COB-2019-0157
THE INFLUENCE OF TURBINE ISENTROPIC EFFICIENCY, PRESSURE AND TEMPERATURE ON A SUBCRITICAL ORGANIC RANKINE CYCLE (ORC) WITH SUPERHEAT USING ISENTROPIC FLUIDS

Douglas Ricardo Fumes
Aline Jauch Antonio
Santiago del Rio Oliveira
São Paulo State University (UNESP), School of Engineering, Bauru
douglas.fumes@hotmail.com
aline_jauch@hotmail.com
santiago.oliveira@unesp.br

Abstract. A thermodynamic model was developed for fluid selection in order to obtain an optimal thermal efficiency of an organic Rankine cycle (ORC). The purpose of this study is to perform a thermodynamic analysis, developing a mathematical model to compare the cycle thermal efficiency (CTE) for different isentropic fluids, considering its critical pressure, its critical temperature and the turbine isentropic efficiency. The theoretical model used to obtain the CTE was developed based on thermodynamic laws. The results showed a direct proportionality between the CTE and the turbine isentropic efficiency. The others parameters of interest, such as dimensionless pressure and dimensionless temperature, presented similar influence for each fluid tested. However, different behaviors were observed for different ranges of values. For dimensionless pressure between 0.1 and 0.4 (10% to 40% of the fluid critical pressure), the CTE values are directly proportional to the dimensionless pressure and to the dimensionless temperature. For dimensionless pressure between 0.5 and 0.8 (50% to 80% of the fluid critical pressure), the CTE values are directly proportional to the dimensionless temperature and practically stable to pressure variation. For dimensionless pressures above 0.9 (90% of the fluid critical pressure) the CTE values are inconsistent. Besides, the CTE was also expressed according to the modified Jacob number (Ja_t) *in order to gather the process relevant variables into a single one.*

Keywords: thermal efficiency, organic Rankine cycles, isentropic efficiency, isentropic fluids.

1. INTRODUCTION

Considered a reliable means of energy production, the organic Rankine cycle (ORC) is a technology that allows the conversion of thermal energy in electricity from low temperatures, obtained from the burning of fuels through a heat system (Tartière and Astolfi, 2017). The main characteristic of this cycle regards to the use of an organic fluid, replacing the water, usually used in conventional Rankine cycles. In this process, the fluid also undergoes through a heat operation turning into steam, promoting the energy generation in the turbine. Besides, the use of an organic fluid allows the system to obtain electricity from lower temperatures, when compared to the conventional Rankine cycle, due to the boiling point of these fluids (lower than the water). Pressure and temperature present an important role for ORC's thermal efficiency. Values of critical pressure and critical temperature must be considered for the using of organic fluids and related calculations. In this study, dimensionless pressure and dimensionless temperature were considered as a reference for the calculation development and results exposition. Another important variable considered for the cycle thermal efficiency calculation was the turbine isentropic efficiency.

Lecompte *et al.* (2015) presents an organic Rankine cycle (ORC) as a similar model to the conventional Rankine cycle, where the main difference regards to the working fluid. In ORC technology, an organic working fluid is select, replacing the conventional fluid (water). A careful selection of the working fluid allows the system to convert thermal energy (heat) in electrical energy (electricity) from low temperatures (lower when compared to temperatures used for conventional fluids in conventional Rankine cycles).

Rayegan and Tao (2011) classified the organic fluids used in Rankine cycles as dry, wet and isentropic. This classification regards to the *T-s* diagram format and slope, which presents considerable difference for each type of fluid. The authors also mention the need of a preliminary fluid selection, which must avoid the use of fluids based on chlorine, considered unsafe for the ozone layer and banished by the Montreal protocol. Another important aspect of the ORC regards to the use of alternative heat sources for the heating operation in the evaporator, discarding the need to burn fossil fuels (Javanshir and Sarunac, 2017). According to Mago *et al.* (2008), ORC is an alternative technology that can be

considered an environmental friendly solution for the problems related to thermal pollution and burning of fossil fuels. The authors describe the coverage of the ORC's systems, which includes the recovering of waste heat, geothermal applications, solar plants, biomass and many others.

The basic architecture of an ORC consists in a feed pump, an evaporator, a turbine connected with a generator and a condenser, that is, the same composition of a conventional Rankine cycle. The process begin in the feed pump, where the fluid is compressed and sent to the evaporator, which performs a heat process until the phase change, from liquid to steam. Entering the turbine, an expansion process starts, and after that, the fluid is sent to the condenser, where a new phase change occurs, but this time, transforming steam into liquid, unlike the first transformation. The cycle meets the end when the fluid is sent to the first point, the feed pump, where the process starts over again (Javanshir and Sarunac, 2017).

According to Chen, Xu and Chen (2012) the researches around ORC technology can be classified into five types: (1) modify of the ORC architecture in order to obtain higher thermal efficiencies; (2) fluid selection; (3) replacement of subcritical ORC's for supercritical ORC's; (4) use of zeotropic mixture as working fluids and (5) reach of optimal operating conditions. All these initiatives aim at obtaining higher thermal performance for the cycle. Dai, Wang and Gao (2009) write about the various parameters of influence over the ORC performance. The authors also mentioned the relation between these parameters and its joint influence over the system, comparing it to a multidimensional surface, which can be optimized in order to obtain a better thermal performance.

The main purpose of this paper is to develop a study about the influence of the operation parameters (evaporator pressure and temperature and the isentropic turbine efficiency) in the ORC thermal performance (cycle thermal efficiency and cycle specific net work). The present study also purposes to express the thermal performance according to the joint influence of the system parameters, that is, a single variable that can represent all the parameters of interest considered. Lastly, it is desirable to obtain a relation of the best organic working fluids for each operation parameters combination, in order to achieve a mathematical model for fluid selection according to a set of operation conditions.

2. THERMODYNAMIC MODEL

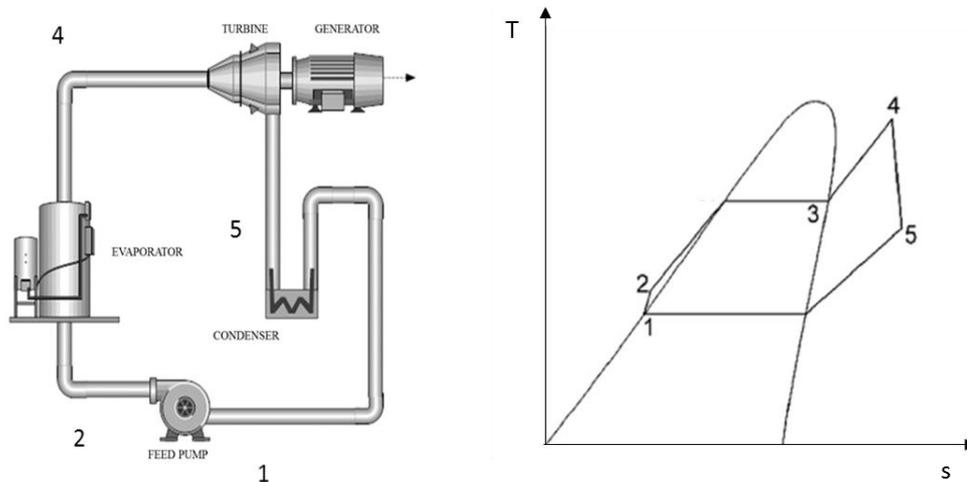A subcritical ORC with superheat can be seen in Fig. 1:

Figure 1. (a) Architecture of a superheated ORC. (b) *T-s* diagram for a superheated ORC. Adapted from: Javanshir and Sarunac (2017).

A mathematical expression for the cycle thermal efficiency was developed based on the model proposed by Javanshir and Sarunac (2017). To reach this model, a development of the thermodynamic first and second laws was performed, considering energy and entropy balances for the evaporator and for the condenser. The following assumptions for the thermodynamic model are: no heat loss due to friction in ducts and equipment, negligible kinetic and potential energies, steady state operation and no pressure drop in ducts and heat exchangers. Based on the assumptions above, from the first law of thermodynamics applied on the evaporator, we obtain:

$$\frac{\dot{Q}_{2,4}}{\dot{m}} = h_4 - h_2 \approx \frac{\dot{Q}_{1,4}}{\dot{m}} = h_4 - h_1 = (h_4 - h_g) + (h_f - h_1) \tag{1}$$

where $\dot{Q}$ is the heat transfer rate, h is the specific enthalpy and $\dot{m}$ is the working fluid mass flow rate. The f index indicates liquid state and the g index indicate gaseous state. Using the latent heat from the evaporation process, we can rewrite Eq. (1) as:

$$\frac{\dot{Q}_{in}}{\dot{m}} = c_{p1,ev}(T_{ev} - T_1) + h_{fg} + c_{p4,ev}(T_4 - T_{ev}) \qquad (2)$$

where c_p is the specific heat at constant pressure and T is the temperature. The index ev indicates the evaporation temperature. From the second law of thermodynamics applied on the evaporator, we obtain:

$$s_4 - s_1 = c_{p1,ev} \ln\left(\frac{T_{ev}}{T_1}\right) + \frac{h_{fg}}{T_{ev}} + c_{p4,ev} \ln\left(\frac{T_4}{T_{ev}}\right) \qquad (3)$$

where s is the specific entropy. From the first law of thermodynamics applied on the condenser, we obtain:

$$\frac{\dot{Q}_{1,5}}{\dot{m}} = h_5 - h_1 \approx \frac{\dot{Q}_{1,4}}{\dot{m}} = h_4 - h_1 \qquad (4)$$

According to Gibbs first equation, we can perform an approach between the condenser heat rate and its specific enthalpy variation, considering negligible pressure drop in the condenser, that is:

$$Tds = dh - vdp \approx dh = \delta\dot{Q}/m \qquad (5)$$

From the second law of thermodynamics applied on the condenser and using Eqs. (4) and (5), we obtain:

$$\frac{\dot{Q}_{out}}{\dot{m}} = T_{eq,c}(s_4 - s_1) = T_{eq,c}\left[c_{p1,ev} \ln\left(\frac{T_{ev}}{T_1}\right) + \frac{h_{fg}}{T_{ev}} + c_{p4,ev} \ln\left(\frac{T_4}{T_{ev}}\right)\right] \qquad (6)$$

where $T_{eq,c}$ is the average temperature between the turbine outlet and the condensation temperature. We can write an expression for the cycle net power as a function of the turbine isentropic efficiency in the following manner:

$$\dot{W}_{cycle} = \dot{m}\left(\frac{\dot{Q}_{in}}{\dot{m}} - \frac{\dot{Q}_{out}}{\dot{m}}\right)\eta_t \qquad (7)$$

where $\dot{W}_{cycle}$ is the cycle net power and η_t is the turbine isentropic efficiency. Substituting Eqs. (2) and (6) in Eq. (7) we obtain:

$$\frac{\dot{W}_{cycle}}{\dot{m}} = \left\{c_{p1,ev}(T_{ev} - T_1) + h_{fg} + c_{p4,ev}(T_4 - T_{ev}) - T_{eq,c}\left[c_{p1,ev} \ln\left(\frac{T_{ev}}{T_1}\right) + \frac{h_{fg}}{T_{ev}} + c_{p4,ev} \ln\left(\frac{T_4}{T_{ev}}\right)\right]\right\}\eta_t \qquad (8)$$

The cycle thermal efficiency is the ratio between the cycle net power and the cycle input heat rate, that is:

$$\eta = \frac{\dot{W}_{cycle}}{\dot{Q}_{in}} \qquad (9)$$

Substituting Eqs. (2) and (6) in Eq. (8) and rearranging we obtain:

$$\eta = \eta_t \frac{\dfrac{c_{p1,ev}(T_{ev} - T_1)}{h_{fg}} + 1 + \dfrac{c_{p4,ev}(T_4 - T_{ev})}{h_{fg}} - T_{eq,c}\left[\dfrac{c_{p1,ev}}{h_{fg}} \ln\left(\dfrac{T_{ev}}{T_1}\right) + \dfrac{1}{T_{ev}} + \dfrac{c_{p4,ev}}{h_{fg}} \ln\left(\dfrac{T_4}{T_{ev}}\right)\right]}{\dfrac{c_{p1,ev}(T_{ev} - T_1)}{h_{fg}} + 1 + \dfrac{c_{p4,ev}(T_4 - T_{ev})}{h_{fg}}} \qquad (10)$$

Wang $et\ al.$ (2013) proposed to express the cycle thermal efficiency according to a variable that denotes the ratio between the sensible heat and the latent heat. This dimensionless parameter is called Jacob Number (Ja), that is:

$$Ja = \frac{\text{sensible heat}}{\text{latent heat}} = \frac{c_{p1,ev}(T_{ev} - T_1)}{h_{fg}} \tag{11}$$

For a superheated cycle, Javanshir and Sarunac (2017) proposed a similar variable called superheat Jacob number (Ja_s). In this case, the authors considered the sensible heat during the superheat process instead of the sensible heat between the beginning of the heating process and the evaporation point, that is:

$$Ja_s = \frac{\text{sensible heat (superheat)}}{\text{latent heat}} = \frac{c_{p4,ev}(T_4 - T_{ev})}{h_{fg}} \tag{12}$$

Javanshir and Sarunac (2017) also purposed the using of the modified Jacob Number (Ja_t) to express the cycle thermal efficiency. This parameter considers both the Jacob number (Ja) and the superheat Jacob number (Ja_s), that is:

$$Ja_t = (Ja + kJa_s)^{1.45} \tag{13}$$

where k is a constant equal to 1 for isentropic fluids and 1.5 for wet and dry ones. For convenience, we can define T_{EC}, a ratio between the evaporation and condensation temperatures of the working fluid, that is:

$$T_{EC} = \frac{T_{evaporation}}{T_{condensation}} = \frac{T_{ev}}{T_1} \tag{14}$$

Using Eqs. (10-12), we can rewrite Eq. (9) in the following manner:

$$\eta = \eta_t \frac{\left\{ Ja + 1 + Ja_s - \frac{T_{eq,c}}{T_1}\left[\frac{Ja}{T_{EC}-1}\ln(T_{EC}) + \frac{1}{T_{EC}} + \left(\frac{Ja_s \ln\left(\frac{T_4}{T_{EC}T_1}\right)}{\frac{T_4}{T_1}T_{EC}} \right) \right] \right\}}{Ja + 1 + Ja_s} \tag{15}$$

3. RESULTS AND DISCUSSION

Javanshir and Sarunac (2017) proposed the use of two parameters for the results presentation, called dimensionless pressure (P_r) and dimensionless temperature (T_r). These two parameters can be easily obtained through the ratio of the maximum value to the critical value, that is:

$$P_r = \frac{P_{max}}{P_{cr}} \tag{16}$$

$$T_r = \frac{T_{max}}{T_{cr}} \tag{17}$$

In this study a range of P_r and T_r was pre-determined for the calculations of the superheating values (temperature and pressure) for each fluid evaluated. For P_r, a range from 0.1 to 1.0 was considered for calculations (it means a range from 10% to 100% of the critical pressure, for each isentropic fluid). For T_r, a range from 0.8 to 1.2 was considered for calculations (80% to 120% of the critical temperature, for each isentropic fluid). A combination between pressure and temperature values was performed to calculate the results of cycle thermal efficiency and cycle specific net work. Thus, it becames possible to analysis the effect of parameters combination on the cycle thermal performance. A condensation temperature of 15 ºC was considered for all calculations. The evaporation temperature was obtained through each fluid saturation point, based on the dimensionless pressure values established, that is, for each evaporation pressure, an evaporation temperature (T_{ev}) was calculated. All calculations were performed through the software Engineering Equation Solver (EES), due to its thermodynamic database available (fluid properties library). Five isentropic fluids were considered for calculations: R11, R12, R123, R141b e R142b. Data about theses fluids can be seen in Tab. 1:

Table 1. Isentropic working fluids properties. Adapted from: Javanshir and Sarunac (2017).

FLUID	P_{cr} (MPa)	T_{cr} (°C)
R11	4.4	197.96
R12	4.13	111.97
R123	3.66	183.68
R141b	4.21	204.35
R142b	4.05	137.11

According to Rayegan and Tao (2012), fluids classified as dry or isentropic are more suitable for ORC systems, due to its thermodynamic state (superheated) after the expansion stage. Thus, there is no generation of fluid droplets at the turbine outlet. In Fig. 2 we can see isocurves of thermal cycle efficiency for several isentropic fluids as a function of a P_r and T_r ranges. For the P_r range from 0.1 to 0.4 (10% to 40% of critical pressure), the CTE values are directly proportional to the dimensionless temperature and to the dimensionless pressure. Still in this range, the influence of the dimensionless temperature on the CTE decreases as the dimensionless pressure increases. Similarly, the influence of the dimensionless pressure on the CTE decreases as the dimensionless temperature increases. For the P_r range from 0.5 to 0.8 (50% to 80% of critical pressure), the CTE values are directly proportional to the dimensionless temperature. In this range, dimensionless shows low influence on the CTE, mainly for values between 0.6 to 0.8 (60% to 80% of critical pressure). The influence of the dimensionless temperature on the CTE increases as the dimensionless pressure increases. Unlike the last mentioned range, the influence of the dimensionless pressure on the CTE efficiency, in this range, seems not be relevant, because it results in a low variation of values of the CTE.

Dimensionless pressure values above 0.9 (90% or superior to critical pressure) resulted in inconsistent CTE, producing values physically incoherent, when compared to the other pressure ranges. For dimensionless temperature over 0.8 (80% of critical temperature), the inconsistency in results increases. However, the CTE values in this range of dimensionless pressure showed a direct proportionality to the dimensionless temperature. Values of dimensionless pressure equal or above 1.0 (100% of critical pressure) presented CTE results physically incoherent for all dimensionless temperatures values. The isentropic turbine efficiency promoted a high influence on the CTE for all fluids considered. The values established for this parameter are between 70% and 90% (conventional values for isentropic turbines). Comparing the results using 70% and 80% of turbine isentropic efficiency, the calculations presented an average increase of 15% of the CTE, when using the second mentioned value, instead of first one. Increasing the isentropic turbine efficiency to 90%, the comparison between the results to the results using 80% presented an average increase of 13% on the CTE. Therefore, the conclusion is an average increase of 14% in the CTE for each 10% of the isentropic turbine efficiency increases, for all isentropic fluids. Lastly, the influence of turbine isentropic efficiency on the CTE does not seems to have a direct relation to the dimensionless pressure or to the dimensionless temperature. However, different combination of all parameters considered as relevant promotes different results of the CTE.

The specific cycle net work presented a behavior very similar to the CTE, although it seems to be more dependent on dimensionless temperature that on dimensionless pressure for almost all operation values. Inconsistent results have been observed for P_r values above 0.9 meeting T_r values above 1.0 for all isentropic fluids considered. The R11 fluid presented a particular behavior, that is, a lower specific cycle net work for all the operation values, when compared to the other isentropic fluids considered. For R12, R141b and R142b fluids, it was obtained a very close specific cycle net work under the same operation conditions. It is also possible to conclude that an increase of the isentropic turbine efficiency values promotes an increase of the specific cycle net work results. Increasing the turbine efficiency from 70% to 80% results in an average increase of 12.5% in cycle specific net work. When the variation promoted in isentropic turbine efficiency regards from 80% to 90%, the average increase in the specific cycle net work results in 11.1%, that is, there is an inverse proportionality between the specific cycle net work and the isentropic turbine efficiency increase. A comparison between the fluids, using the modified Jacob number (Ja_t) shows that the R141b fluid achieved the highest CTE and specific cycle net work for all the Ja_t values, becoming the best isentropic fluid among the fluids considered. The worst result related to the CTE and specific cycle net work regards to the R12 fluid. The results according to the modified Jacob number, Figs. 4 to 6, meets the results obtained and showed in the isocurves, Figs. 2 and 3.

The influence of the temperature and pressure in the CTE and specific cycle net work can be expressed according to a single variable, the modified Jacob number (Ja_t). The isentropic turbine efficiency was considered separately, as a second variable of influence in the mentioned results, but also presenting high importance to the cycle thermal performance. The CTE seems to present a directly proportionality to the modified Jacob number (Ja_t), when the latter regards to values lower than 2. Values of Ja_t greater than 2 results in CTE values practically stable for all the isentropic fluids considered. The variation of isentropic turbine efficiency seems to promote almost the same influence in cycle thermal efficiency for all range of Ja_t, mainly for the latter values greater than 2 (highest cycle thermal efficiencies). The behavior of the specific cycle net work according to the Ja_t follow the same trend, but shows a certain instability for Ja_t values greater than 2. This result replicates itself for all values of the turbine isentropic efficiency, and the latter promotes almost the same influence in specific cycle net work for all range of Ja_t, just like for the CTE results.
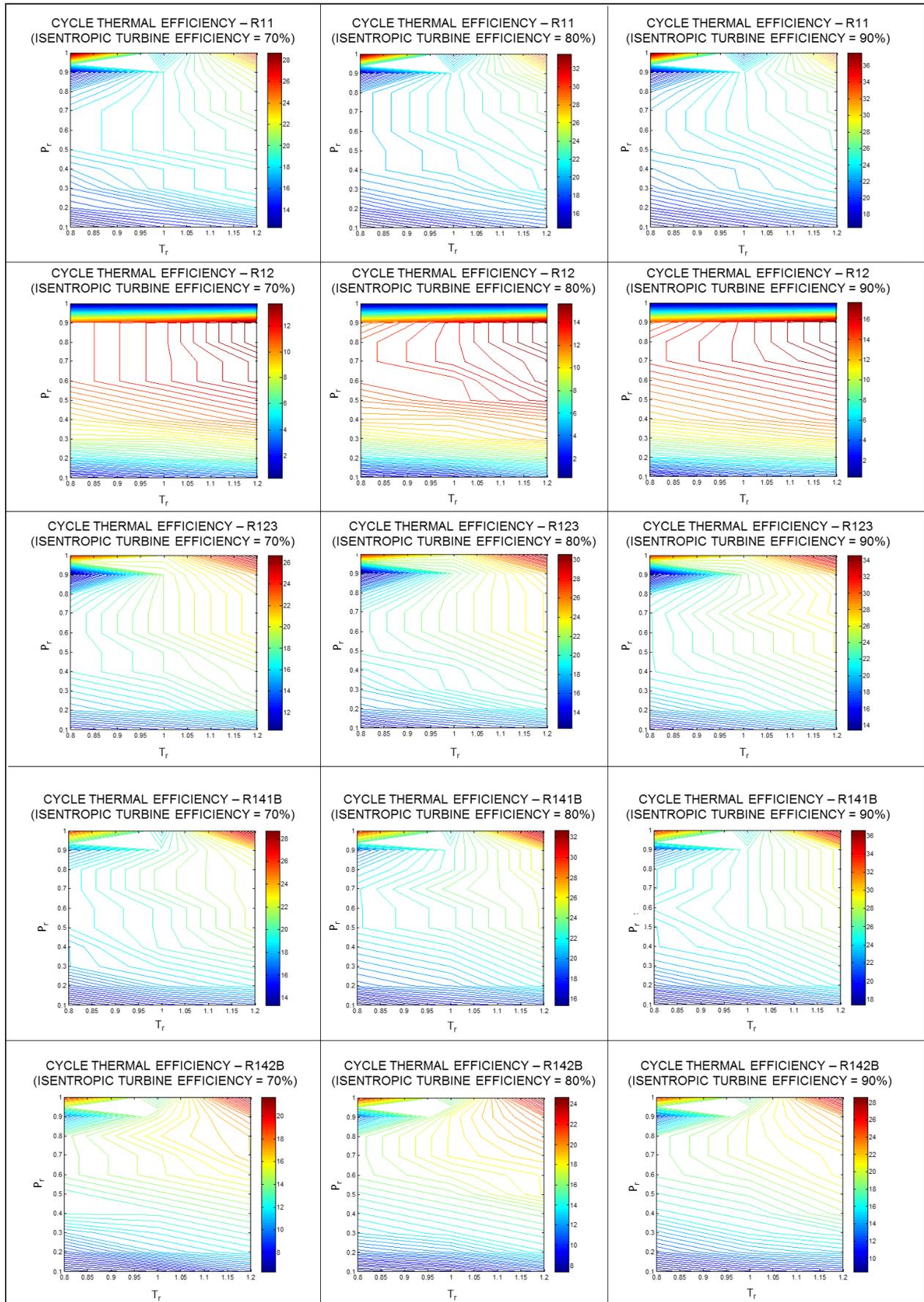
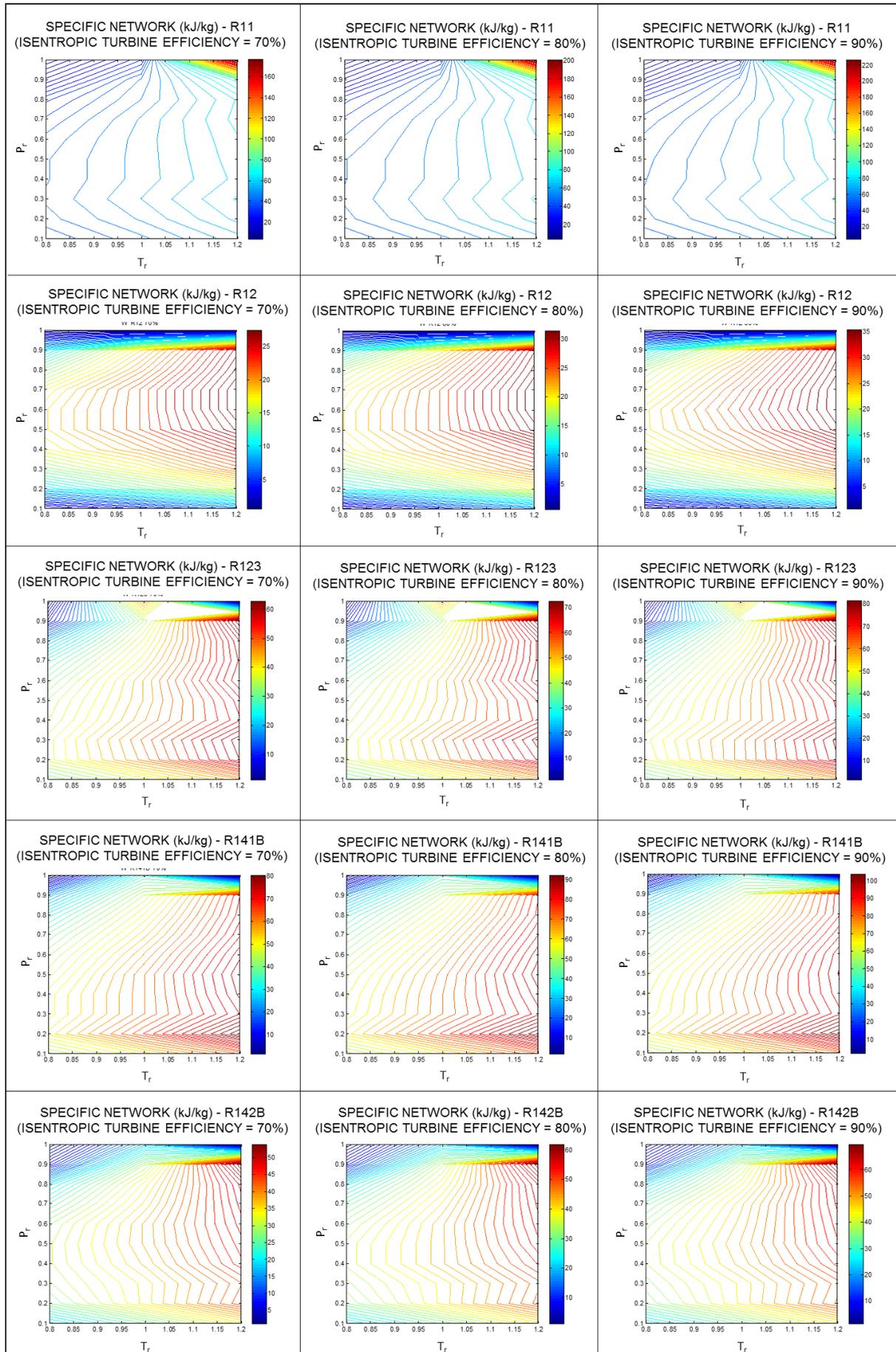Figure 2. Cycle thermal efficiency for isentropic fluids.

Figure 3. Specific cycle net power for isentropic fluids.

Figure 4. Specific cycle net work and cycle thermal efficiency according to the modified Jacob number (Ja$_t$) for isentropic turbine efficiency equal to 70%.

Figure 5. Specific cycle net work and cycle thermal efficiency according to the modified Jacob number (Ja$_t$) for isentropic turbine efficiency equal to 80%.

Figure 6. Specific cycle net work and cycle thermal efficiency according to the modified Jacob number (Ja$_t$) for isentropic turbine efficiency equal to 90%.

4. CONCLUSIONS

The results of this study express the level of influence of the considered operating parameters over the thermal performance of a simple ORC system. The evaporator pressure (expressed through its dimensionless pressure) and the evaporator temperature (expressed through its dimensionless temperature) presents a directly influence in the cycle thermal efficiency and in the specific cycle net work. The cycle thermal efficiency presents a different behavior for each pressure level considered. Dimensionless pressure between 0.1 and 0.4 makes the cycle thermal efficiency directly dependent on the dimensionless pressure and dimensionless temperature. However, a decrease in the dimensionless temperature results in a decrease of the dimensionless pressure influence over the cycle thermal efficiency. For dimensionless pressure between 0.5 and 0.8, the cycle thermal efficiency presents a direct proportionality to the

dimensionless temperature, and for this situation, an increase in dimensionless temperature results in an increase of the dimensionless pressure influence over the cycle thermal efficiency.

Operating values of dimensionless pressure above 1 are not recommend due to the inconsistent results observed in calculations. The cycle specific network is clearly more dependent on dimensionless temperature that on dimensionless pressure for almost all operation values. However, the influence level of each parameter presents a variation according to the other parameter level variation, that is, the influence of the temperature varies for different pressure levels, for instance, just like the influence of the pressure varies for different temperature levels. Thus, a single variable is required to express the global influence of the combination between evaporator pressure and temperature over the cycle thermal performance – the modified Jacob Number (Ja_t). Using this parameter as a reference, it is possible to conclude that the highest cycle thermal efficiencies values are obtained for Ja_t greater than 2, presenting a certain stability for all the following values.

The highest cycle specific network values can be achieved for the same Ja_t range. A consideration about the fluids studied regards to the R141b, which presented the higher cycle thermal efficiency and cycle specific net work for all Ja_t values, becoming the best fluid among the isentropic fluids considered in this work. Lastly, the isentropic turbine efficiency plays an important role for the cycle thermal efficiency and for the cycle specific net work results. Through the calculations performed, it is possible to conclude that an increase of 10% of the isentropic turbine efficiency results in an average increase of 14% in the cycle thermal efficiency and a decrease of 11.8% in the cycle specific net work. However, a variation of the isentropic turbine efficiency does not present a significant influence according to the Ja_t, promoting almost the same influence over the cycle thermal efficiency and the cycle specific net work for all the Ja_t range.

5. REFERENCES

Chen, Q., Xu, J. and Chen. H., 2012. "A new design method for Organic Rankine Cycles with constraint of inlet and outlet heat carrier fluid temperatures coupling with the heat source". *Applied Energy*, Vol. 98, pp. 562-573.

Dai, Y., Wang, J. and Gao. L., 2009. "Parametric optimization and comparative study of organic Rankine cycle (ORC) for low-grade waste heat recovery". *Energy Conversion and Management*, Vol. 50, No. 3, pp. 576-582.

Javanshir, A. and Sarunac, N., 2017. "Thermodynamic analysis of a simple Organic Rankine Cycle". *Energy*, Vol. 118, pp. 85-96,

Lecompte, S., Huisseune, H., van den Broek, M., Vaslambrouck, B., De Paepe, M., 2015. "Review of organic Rankine cycle (ORC) architectures for waste heat recovery". *Renewable and Sustainable Energy Reviews*, Vol. 47, pp. 448-461.

Mago, P., Chamra, L.M., Srinivasan, K. and Somayaji, C., 2008. "An examination of regenerative organic Rankine cycles using dry fluids". *Applied Thermal Engineering*, Vol. 28, Issues 8-9, pp. 998-1007.

Rayegan, R. and Tao, Y.X., 2011. "A procedure to select working fluids to solar organic Rankine cycle (ORCs) review of organic Rankine cycles (ORCs)". *Renewable Energy*, Vol. 36, No. 2, pp. 659-670.

Tartière, T. and Astolfi, M., 2017. "A World Review of the Organic Rankine Cycle Market". *Energy Procedia*, Vol. 129, pp. 2-9.

Wang, D., Ling, X., Peng, H., Liu, L. and Tao, L., 2013. "Efficiency and optimal performance evaluation of organic Rankine cycle for low grade waste heat power generation". *Energy*, Vol. 50, pp. 343-352.

6. RESPONSIBILITY NOTICE

The authors are the only responsible for the printed material included in this paper.