-uk/product-range/single-cylinder-series/eb300/. [Accessed 5 January 2014].

[33] BSI, "EN14511: Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors for space heating and cooling (Part 1-4)," BSI, 2013.

[34] DEFRA, "Greenhouse Gas Conversion Factor," Department for Environment Food and Rural Affairs, London, 2009.

[35] G. Simader, R. Krawinkler and G. Trnka, "Buildup.eu (Micro CHP System: State-of-the -art)," 30 September 2010. [Online]. Available: http://www.buildup.eu/publications. [Accessed 29 January 2011].