EXPERIMENTAL STUDY ON A LOW TEMPERETATURE ORC UNIT FOR ONBOARD WASTE HEAT RECOVERY FROM MARINE DIESEL ENGINES

Aris-Dimitrios Leontaritis[1], Platon Pallis[1], Sotirios Karellas[1]*, Aikaterini Papastergiou[1], Nikolaos Antoniou[1], Panagiotis Vourliotis[1], Nikolaos Matthaios Kakalis[2], and George Dimopoulos[2]

[1]Laboratory of Steam Boilers and Thermal Plants, School of Mechanical Engineering, National Technical University of Athens, 9 Iroon Polytechniou, 15780, Athens, Greece.
Web page: http://www.lsbtp.mech.ntua.gr/

[2]DNV GL Strategic Research & Innovation East Med., Black & Caspian Seas/ Piraeus Hub

* Corresponding Author
e-mail: sotokar@mail.ntua.gr

ABSTRACT

The aim of this work is the experimental study of an ORC prototype unit which has been designed as a waste heat recovery system for the jacket water of marine diesel auxiliary internal combustion engines (ICEs). In order to simulate the operating characteristics of such engines, the heat input is in the order of 90kWth at low-temperature (90 ^{o}C) and is supplied by a natural gas boiler. The ORC unit produces 5 kWel of net electrical power, using as working medium the refrigerant R134a at a design cycle pressure of 25 bar and a temperature of 82 ^{o}C.
The experimental evaluation of the unit focuses more on operational issues than overall performance which has already been experimentally studied by a number of researchers. Accordingly, this study includes the investigation of the behaviour of the whole ORC system as well as of its key components under varying operational parameters, such as the occurrence of cavitation in the system feed pump and optimal scroll expanders operation. These outcomes contribute to an optimized configuration of the ORC system components and of the necessary measuring equipment as well as to the development of an efficient automatic control strategy of a dedicated ORC test bench which could then be directly coupled to an adequately sized marine auxiliary ICE for real time operation assessment.

1. INTRODUCTION

The energetic consumption of a commercial ship consists of propulsion and internal consumption electricity needs, which are entirely covered by specially designed marine diesel engines (main and auxiliary respectively). For large ships, the fuel expenses constitute about 30-55% of the total operational costs, depending on the type of vessel (Kalli et al., 2009). Strong motivation exists within the marine sector to reduce fuel expenses and to comply with ever stricter efficiency regulations, e.g. the EEDI -Energy Efficiency Design Index (Larsen et al., 2014a). Moreover, regarding emissions of CO_2 and oxides of sulfur and nitrogen (SO_x and NO_x) the international regulations are changing towards stricter limits (IMO, 2011). Currently, emphasis is being put on the improvement of the thermal efficiency of engines by optimizing their configuration in order to achieve lower fuel consumption (Jaichandar and Annamalai, 2012; Park, 2012). Also, research has been focused on advanced combustion technologies, such as the HCCI (Gan et al., 2011; Yao et al., 2009), the lean combustion (Zheng and Reader, 2004), and the stratified combustion (Park et al., 2012; Lu et al., 2011), in order to achieve a higher overall efficiency and reduce overall emissions. However, as these technologies have achieved quite a matured stage, it becomes harder to achieve further improvements by using these methods and thus a valuable alternative approach to improve overall energy efficiency is to capture and reclaim the ‘‘waste heat’’ (Shu et al., 2013). Hence, in times of high fuel prices, there are significant economic advantages associated with investing in marine diesel engine waste heat recovery (WHR) systems (MAN, 2012; Shu et al., 2013). WHR systems for electrical or mechanical power production can significantly contribute to dealing with these issues, with the ORC (Organic Rankine Cycle), the Kalina cycle and the steam Rankine cycle receiving the majority of attention in the literature. The steam Rankine cycle is focused on higher temperature WHR mainly from the exhaust gases of the main engines of a ship, while the ORC/Kalina cycles are more suitable for smaller engines (MAN, 2012), like the marine auxiliary ones, while at the same time they can also be used for WHR from lower temperature heat sources (e.g. the jacket water of diesel engines). However, optimisation results suggest that the Kalina cycle possess no significant advantages compared to the ORC or the steam cycle (Larsen et al., 2014a).

Although the Diesel process is highly efficient, large marine diesel engines are particularly well suited to be coupled with a WHR system (Larsen et al., 2014b), as the engine loses a large part of the fuel energy to the environment, mainly with the exhaust gases (up to 25% of the input energy) and the jacket water (up to 5.1% of the input energy; MAN, 2012). However, both of these heat sources, originating from the main engine, are used for covering the internal heating needs of a ship (e.g. heavy fuel oil pre-heating, fresh water generation, exhaust gas boiler), while the respective ones from the auxiliary engines remain unused. Several researchers have proposed WHR systems for main marine diesel engines (Larsen et al., 2014a-2014b; Bounefour and Ouadha, 2014; MAN, 2012; Yang and Yeh, 2014), but no experimental study or even a theoretical analysis has ever been conducted regarding the jacket water of auxiliary engines. In this perspective, the present work focuses on the recovery of heat from the auxiliary engines and more specifically from their jacket water. For the scale and heat source temperature level considered, both the ORC and the Kalina could be used as bottoming cycles. Bombarda et al. (2010) compared the two processes applied for WHR on large marine engines and found that both cycles, when optimized, produced equal power outputs. In the present paper, an ORC WHR system, specially designed for the jacket water of a marine auxiliary diesel engine is experimentally studied.

2. THE MARINE ORC PROTOTYPE TEST BENCH

The marine ORC prototype unit is based on a conventional low-temperature subcritical Organic Rankine Cycle using R134a as working medium. This experimental unit has been designed as a waste heat recovery system for the jacket water of marine diesel auxiliary internal combustion engines (ICEs). In order to simulate the operating characteristics of such engines, the heat input is in the order of 90kWth at a low-temperature (90°C), and is supplied by a natural gas boiler via an intermediate plate heat exchanger (evaporator). The boiler thermal output is adjustable and thus part load operation can be simulated as well. A schematic diagram of the unit is presented in **Figure 1**.

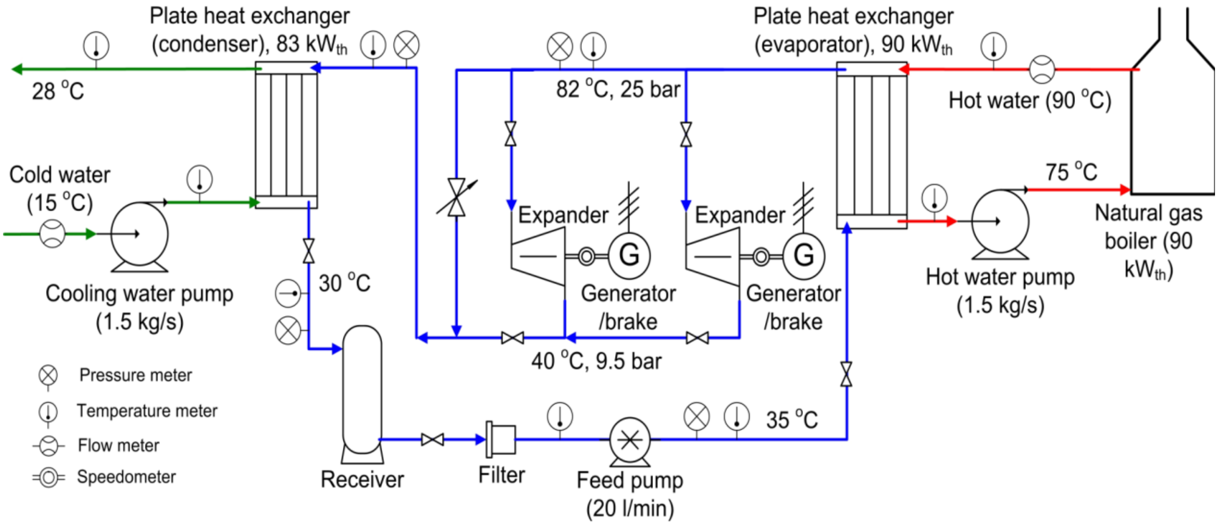

Figure 1: Schematic diagram of the ORC prototype unit.

2.1 Operation and control parameters of the experimental unit

The cycle is fed by a receiver (feed tank) at an average pressure of 9.5 bar and an average temperature of 30 ^{o}C. These parameters are controlled by the cold water flow in the condenser, which is adjusted by a regulatory valve.

The feed pump is a positive displacement multi-diaphragm pump that subsequently raises the pressure of the fluid at about 22-25 bar, depending on the operational conditions, and leads it to the evaporator. At a nominal speed of 960rpm a flow rate of 20lt/min is achieved. The rotational speed of the pump is controlled by a frequency drive. As a result, the refrigerant mass flow rate can be adjusted according to the unit load and the desired superheating temperature of the vapor, given the fact that the delivered volume flow rate of diaphragm pumps is in most cases a linear function of their rotational speed.

The high pressure vapor is expanded in two parallel scroll expanders, while a by-pass section controlled by an electromagnetic valve can alternatively lead the flow directly to the condenser. Actually, these expanders are two open-drive scroll compressors in reverse operation as it is thoroughly explained in the next paragraph. Each scroll expander drives an asynchronous motor/generator through a 1:1 belt drive, which can be coupled and uncoupled by an electromagnetic clutch. Both generators are connected to the 50Hz/400V electrical grid via a regenerative inverter module, which provides both grid stability and rotational speed control of the generators and hence of the expanders. For a given pump rotational speed (and thus mass flow rate), the inlet pressure of the scroll expanders is directly adjusted by their rotational speed, since the processed mass flow rate for volumetric machines is given by the product of the inlet density (ρ_{in}) multiplied by the swept volume (V_H) and the rotational speed (N_{rot}) of the machine (1).

$$\dot{M} = \rho_{in} \cdot V_H \cdot N_{rot} \tag{1}$$

An increase (decrease) of the rotational speed allows for a decrease (increase) of the density of the refrigerant at the expander inlet and thus causes a decrease (increase) of the respective inlet pressure. Finally, the expanded vapor is led to the condenser (plate heat exchanger), the condensate returns to the feed tank, and the cycle starts over.The ORC unit (**Figure 2**) produces 5 kWel of net electrical power, at a design cycle pressure of 25bar and a temperature of 82^{o}C.

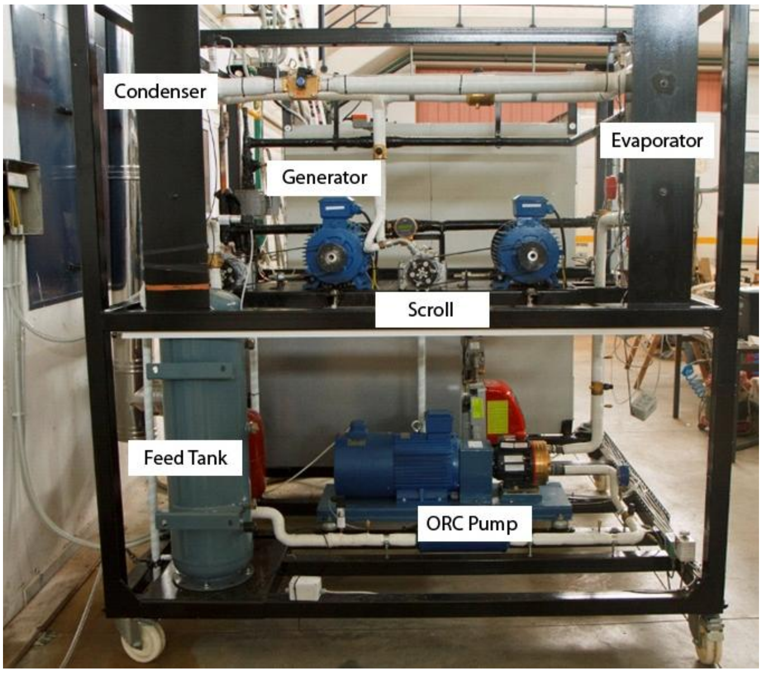

Figure 2: The Marine-ORC experimental unit

Various instruments have been mounted at all key-points of the cycle (**Figure 1**), in order to evaluate the performance of the different components of the ORC unit. Thermocouples and pressure transducers record the thermodynamic procedure; an electromagnetic flow-meter supervises the hot

water volume flow rate and two tachometers the scrolls' actual rotational speed. All important parameters regarding the electrical motors of both the pump and the generators, such as the consumed/produced active power are retrieved by the respective frequency drives.
It is noted that the automatic control of the system (including the frequency drives), the measurements and the data logging is materialised with the use of an industrial PLC (Programmable Logic Controller) and a SCADA (Supervisory Control and Data Acquisition) environment, which constitutes an important step towards the standardization and commercialization of such micro-scale units.

2.2 The scroll expanders

The expansion machine of an ORC system is a key component with critical influence on the overall system performance. Volumetric expanders are mostly suitable for micro-scale ORC systems (Declaye et al., 2013) as they are characterized by low mass flows, relatively high pressure ratios and much lower rotational speeds compared to turbo-machines (Quoilin, 2011). Scroll machines, in particular, are more favorable for such applications due to high performance and reliability, reduced number of moving parts, low price and broad availability at a wide power output range (Zanelli and Favrat, 1994).
In the power range of micro scale ORCs (up to a few kWs) there are currently no dedicated commercial scroll expanders available at an affordable price. Therefore, a viable solution is the use of a modified commercial scroll compressor (hermetic or open drive), designed either for air compression or for refrigeration applications, at reverse operation (Declaye et al., 2013).
In this study, two identical commercial open drive scroll compressors, originally designed for trucks' A/C systems were used as expanders. Their modification mainly focused on the inlet/outlet connectors and on the removal of the outlet (at compressor mode) check-valves so that they don't block the flow at reverse operation (expander mode).

Table 1: Scroll expanders characteristics

Swept volume (compressor mode)	121 cm^3/rev
Maximum Pressure	35bar
Built-involume ratio	≈2.3
Nominal Power output (expander mode)	3.5 kW

3. RESULTS and DISCUSSION

3.1 Cavitation effect on the ORC pump operation

A typical problem in micro scale ORC systems is the cavitation effect on the feed pump. Indeed, this problem was faced during the first steps of operation of the presented experimental unit, causing serious oscillations in its operation (mass flow rate, cycle pressure and temperature). In order to thoroughly understand this problem and finally solve it, an analysis of the pump operational conditions was conducted and is presented next.
First of all, in order to ensure stable operation of a pump, the **available** Net Positive Suction Head (**NPSHa**) at the pump inlet should exceed the respective **required** Net Positive Suction Head (**NPSHr**), given by the operation curves provided by the manufacturer, by at least 100mbar or an equivalent of 1 mH_2O. The NPSHa (mH_20) is calculated by the following equation:

$$NPSHa = Pt + Hz - Hf - Ha - Pvp \tag{2}$$

Where:
Pt = Pressure at the pump inlet
Hz=Vertical distance from liquid surface to pump center line
Hf = Friction losses in suction piping
Ha = Acceleration head at pump suction
Pvp = Absolute pressure of liquid at pumping temperature

The acceleration head factor (Ha) is calculated by equation (3).

$$\mathrm{Ha} = \frac{C \cdot L \cdot V \cdot N}{K \cdot G} \tag{3}$$

Where:

C = Constant determined by type of pump (in our case Wanner Engineering, Hydra Cell D/G10)
L = actual length of suction line
V = Velocity of liquid in suction line
N = RPM of crankshaft
G= Gravitational constant
K = Constant to compensate for compressibility of the fluid

For the operation conditions at the design point of the experimental unit the NPSHr is 500mbar, Hz=0.3m, Ha≈200mbar, and Hf=200mbar.

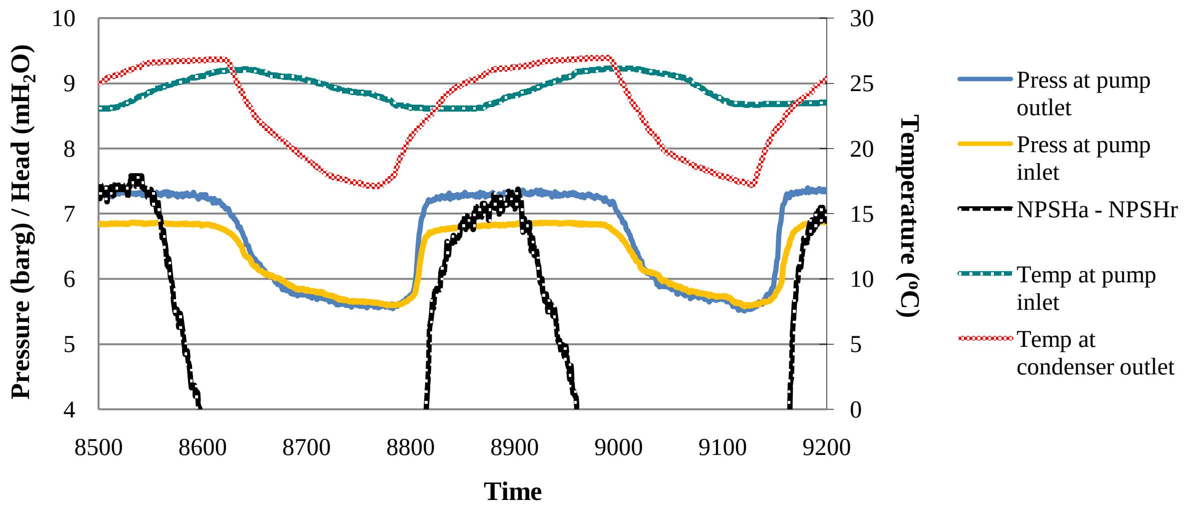

Figure 3: Cavitation effect on ORC pump operation.

The main parameters of the ORC feed pump under operation with cavitation effect are depicted in **Figure 3**. In fact, the ORC pump was tested while just circulating the refrigerant around the ORC circuit via the scroll by-pass section and thus practically no pressure raise is implemented by the pump. Analyzing the pump operation at the first oscillation cycle (cold start), it is observed that initially the pressure at the pump inlet/outlet remains constant with time, indicating a constant mass flow rate, and that the NPSHa - NPSHr difference is maintained well above the threshold of 100mbar (1 mH_2O). As the whole system is ramping up, the temperature at the condenser outlet raises due to the increase of the evaporator outlet temperature. Consequently, the temperature at the pump inlet raises but with a significant time lag caused by the thermal inertia of the feed tank, which stands between the condenser outlet and the pump inlet. With raising temperature at the pump inlet, the absolute pressure of the refrigerant (factor Pvp of equation (2)) raises and as a result the NSPHa drops. When the difference NPSHa – NPSHr reaches a critical value of around 100mbar (1 mH_2O), the cavitation effect is initiated and the circulating mass flow rate drops significantly. Simultaneously, the pressure of the circuit (controlled by the condensation temperature which drops due to the reducing refrigerant mass flow rate) also drops (factor Pt of equation (2)) and the unit's operation practically collapses. To make things worse, even though at this point the refrigerant temperature at the condenser outlet drops dramatically, since there is practically a zero mass flow rate, the feed tank needs time to cool down and keeps feeding the pump at relatively high temperature (and thus high Pvp); at this point the NPSH difference is strongly negative. Eventually the feed tank cools down, lowering the pumping temperature and thus raising the NPSH difference. Gradually the cavitation effect fades, the mass flow rate raises, and a new cycle starts over.
In order to solve this problem a water cooled heat exchanger was installed in the suction line of the ORC pump, downstream of the liquid receiver, so that the pumping temperature and thus the absolute

pressure of the refrigerant (factor Pvp of equation (2)) are maintained at lower values, ensuring stable pump operation. The main parameters of the ORC pump under operation with the additional sub-cooling heat exchanger are depicted in **Figure 4**. The measurements have been obtained at similar operation conditions with **Figure 3**, allowing their direct comparison. The sub-cooling heat exchanger causes an average 2K temperature drop at the suctioned refrigerant which has proved to be sufficient for the stable operation of the unit. As it can be seen in the diagram, the NSPHa is constantly kept above 17 mH_2O with a required NPSH of 5 mH_20. Its main fluctuations are caused by the suction pressure, which in turn depends on the cooling water mass flow at the condenser (or equivalently on the condenser outlet temperature) and naturally by the temperature at the pump inlet which affects the factor Pvp as already discussed. Accordingly, between t=300 and t=690 the NPSHa is slightly dropping even though the suction pressure is slightly raising, due to the greater influence of the raising temperature at the pump inlet (factor Pvp). The evident drops of NPSHa at t=700, t=840 and 1040 are caused by marginal steps of increasing cooling water mass flow rate at the condenser which directly influence the pressure at the pump inlet and thus the NPSHa. At the respective intervals the observed NPSHa raise is caused by the slightly decreasing temperature at the pump inlet. It is finally noted that the stable operation of the feed pump can be confirmed by the observation of the almost constant delivered Head of the pump over time (Pout-Pin).

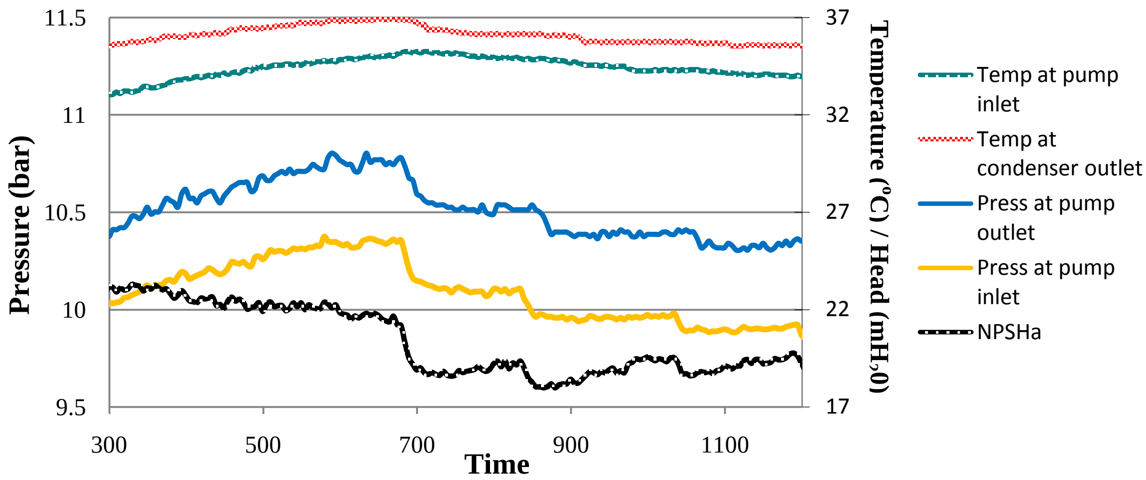

Figure 4: Cavitation free ORC pump operation with the addition of a sub-cooling heat exchanger

Extensive testing under different load and other operational conditions proved this solution as sufficiently effective to ensure the stable operation of the ORC pump and thus of the whole system. All the results presented from this point on in this paper refer to the experimental unit with the fitted additional heat exchanger. However, there are indications of partial cavitation occurrence at high pump rotational speeds which need to be further studied. As cavitation can be detected by the reduced delivered mass flow under constant pump rotational speed, a coriolis type mass flow meter has to be installed to the test rig in order to further investigate this issue, while the influence of the sub-cooling degree also requires to be studied.

3.2 Scroll expanders' operation

In this section the scroll expander's operation is presented through diagrams of its main operational parameters. It is noted that at this point of studying, the presented data can be used for a qualitative analysis of the behavior of the used scroll expanders under various conditions and its comparison with the findings of other researchers. **Figure 5** depicts the influence of the **scroll rotational speed (**$Nrot_{exp}$) and the **supply (inlet) temperature** (Tsu) of the refrigerant on the **overall isentropic efficiency** ($epsilon_s$). The presented surface has been produced by the statistical fitting of the experimental data. $Epsilon_s$ is defined as:

$$\text{epsilon}_s = \frac{Pel_{gross}}{\dot{M}_{in} \cdot (h_{su} - h_{ex,is})} \quad (4)$$

Where:

Pel_{gross} the gross electrical power output of the ORC unit (equal to the net electrical power output of the two scroll expanders)
$\dot{M}_{in}$ the total circulating mass flow rate
h_{su} the suppy enthalpy of the refrigerant at the expander inlet
$h_{ex,is}$ the enthalpy at the ideal isentropic expansion point at the measured exhaust pressure

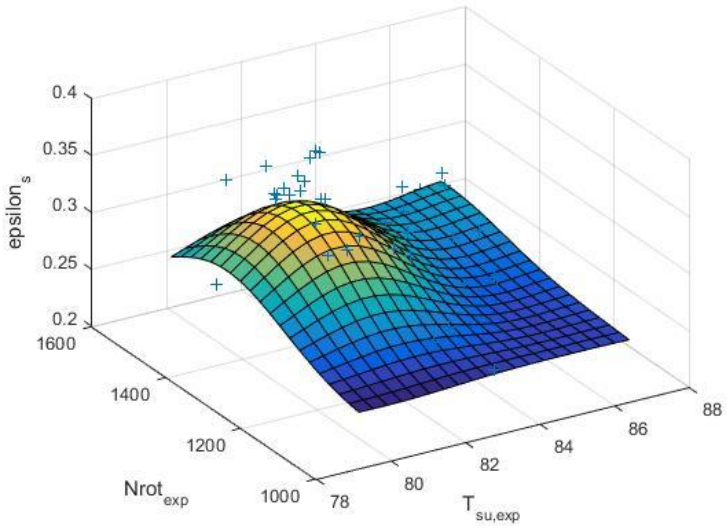

Figure 5: Scroll expander overall isentropic efficiency as a function of supply temperature ($Tsu_{,exp}$) and rotational speed ($Nrot_{exp}$)

Using this definition for the isentropic efficiency, all electromechanical losses, which can account for up to 40% of the gross generated electircal power, are included and that is the reason why the efficiency appears to be quite low.. The use of inverter frequency drives and induction motors/generators gives much room for improvements which is part of the work planned ahead. This was confirmed by low measured values of the power factor which indicates the low electrical efficiency of the generators. In order study the behaviour of the expander itself separately a dyno-meter or torque meter should be installed directly on its shaft. At any case, the results are useful for the qualitative analysis of the system. Analysing this diagram, the first obvious observation is that the isentropic efficiency is maximised near the nominal design conditions ($Nrot_{exp}$=1500 rpm / Tsu=82^{o}C). From a thermodynamic point of view, this was expected since the optimisation objective during the design of this system (as in most heat recovery systems) was the power output and not the cycle thermal efficiency (Braimakis et al., 2015; Quoilin et al., 2011). Moreoever, the expander itself is expected to have a better efficiency near its design point (i.e.when the imposed volume/pressure ratio is near the built in volume ratio of the expander) where over-expansion and under-expansion losses are minimized.

Focusing on the influence of the supply temperatue, it is concluded that a 4 to 5K degree of superheating of the live vapor gives the optimum results (at 25 bar the saturated vapor has a temperature of 77.5^{o}C). As other researchers have pointed out (Quoilin et al., 2011; Mago et al., 2008; Yamamoto et al. 2001), the superheating at the evaporator exhaust should be at low levels when using high molecular weight organic fluids, such as R134a .

The impact of the rotational speed can be better understood by the explanation of Figure 6,which presents the overall isentropic efficiency of the expander as a function of the filling factor (Φ) under various rotational speeds. The filling factor expresses a relative measure for the internal mass flow leakages and the respective power losses. The filling factor is defined as:

$$\Phi = \frac{\dot{M}_{in}}{V_{swept} \cdot N_{rot} \cdot \rho_{su}} > 1 \qquad (5)$$

Where:

$\dot{M}_{in}$ the total circulating mass flow rate
V_{swept} the bulit in swept volume of the scroll machine at expander mode
N_{rot} the expander rotational speed
ρ_{su} the suppy density of the refrigerant at the expander inlet

As expected, the rotational speed highly affects Φ, and as it can be observed in the diagram, the lower the rotational speed is the higher the filling factor gets due to the larger relative impact of the internal leakages. This effect has also been noted by Lemort et al. (2009) in an experimental analysis of an open drive scroll expander. Other operational parameters such as the inlet pressure and the imposed pressure ratio also affect the filling factor and that is the reason why its value varies for constant rotational speeds. Naturally, Φ in turn affects the overall isentropic efficiency which gets reduced by increasing internal leakages.

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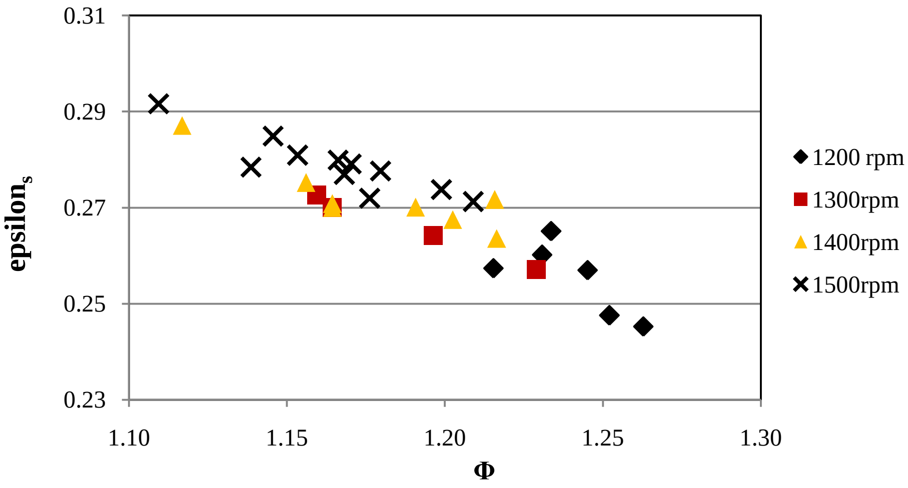

Figure 6: Scroll expander overall isentropic efficiency as a function of the filling factor (Φ)

Another interesting issue is the actual mass flow rate that circulates through the ORC circuit. Currently this value is calculated through the heat balance in the evaporator. The temperature and pressure of the heat source (water) as well as the volume flow rate are measured so the heat input rate ($\dot{Q}_{in}$) is known. At the same time the inlet and outlet conditions (pressure and temperature) of the refrigerant are also measured and thus considering zero heat exchange losses the circulating mass flow rate ($\dot{M}_{in}$) can be calculated by the following equation:

$$\dot{M}_{in} = \frac{\dot{Q}_{in}}{(h_{out} - h_{in})_{refrigerant}} \tag{6}$$

A cross-check of this value can be done through the heat balance at the system condenser. For this purpose an ultrasonic mass flow meter and two thermocouples (condenser inlet-outlet) were installed at the cooling water circuit and the dissipated heat rate was this way indirectly measured. In the following diagram (**Figure 7**), the values of the measured condenser dissipated heat rate (Qmeas) vs the respective calculated values (Qcalc) using the above-mentioned value (equation 6) of the mass flow rate ($\dot{M}_{in}$) are presented. The relative declination (ΔQ%) between these values is within 2-8% which is satisfactory for the needs of the present study. Moreover, the total system heat balance ($Q_{tot} = Q_{evap} + W_{pump} - W_{scrolls} - Q_{cond} - Q_{subcooler} - Q_{amb}$) gives a calculated Qamb of about 600W which is a realistic value for the non calculated ambient heat losses through the pipes of the system. However, in order to investigate certain operational issues such as the occurence of partial cavitation in the ORC pump and the filling factor of the expander, accurate measurements of the mass flow meter with a coriolis mass flow meter are absolutely necessary.

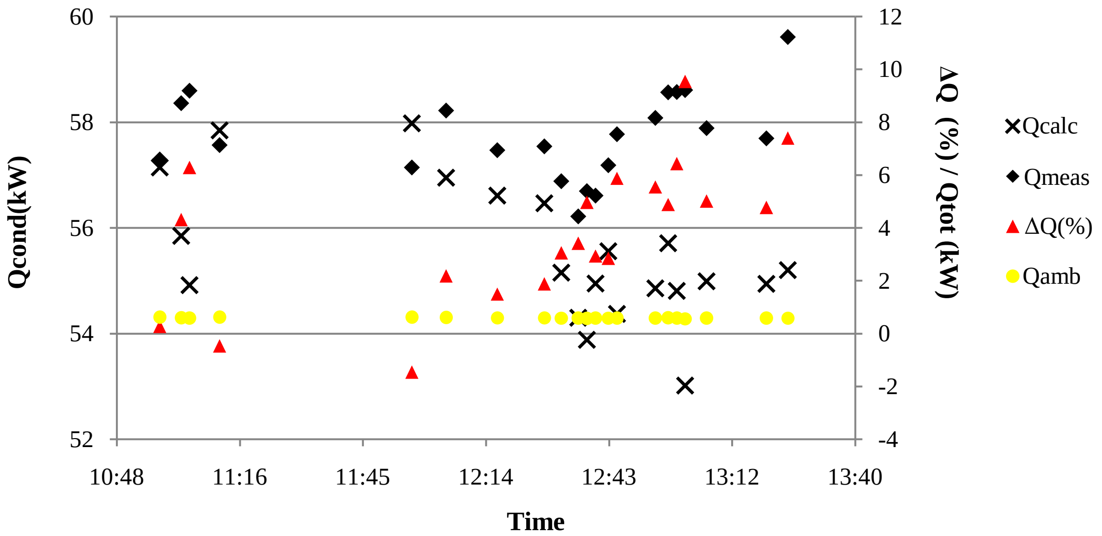

Figure 7: Condenser and overall heat balance

4. CONCLUSIONS

- A coriolis mass flow meter is an indispensable part of the measuring equipment of such experimental benches.
- Cavitation effect in ORC feed pumps can be dealt with the addition of a sub-cooling heat exchanger at the suction line of the pump. The effect of the sub-cooling degree on the cavitation effect as well as on the system performance needs further investigation.
- Partial cavitation at high pump loads and/or rotational speeds and its effect on the feed pump as well as on the overall system efficiency is another issue that requires further studying.
- The relatively low overall scroll expanders isentropic efficiency implies a low conversion of mechanical power into electrical. This was confirmed by low measured values of the power factor. Much room for performance enhancement through the optimization of the main operational parameters of the the induction motors/generators, such as the motor control method and its relative slip speed.
- A torque meter has to be installed directly on the expander's shaft in order to evaluate its performance separately. This is also necessary for the performance optimization of the electrical power generation sub-system.

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(Quoilin et al., 2011; Mago et al., 2008; Yamamoto et al. 2001)

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