EXPERIMENTAL TESTING OF A SMALL-SCALE SUPERCRITICAL ORC AT LOW-TEMPERATURE AND VARIABLE CONDITIONS

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ABSTRACT

The detailed experimental investigation of an organic Rankine cycle (ORC) is presented, which is designed to operate at supercritical conditions. The net capacity of this engine is almost 3 kW and the temperature of the hot water is always lower than 100 ^{o}C. The laboratory testing of the engine includes the variation of the heat input and of the hot water temperature. The maximum heat input is 48 kW, while the hot water temperature ranges from 65 up to 100 ^{o}C.

The tests are conducted at the laboratory and the heat source is a controllable electric heater, which can keep the hot water temperature constant, by switching on/off its electrical resistances. The expansion machine is a modified scroll compressor with major conversions, in order to be able to operate with safety at high pressure (max. pressure around 40 bar). The ORC engine is equipped with a dedicated heat exchanger of helical coil design, suitable for such applications. The speeds of the expander and ORC pump are regulated with frequency inverters, in order to control the cycle top pressure and heat input. The performance of all components is evaluated, while special attention is given on the supercritical heat exchanger and the scroll expander.

The performance tests examined here are the ones for hot water temperature of 95 ^{o}C, with the aim to examine the engine performance at the design conditions, as well as at off-design ones. Especially the latter are very important, since this engine will be coupled with solar collectors at the final configuration, where the available heat is varied to a great extent.

The engine has been measured at the laboratory, where a thermal efficiency of almost 6% has been achieved, while supercritical operation did not show superior performance as expected, due to the oversized expander. A smaller expander would allow operation at even higher pressures for higher speed with increased electric efficiency, which would probably reveal the full potential of the supercritical operation.

1. INTRODUCTION

The organic Rankine cycle (ORC) technology is suitable for heat recovery applications for low-temperature of even lower than 100 ^{o}C (Manolakos *et al*., 2009a). At such conditions its efficiency is rather low, usually in the range of 3-5%, but still there are cases where its cost-effectiveness can be secured, especially when the heat input comes from waste energy sources. The main advantage at this temperature range is the simple and low-cost heat source circuit, since even pure water can be used with low-pressure piping, while the use of glycol or thermal oil is avoided, as well as a simple ORC configuration with a single expansion machine and no internal heat exchangers (Kosmadakis *et al*., 2013a).

Supercritical cycles have some interesting features, since a better thermal match exists between the hot source and the organic fluid (Schuster *et al*., 2010). This aspect is beneficiary for thermal and exergy efficiency, while at the same time some restrictions are introduced to the components selection due to the high pressure operation. This is the common case for steam cycles, which tend towards such configuration, since in large-scale utility steam power plants the use of supercritical cycles has been already accomplished. But the largest challenges are for the small-scale systems using an ORC, since it is difficult to find and select the appropriate components for such heat-to-power engines. Moreover, low-temperature operation (below 100 ^{o}C) brings some additional restrictions, since a limited number of fluids can be used for such purposes and the cycle configuration does not allow any flexibility.

The most important component in ORC engines is the expansion machine and there is intensive research effort for producing expanders (of positive displacement type or even turbines for larger systems). For small-scale systems with power production lower than around 20 kW, scroll expanders have been widely used and showed adequate performance and expansion efficiency (Lemort *et al*., 2012). The present authors have also used the same expansion technology (both open-drive and hermetic ones) and revealed the good performance at a wide range of pressure ratios (Manolakos *et al*., 2009a). This brings confident that such expander can be also used at a supercritical cycle, where the biggest challenge is the increased pressure. One positive aspect is that for low-temperature applications, the pressure ratio for both cycle types (supercritical and subcritical) is low and usually in the range of 2-3 (Kosmadakis *et al*., 2013a). In such pressure ratio range the scroll expander operates with good efficiency.

There are various studies focusing on supercritical ORC at theoretical level (Karellas *et al*., 2012). Most of the studies provide a thermodynamic overview, focusing on the performance potential of such cycles, while others focus on the fluid selection under such conditions (Chen *et al*., 2011). These theoretical studies treat some key components as black boxes, providing few details about their performance and operation at off-design conditions (Li *et al*., 2013), such as the pump and expander, while more focus is given on the heat transfer at supercritical conditions (Lazova *et al*., 2014).

Such aspects, especially in small-scale systems, are very important, in order to evaluate and compare the performance of a supercritical ORC. Here, an experimental study is implemented, testing a small-scale supercritical ORC with a net capacity of 3 kW under various conditions (Kosmadakis *et al*., 2014). The heat input is varied, while the hot source temperature is held constant. The first series of measured data are presented, focusing on the whole engine operation, as well as its capability to reach supercritical operation at some conditions.

Such study is very important, since real test data are presented, and the advantages and potential of such technology can be identified. Also, this study provides some first proof, whether a supercritical ORC can be indeed more efficient than a subcritical cycle, and if the theoretical results can be verified. Focus is also given on the expansion machine, evaluating the converted scroll expander at such conditions.

2. THE INSTALLED ENGINE

A small-scale ORC engine has been designed and constructed. One of its biggest challenges during its construction was the modification of a scroll compressor (manufactured by Copeland, type ZP137KCE-TFD with swept volume 127.15 cm^3/rev, maximum isentropic efficiency 75.2%, and built-in volume ratio of around 2.6 at compressor mode) to operate as scroll expander (reverse operation). A new casing had to be made, while many internal parts have been re-designed for better matching its operation as expander (such as the inlet volume before the fluid enters the steady scroll). The internal design has been optimized and all issues have been resolved, such as to find an efficient way to get the three wirings of the asynchronous motor out of the casing, ensuring at the same time electrical insulation and pressure sealing. An electric brake is connected with a frequency inverter of the expander, in order to control the test conditions and evaluate this expansion machine. This dynamic brake has a 100% duty cycle and is connected to the frequency inverter of the expander. Its main operation is to convert the variable frequency AC power through transistors into DC electricity, which is then dissipated to heat through a bank of appropriate electrical resistances. This method is followed, in order to have the full control of the expander speed, and find its performance maps.

Also, a dedicated evaporator has been developed for this application, following a helical-coil design. The organic fluid is R-404a. The cooling of the ORC engine is accomplished with a cooling water circuit, using a conventional shell and tube heat exchanger. Cold water with temperature around 16 ^{o}C is circulated and drawn from a large water reservoir with capacity of 320 m^3, rejecting the heat of the

ORC engine. The condensation temperature of the organic fluid with this method is around 25 °C (fluctuating according to the engine load).

The ORC engine has been then installed at the laboratory for performance tests under controlled conditions. The heat input is provided by an electric heater and its heat production can be altered covering a large range of its capacity (from 25% of the total heat capacity: 12-48 kW_{th}) by operating different number of electric resistances and switching on/off the heater. For the investigation of the ORC engine, the maximum heat produced by the electric heater is around 48 kW_{th}, while the hot source temperature is held constant and equal to 95 °C. The heat transfer fluid (HTF) is pressurized water at around 2.5 bar at such temperature, and circulated with an inline centrifugal pump, operating at constant speed (2900 rpm). A simplified design of the system installed at the laboratory is depicted in Figure 1, together with the heating and cooling circuits (Kosmadakis *et al.*, 2013b).

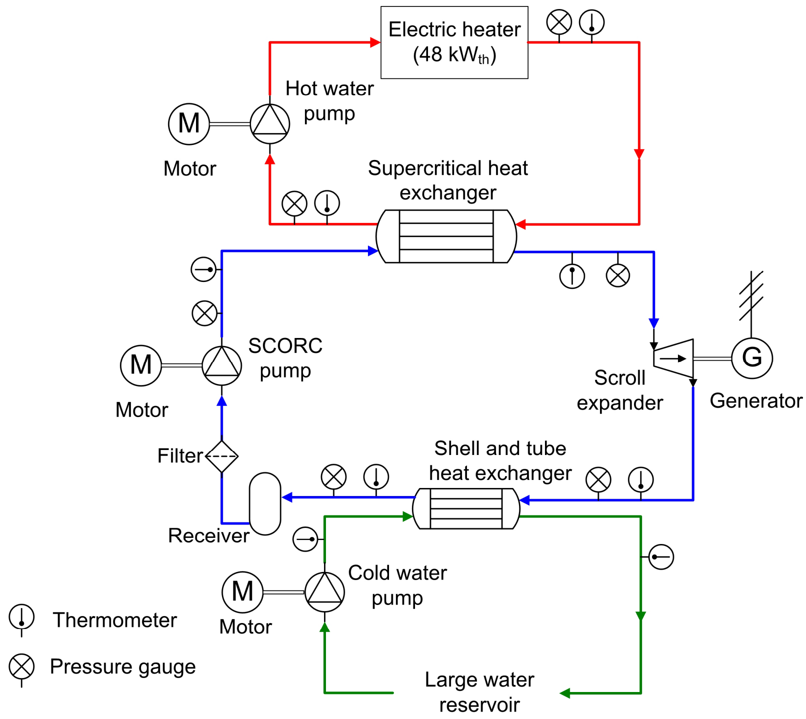

Figure 1: Supercritical ORC design

In Figure 1, the location of the measurement instruments is also depicted at the three circuits (hot water circuit, ORC engine, cold water circuit), in order to measure the key properties and evaluate the performance of this engine at controlled conditions. These instruments are mainly temperature (Pt100 thermocouples, accuracy up to ±0.2 °C) and pressure sensors (pressure transducers, Keller 21Y type, accuracy 1% of the pressure scale), in order to calculate the thermodynamic state of the organic fluid and hot/cold water at each location. With the above uncertainties, the thermodynamic properties are calculated with an accuracy of around 1.2% (Manolakos *et al.*, 2009b). Flow meters are not used, since steady-state conditions are examined, after the engine has reached a balanced operation at each case. The heat input is calculated from the ORC side, since the organic fluid pump is of diaphragm type and has a linear correlation of flow rate/speed, which is provided by the manufacturer (accuracy estimated at 2%). The accuracy of the calculated parameters is given in Table 1 (Manolakos *et al.*, 2009b), with the maximum error concerning the thermal efficiency, since it includes many calculated parameters. Nevertheless, this error is still low and does not influence the relative differences of the results. The installed engine at the laboratory is depicted in Figure 2.

Table 1: Accuracy of calculated parameters

Parameter	Maximum error (%)
Heat input to ORC	2.62

Expander power production	2.62
Pressure ratio	1.40
Expansion efficiency	2.66
Thermal efficiency	3.71

Figure 2: Installed ORC engine. Left: converted hermetic scroll expander. Right: Side view of the ORC engine

3. PERFORMANCE TESTS

The HTF temperature is set constant to 95 ^{o}C, while the heat input is varied. The goal is to examine and evaluate the ORC engine performance at these conditions, while supercritical operation is also attempted, in order to identify if and how much the efficiency can be increased at such conditions.
The pump frequency is altered from 15 Hz up to 50 Hz (from 288 up to 960 rpm) and the expander frequency is regulated from 10 Hz up to 45 Hz (from 580 up to 2610 rpm). By regulating the pump speed, the heat input is varied up to almost 50 kW$_{th}$, which is depicted in Figure 3. The variation of the expander speed has just a minor effect on the absorbed heat (see right hand side of Figure 3).

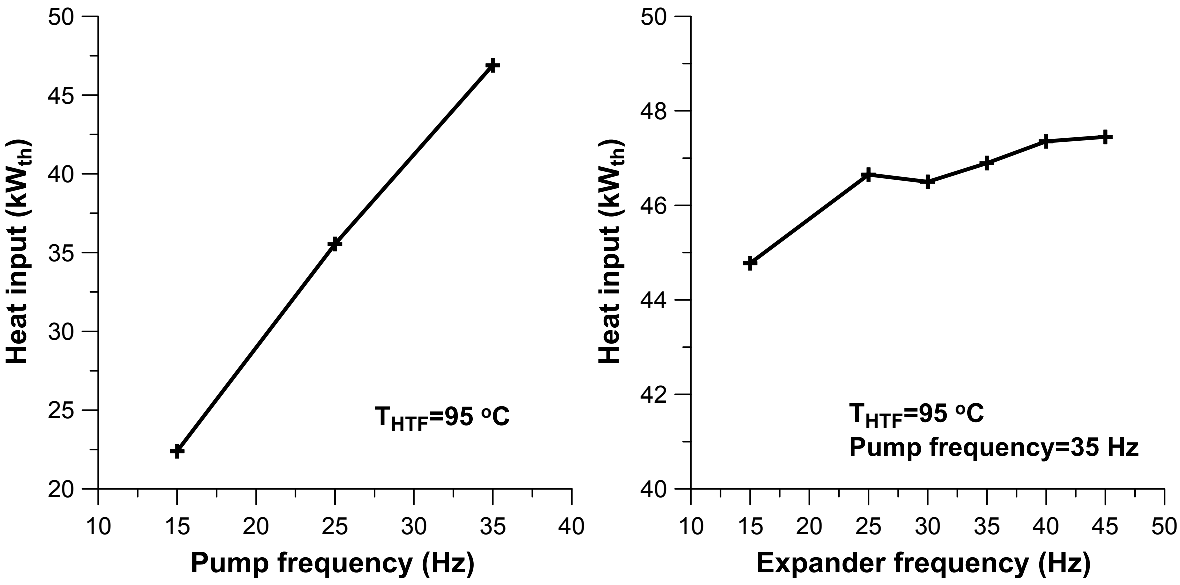

Figure 3: Heat input as a function of the pump and expander frequency

The regulation of the organic fluid pump speed is an effective way to control the heat absorbed by the ORC, while at the same time the flow rate of the organic fluid is adjusted. Therefore, there is the option to keep the speed of the hot water pump constant, simplifying the overall control. In that case, the temperature difference of the HTF would change according to the operating condition (large difference for high heat input). The regulation of the organic fluid pump speed has an important effect on high pressure as well, especially when the expander speed is kept constant, as it will be shown later in this section.
In Figure 4 is shown the expander power production as a function of expander and pump speed, covering a very wide range of operation.

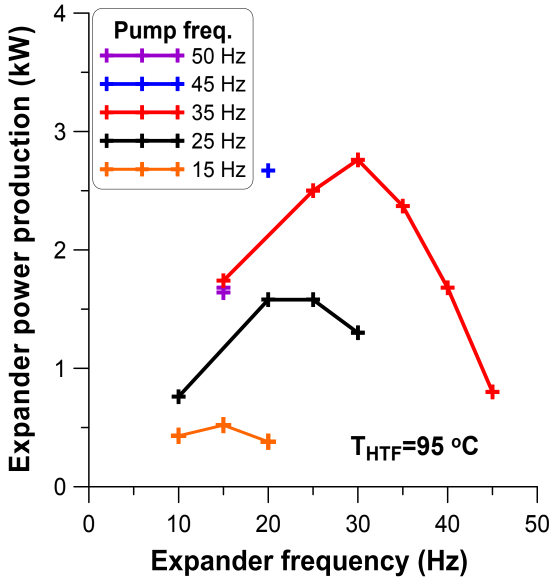

Figure 4: Expander power production as a function of the expander and pump speeds

This power production is actually electricity, since the scroll expander is directly coupled with a three-phase asynchronous motor/generator (capacity of around 10 kW) inside the hermetic casing. This motor operates up to around half of its nominal power, avoiding overheating, due to the absence of cooling (in compressor mode it is cooled by the refrigerant itself).
The power production here reaches 3 kW for the moderate pump frequency of 35 Hz (see Figure 4). For higher pump speed the power production decreases, while additionally more pump power is required. For the pump frequency of 35 Hz and for high expander speed, the power decreases due to lower pressure ratio, while the temperature at the expander outlet is increased. This temperature is shown in Figure 5, together with the expander inlet temperature, which slightly decreases as the expander speed increases.
For high expander speed, the temperature difference across the expander is just 15 ^{o}C, which is very low, leading to a poor power production, as already shown in Figure 4. Also, such high outlet temperature provides a first view of the low process efficiency, since large quantities of heat are rejected in this case at the condenser, having to de-superheat the organic fluid. One interesting solution would be to recover this heat, either by using an internal heat exchanger, although it is not recommended for low-temperature applications, or even use this heat for heating purposes (e.g. operation at CHP mode). But the most efficient way is to further adjust and optimize the engine operation, by avoiding operation at such conditions.
The decrease of pressure and pressure ratio for higher expander speed is presented in Figure 6, where the expander inlet pressure and pressure ratio are shown as a function of the expander frequency for pump frequency of 35 Hz. The low cycle pressure is almost constant and equal to 11-12 bar (higher condensation pressure for high load operation). Therefore, the main effect on the pressure ratio is introduced with the variation of the high cycle pressure.

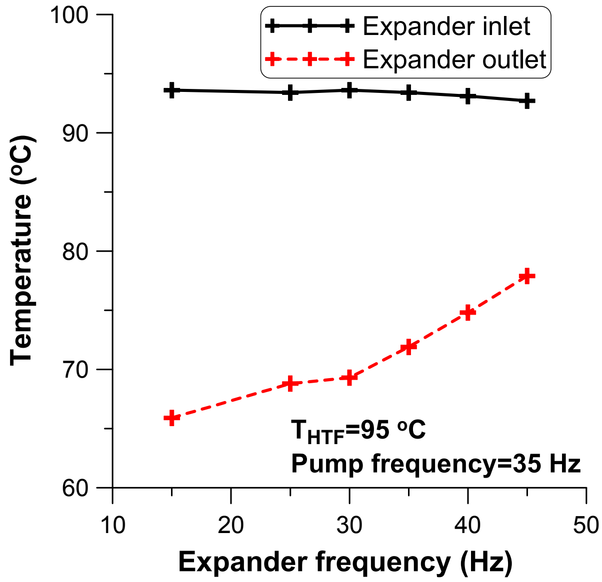

Figure 5: Expander inlet and outlet temperature as a function of the expander speed for pump frequency of 35 Hz

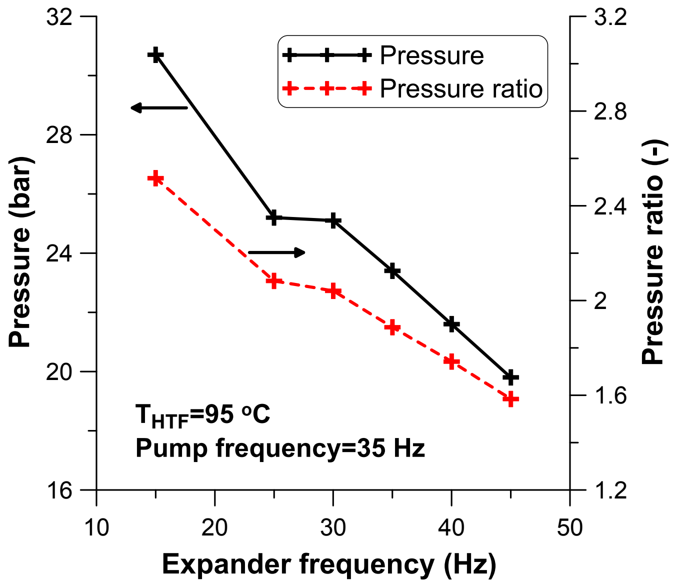

Figure 6: Expander inlet pressure and pressure ratio as a function of the expander speed for pump frequency of 35 Hz

There is a large variation of pressure ratios as the expander speed changes. The maximum pressure ratio is almost 2.6, while the maximum expansion efficiency is observed at a lower expander speed/pressure ratio (equal to around 85%), due to the low electrical efficiency of the asynchronous generator at low speeds, as shown in Figure 7 for three different pump frequencies. It should be mentioned that expansion efficiency is used here as the actual electric power produced divided by the theoretical power produced if the expansion was isentropic. This parameter includes all possible losses (electrical, friction, heat transfer, etc.) and provides a reliable evaluation parameter of all types of expansion machines.

The maximum expansion efficiency is observed for a pressure ratio of 2 (for pump frequency of 35 Hz), which is highly relevant to the built-in volume ratio of the original compressor (Declaye *et al.*, 2013). This pressure ratio value is lower than usual (common values for maximum expansion efficiency are around 3-4), since this compressor is intended for air-conditioning applications, where the pressure/temperature differences are not high. For lower pump speeds and pressure ratios, the expansion efficiency is decreased. For pump frequency of 25 Hz the pressure ratio with the maximum expansion efficiency of around 80% is 1.87, while for pump frequency of 15 Hz is even lower and equal to 1.62. But again for lower pump speeds, the maximum expansion efficiency is high enough and shows good production potential within a narrow range of operating conditions.

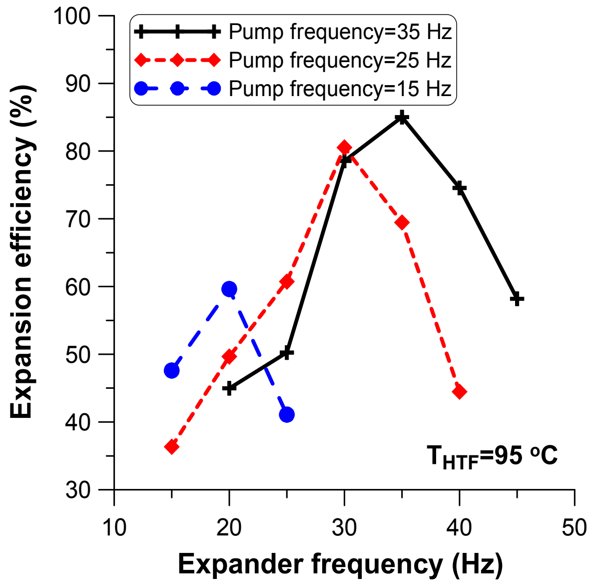

Figure 7: Expansion efficiency as a function of the expander speed for three pump frequencies

The thermal efficiency as a function of pump and expander speeds is depicted in Figure 8. The thermal efficiency is expressed as the net power output (power produced minus the pumping work) divided by the heat input. The maximum values of thermal efficiency are in the range of 5.5%, and are observed for pump frequency of 35 Hz and expander frequency of 30 Hz, which correspond to the conditions, where the maximum expansion efficiency was noticed as well.

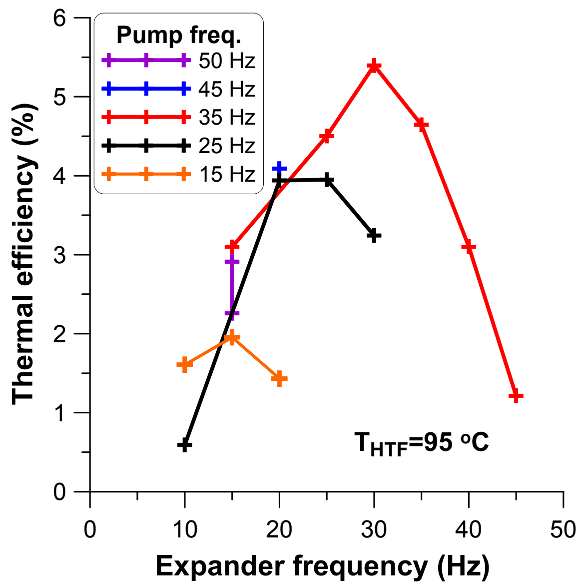

Figure 8: Thermal efficiency as a function of the expander and pump speeds

In all previous tests, supercritical operation was very difficult to be achieved. Although the temperature was over the critical fluid's temperature, the pressure was lower and around 80% of the critical pressure. Such high pressure could be reached only for very large pump speed and low expander speed. One such condition (pump frequency: 50 Hz, expander frequency: 15 Hz) that has been also recorded showed that the high pressure is 40 bar, which is actually the design pressure, and the expander operated with a pressure ratio of 2.64. The thermal efficiency is equal to 2.9%, due to the low expansion efficiency, being equal to 25% (the low expander speed leads to a low electric efficiency of the generator). Nevertheless, this efficiency value is much higher than all the other recorded for such low expander speed (usually in the range of 1.5-2%), and provides a first positive aspect of the supercritical cycle, although such conditions were difficult to be reached.

The next tests included a lower cooling water flow rate, decreasing the cooling capacity and increasing the condensation pressure/temperature. Such conditions are realistic, since the cooling water had low temperature (around 16 ^{o}C), and much higher temperatures are expected to be reached with either air-cooled condenser or evaporative condenser. The pump frequency was high (45 Hz) and the condenser pressure was around 16-17 bar (close to the pump inlet pressure limit). The pressure ratio was not high (equal to 2.2) and the expander frequency varied from 16 up to 30 Hz. The expansion efficiency is observed in Figure 9, where it is shown that the maximum value is almost 80%, which is achieved for moderate expander speed. Moreover, for the supercritical condition (for expander frequency of 16 Hz) the expansion efficiency is low and around 45%, mainly due to the low electrical efficiency of the generator at such conditions.

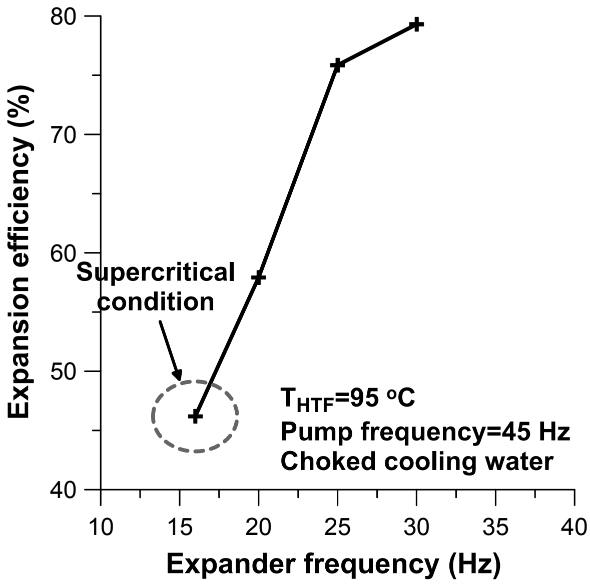

Figure 9: Expansion efficiency as a function of the expander frequency

The thermal efficiency is depicted in Figure 10, where it is shown that the maximum efficiency reached is almost 6% for subcritical operation.

It should be reminded that this high value is reached, when the condensation pressure and temperature are increased, decreasing the power production capability. Therefore, the potential can be even higher than that, which can be achieved with an improved design, especially of the expander, and further testing and development.

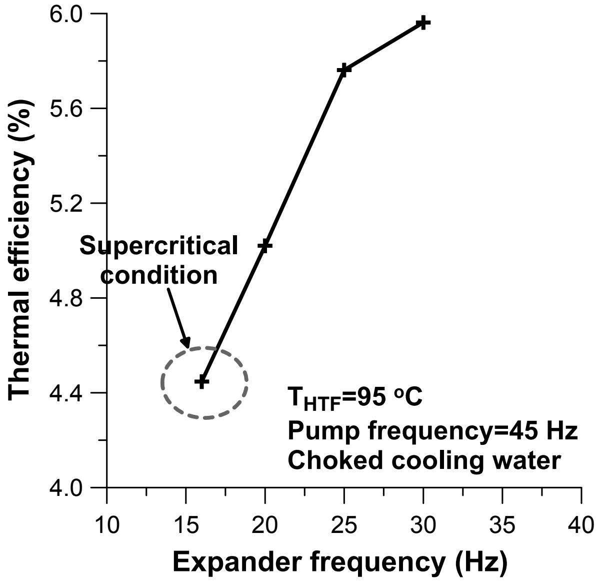

Figure 10: Thermal efficiency as a function of the expander frequency

The efficiency at supercritical condition is lower, mainly due to the low expansion efficiency (see Figure 9). If this condition had a higher expansion efficiency (by including a smaller expander with lower swept volume) and extrapolating the performance curves, then the efficiency would even reach the value of 7%, which is very promising for such small-scale engine at low-temperature and much higher than any other value observed during the current performance tests.

4. CONCLUSIONS

The detailed experimental results of the ORC engine testing at the laboratory have been presented, revealing its performance capability. The tests here concern a constant HTF temperature and equal to 95 ^{o}C with variable heat input. This heat is provided by an electric heater and is controlled with the speed variation of the organic fluid pump. A converted hermetic scroll expander is used for power production, showing increased expansion efficiency at some favorable conditions. Various parameters have been examined, mainly when regulating the expander and pump speed, showing the heat-to-power conversion efficiency of such engine.

Moreover, supercritical operation was difficult to be achieved and only when the cooling water flow rate was decreased, could the engine operate at supercritical conditions and maintain such operation. At these conditions, the expander frequency/speed was low (around 15 Hz), keeping a pressure ratio close to the designed one, but leading to a low expansion efficiency, due to the low electric efficiency of the asynchronous generator at these conditions. If the expansion efficiency could be increased (mainly having to do with the frequency operation and the increase of the electrical efficiency), by using a smaller expander, then the thermal efficiency at supercritical conditions seems to be superior to the one at subcritical ones, and fully exploit the theoretical performance.

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