DEVELOPMENT OF AN ANALYTICAL MODEL FOR PREDICTING THE PERFORMANCE OF AN ORGANIC RANKINE CYCLE

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This investigation is a theoretical study of the thermodynamic performance of the organic Rankine cycle. A simplified analytical model is developed regarding the basic organic Rankine cycle by applying reasonable and realistic assumptions. Specifically, the analytical approach for the organic Rankine cycle efficiency uses the low cycle temperature, the high cycle temperature, the superheating degree, the liquid and vapor-specific heat capacities and the fluid latent heat at the high-temperature level. The calculated average deviation of the presented analytical model compared to the detailed thermodynamic one was calculated at 5.03% which is an acceptable value. Additionally, analytical approximations for the efficiency with regression models were created for three different working fluids named n-pentane, toluene and R600. The results of this work can be exploited for the quick and accurate analysis of organic Rankine cycles, and they can be used for the development of optimization models for reducing the computational time of the analysis.

Key words*: thermodynamic cycle, organic Rankine cycle, thermodynamic efficiency, parametric analysis, theoretical investigation*

1. Introduction

The organic Rankine cycle (ORC) is an established technology for the efficient utilization of low-grade energy sources such as solar [1], geothermal [2] biomass and waste heat [3]. They can also be used for complex systems such as thermal Carnot batteries [4] and trigeneration units [5]. Practically, the ORC can operate with various working fluids [6] and therefore achieve a proper exploitation of the low and medium-grade heat sources. Different topologies have been studied in the literature [7] to enhance the ORC performance and also a lot of research has been devoted to improving the performance of the cycle devices (e.g. enhancing the isentropic efficiency of the expanders [8]).

The modeling of ORC is based on the development of analytical thermodynamic models based on energy and mass balances that use thermodynamic properties from libraries and tools [9], [10]. While this approach is reliable, it creates difficulties when the developed programs have to be connected with other simulation tools. Specifically, in dynamic simulation and optimization procedures, there is a need to develop simpler models for the ORC analysis that present simplicity, and they lead to reduced complexity in the simulation and generally reduced computational time. In the literature, a few studies have examined the ORC in detail aiming to develop analytical approaches for its performance. Li [11] and Li et al. [12] performed two studies about the trapezoidal approach for estimating the performance of the ORC. This idea was further expanded by Wang et al. [13] who separated the ORC into three smaller cycles: trilateral, Carnot and Brayton cycles. The same research team performed another work with the trapezoidal design [14] and both studies ([13], [14]) emphasized the working fluid investigation. The separation of the ORC into three cycles was also studied by Scangolatto et al. [15] and they focused on the detailed modeling of the subcomponents. Moreover, another literature study worked with the entropy generation principle for analyzing the ORC [16].

In this direction, the investigation suggests a new idea regarding ORC modeling by using the proper assumptions. The primary version of this idea has been based on the previous literature review, and mainly in Ref. [13], but this paper, it has some important different points aiming to increase the accuracy and to be presented as a simpler one. Also, this work suggests some regression equations for further simplification of the calculation of the ORC thermodynamic performance. Therefore, this work has added value to the existing literature. The results of this study can be used for optimization procedures aiming to minimize the computational cost, as well as they can be used in dynamic simulation models to increase the flexibility of the developed programs. In this way, the design of ORCs and the respective systems can be conducted in a quicker way, something that can lead to better design of sustainable future energy systems. Last but not least, they can be used to better understand the thermodynamic phenomena that play a significant role in the ORC performance.

2. Material and methods

2.1. The suggested theoretical cycle modeling

2.1.1 Cycle description

The organic Rankine cycle is modeled in a simplistic way as depicted in **Figure 1**. The heat input is separated into three parts (1→2), (2→3) and (3→4). The heat input of (1→2) is the preheating, which is an approximately isobaric process with temperature increase, as well as the (3→4) which is the superheating. On the other side, the evaporation (2→3) is performed under the constant high-temperature level (T_H) due to phase change. The heat rejection is conducted in the process of (5→6), (6→7) and (7→1). The temperature is not constant in the process (5→6), while in the others is constant due to phase change. The work production of the ideal cycle can be found by the area of the space 1-2-3-4-5-6-7-1.

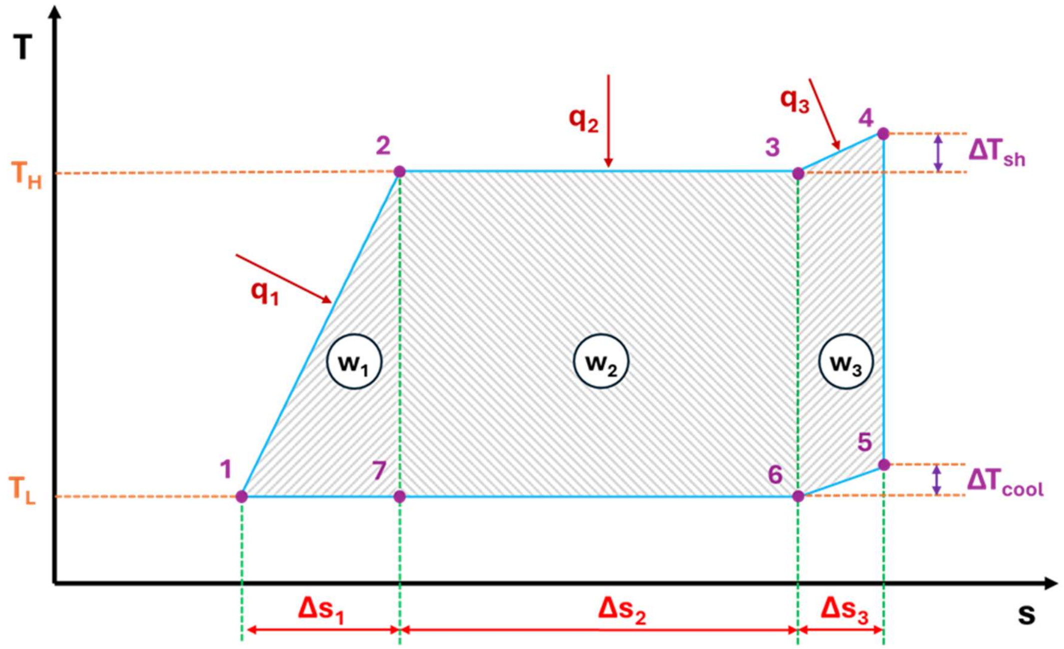


Figure 1. Thermodynamic cycle depiction in temperature (T) – specific entropy (s) diagram

2.1.2 Suggested cycle modeling

The following equations present the mathematical modeling for simulating the ORC cycle efficiency in an analytical way and they comprise the suggested model of this work.

The specific work production (w) is calculated as below:

$$w = w_1 + w_2 + w_3 \tag{1}$$

The specific work (w_1) of the space (1-2-7-1) is calculated as:

$$w_1 = \frac{1}{2} \cdot \Delta s_1 \cdot (T_H - T_L) \tag{2}$$

The specific work (w_2) of the space (2-3-6-7-2) is calculated as:

$$w_2 = \Delta s_2 \cdot (T_H - T_L) \tag{3}$$

The specific work (w_3) of the space (3-4-5-6-3) is calculated as:

$$w_3 = \Delta s_3 \cdot (T_H - T_L) + \frac{1}{2} \cdot \Delta s_3 \cdot \Delta T_{sh} - \frac{1}{2} \cdot \Delta s_3 \cdot \Delta T_{cool} \tag{4}$$

The specific heat input (q) is calculated as below:

$$q = q_1 + q_2 + q_3 \tag{5}$$

The specific heat input in the process (1⟶2) is calculated as:

$$q_1 = \int_{s_1}^{s_2} T(s) \cdot ds = \int_{s_1}^{s_2} \left[T_L + \frac{s - s_1}{s_2 - s_1} \cdot (T_H - T_L) \right] \cdot ds = \frac{1}{2} \cdot \Delta s_1 \cdot (T_L + T_H) \tag{6}$$

The specific heat input in the process (2⟶3) can be calculated as:

$$q_2 = \Delta s_2 \cdot T_H \tag{7}$$

The specific heat input in the process (3→4) can be calculated as:

$$q_3 = \int_{s_3}^{s_4} T(s) \cdot ds = \int_{s_3}^{s_4} \left[T_H + \frac{s-s_3}{s_4-s_3} \cdot \Delta T_{sh}\right] \cdot ds = \frac{1}{2} \cdot \Delta s_3 \cdot \left(T_H + \frac{\Delta T_{sh}}{2}\right) \tag{8}$$

The specific entropy generation in the process (1→2) is calculated as below:

$$\Delta s_1 = c_{p,l} \cdot ln\left(\frac{T_H}{T_L}\right) \tag{9}$$

Practically, this process is approximately the sum of an isentropic compression and an isobaric liquid heat input. Thus, the previous formula is a very good approximation.

The specific entropy generation in the process (2→3) is calculated as below:

$$\Delta s_2 = \frac{r_H}{T_H} \tag{10}$$

The latent heat of the high-temperature $[r_H = r(T_H)]$ is used in the previous formula. Practically, in this analysis, the latent heat is the second heat input (q_2=r_H).

The specific entropy generation in the process (3→4) is calculated as below:

$$\Delta s_3 = c_{p,v} \cdot ln\left(\frac{T_H+\Delta T_{sh}}{T_H}\right) = c_{p,v} \cdot ln\left(1 + \frac{\Delta T_{sh}}{T_H}\right) \approx c_{p,v} \cdot \frac{\Delta T_{sh}}{T_H} \tag{11}$$

Practically, this process is approximately an isobaric gas heat input. Thus, the previous formula is a reliable approximation. Moreover, the ratio $\left(\frac{\Delta T_{sh}}{T_H}\right)$ is relatively low because the superheating is relatively low, thus the previous approximation can be done (with the Taylor series application).

The mean specific heat capacity of the liquid $[c_{p,l} = c_{p,l}(T_m)]$ and mean specific heat capacity of the vapor $[c_{p,v} = c_{p,v}(T_m)]$ can be estimated at the mean operating temperature:

$$T_m = \frac{T_L+T_H}{2} \tag{12}$$

Regarding the temperature difference between the state points (5) and (6), it has to be estimated by using the superheating degree, the low and the high temperatures. The specific entropy variation in the process (3→4) and (5→6) have the same absolute value. Assuming that these processes are approximately isobaric, it can be said:

$$\Delta s_3 = c_{p,v} \cdot ln\left(1 + \frac{\Delta T_{sh}}{T_H}\right) = c_{p,v} \cdot ln\left(1 + \frac{\Delta T_{cool}}{T_L}\right) \tag{13}$$

Therefore, by assuming constant specific heat capacity for the vapor, it is concluded:

$$\Delta T_{cool} = \frac{T_L}{T_H} \cdot \Delta T_{sh} \tag{14}$$

The specific net work production can be written as below, after combining the reported equations:

$$w = \left[\frac{1}{2} \cdot c_{p,l} \cdot T_H \cdot ln\left[\frac{T_H}{T_L}\right] + r_H + c_{p,v} \cdot \Delta T_{sh} + \frac{1}{2} \cdot c_{p,v} \cdot T_H \cdot \left(\frac{\Delta T_{sh}}{T_H}\right)^2\right] \cdot \left[1 - \frac{T_L}{TH}\right] \tag{15}$$

The specific heat input can be written as below, after combining the reported equations:

$$q = c_{p,l} \cdot \left(\frac{T_L+T_H}{2}\right) \cdot ln\left[\frac{T_H}{T_L}\right] + r_H + c_{p,v} \cdot \Delta T_{sh} + \frac{1}{2} \cdot c_{p,v} \cdot T_H \cdot \left(\frac{\Delta T_{sh}}{T_H}\right)^2 \tag{16}$$

The theoretical (or model) ORC efficiency ($\eta_{th,ORC}$) can be written as:

$$\eta_{th,ORC} = \frac{\frac{1}{2}\cdot c_{p,l}\cdot T_H\cdot ln\left[\frac{T_H}{T_L}\right]+r_H+c_{p,v}\cdot \Delta T_{sh}+\frac{1}{2}\cdot c_{p,v}\cdot T_H\cdot\left(\frac{\Delta T_{sh}}{T_H}\right)^2}{c_{p,l}\cdot\left(\frac{T_L+T_H}{2}\right)\cdot ln\left[\frac{T_H}{T_L}\right]+r_H+c_{p,v}\cdot \Delta T_{sh}+\frac{1}{2}\cdot c_{p,v}\cdot T_H\cdot\left(\frac{\Delta T_{sh}}{T_H}\right)^2}\cdot\left[1-\frac{T_L}{TH}\right] \tag{17}$$

The thermal efficiency of the "similar" Carnot cycle is given below, assuming a high temperature (T_H) and avoiding the use of superheating which is generally low.

$$\eta_{carnot} = 1-\frac{T_L}{T_H} \tag{18}$$

Therefore, it can be written as:

$$\eta_{th,ORC} = \frac{\frac{1}{2}\cdot c_{p,l}\cdot T_H\cdot ln\left[\frac{T_H}{T_L}\right]+r_H+c_{p,v}\cdot \Delta T_{sh}+\frac{1}{2}\cdot c_{p,v}\cdot T_H\cdot\left(\frac{\Delta T_{sh}}{T_H}\right)^2}{c_{p,l}\cdot\left(\frac{T_L+T_H}{2}\right)\cdot ln\left[\frac{T_H}{T_L}\right]+r_H+c_{p,v}\cdot \Delta T_{sh}+\frac{1}{2}\cdot c_{p,v}\cdot T_H\cdot\left(\frac{\Delta T_{sh}}{T_H}\right)^2}\cdot\eta_{carnot} \tag{19}$$

or

$$\eta_{th,ORC} = f\cdot\eta_{carnot} \tag{20}$$

The factor (f) shows how the theoretical ORC is close to the respective Carnot cycle, and it is defined as:

$$f = \frac{\frac{1}{2}\cdot c_{p,l}\cdot T_H\cdot ln\left[\frac{T_H}{T_L}\right]+r_H+c_{p,v}\cdot \Delta T_{sh}+\frac{1}{2}\cdot c_{p,v}\cdot T_H\cdot\left(\frac{\Delta T_{sh}}{T_H}\right)^2}{c_{p,l}\cdot\left(\frac{T_L+T_H}{2}\right)\cdot ln\ \frac{T_H}{T_L}]+r_H+c_{p,v}\cdot \Delta T_{sh}+\frac{1}{2}\cdot c_{p,v}\cdot T_H\cdot\left(\frac{\Delta T_{sh}}{T_H}\right)^2} \tag{21}$$

2.1.3 Classical thermodynamic modeling

This subsection includes the main expressions for the typical thermodynamic modeling for the ORC. This modeling is used in the present work to compare the suggested modeling of section 2.1.2 with the typical one. The ORC depiction is given in Figure 2, and it is the simplest possible illustration of this power cycle. Specifically, the basic ORC includes the heat recovery system, the expander, the condenser and the pump. The heat input is given in the heat recovery system, the expander device produces the useful work, the condenser rejects heat to the environment and the pump compresses the fluid in the proper pressure level.

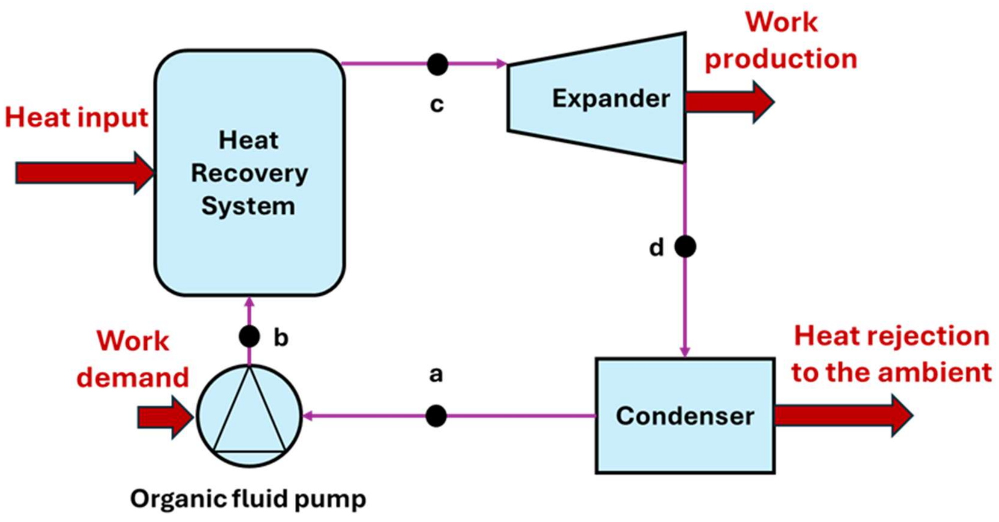


Figure 2. Depiction of the thermodynamic ORC configuration

The specific heat input in the unit is calculated as:

$$q_{in} = h_c - h_b \tag{22}$$

The specific heat rejection to the environment is given as:

$$q_{out} = h_d - h_a \tag{23}$$

The specific work production in the expander device is given as:

$$w_{exp} = h_c - h_d \tag{24}$$

The specific work demand in the organic fluid pump is given as:

$$w_{pump} = h_b - h_a \tag{25}$$

The specific net work production of the cycle is given as:

$$w_{net} = w_{exp} - w_{pump} \tag{26}$$

The ORC thermal efficiency is given as [18]:

$$\eta_{real,ORC} = \frac{w_{net}}{q_{in}} \tag{27}$$

The previous formula describing the ORC efficiency, calculated with the traditional thermodynamic models and the model developed in this work, is assumed to be the benchmark of the accuracy of the suggested modeling approach. More details regarding the modeling can be found in Ref. [19].

2.2. Followed methodology

In this work, the analytical approach of section 2.1.2 is compared with the analytical thermodynamic approach of section 2.1.3. The main analysis is conducted with n-pentane as a working fluid, while extra analysis is conducted for toluene and R600. Different combinations of low temperature, high temperature and superheating degree are investigated. All the examined scenarios concern subcritical operations. The thermodynamic properties are retrieved by the libraries of Engineering Equation Solver [17]; a tool that is used for the thermodynamic analysis that is performed with the modeling approach described in section 2.1.3.

At this point, it is useful to state that the proposed modeling approach faces the cycle as a geometrical depiction in the T-s diagram and the assumptions, as well as the modeling, are based on this representation. For example, the process (1→2) is treated as linear, the work production is calculated by the area of the space of the closed cycle, the subcooling in the condenser inlet was related as a fraction of the superheating, Taylor series approximations are used, as well as the entropy generation is used to estimate the heat inputs. Therefore, this modeling is new and adds extra value to the literature. Other interesting approximations in the literature follow different methodologies. For example, Ref. [13] separates the cycle into three smaller cycles and then tries to combine them to develop an equivalent ORC. These methodologies lead also to analytical solutions but with different assumptions and different final results. The suggested model is verified by using data that has been created with a traditional thermodynamic model written in Engineering Equation Solver. Moreover, a limitation of the present developed model is based on the assumption of a relatively low superheating degree in the expander inlet.

3. Results and discussion

3.1. ORC performance and model verification

This subsection includes the verification results of the suggested analytical model compared to the traditional thermodynamic one. Specifically, **Figure 3** depicts the thermodynamic performance of the ORC with the analytical model and with the thermodynamic analysis which is called real efficiency with n-pentane as the working fluid. Also, the respective Carnot cycle efficiency is depicted in this figure aiming to show that the ORC efficiency is lower, and the curves have similar increasing trends with the increase of the high cycle temperature.

Figure 4 depicts the ORC efficiency according to the present model versus the real efficiency through the thermodynamic procedure with n-pentane as the working medium. The depicted points are all the possible combinations of the studied high temperatures from 350 K up to 460 K with step 10 K, superheating degrees of 0 K, 10 K and 20 K, as well as low cycle temperatures of 290 K, 300 K and 310 K. It was calculated that the mean deviation between the models is 5.03% which is a relatively low and acceptable value. Therefore, it is clear that the suggested analytical model is accurate enough. Also, it is remarkable to state that the deviation is generally constant for the studied high temperatures, which shows good behavior of the model under different operating conditions. Moreover, it is useful to state that the present definition of the Carnot cycle efficiency neglects the superheating degree because it is relatively low, something that makes the present conclusions valid for low superheating degree values.

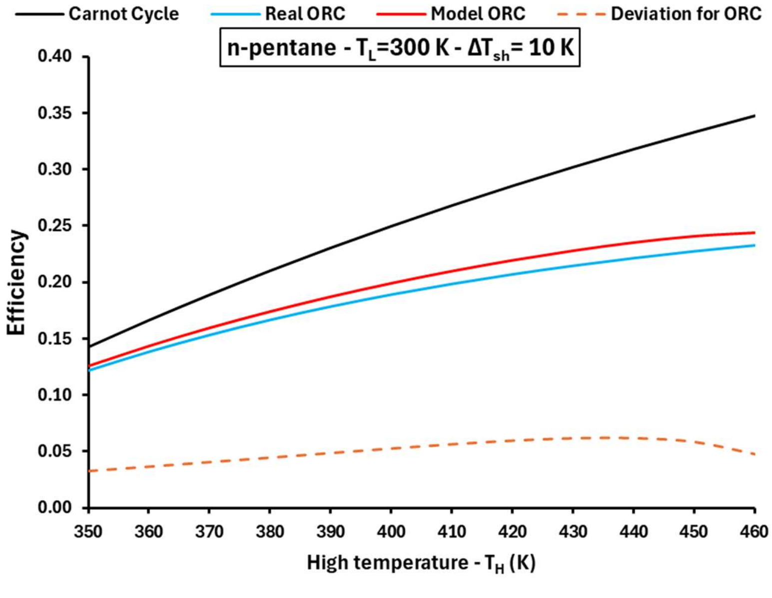


Figure 3. Efficiency cycle for n-pentane with low cycle temperature at 300 K, 10 K superheating degree and variable high cycle temperatures (Deviation regards the Real ORC and the Model ORC)

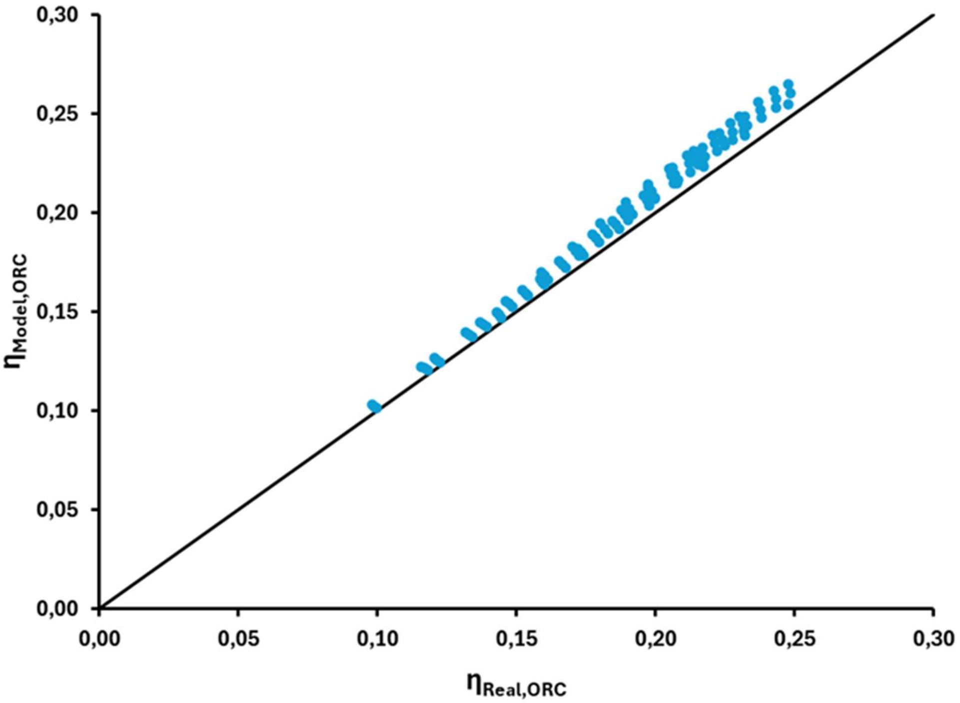
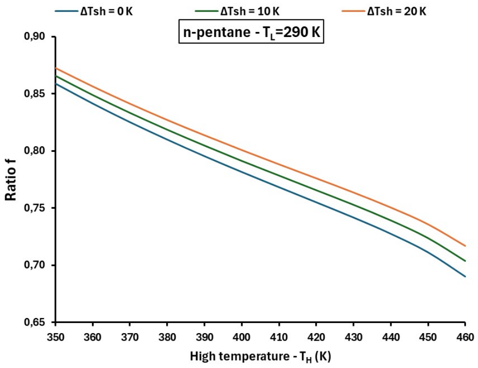


Figure 4. Comparison of the developed ORC model (vertical axis) to the real ORC performance according to the thermodynamic analysis (horizontal axis)

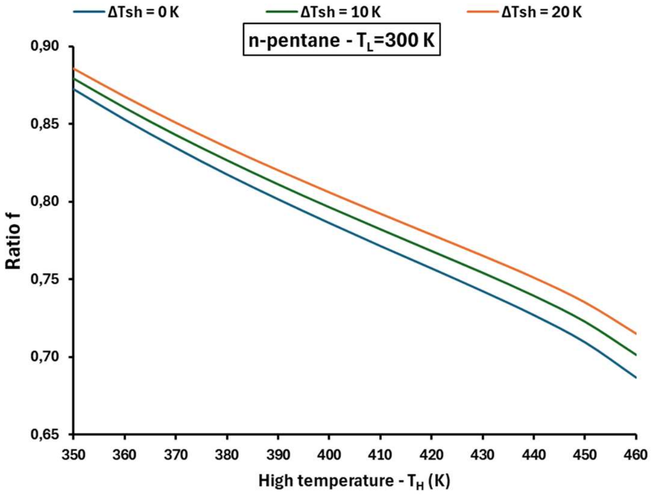
3.2. Variation of the (f) factor

The next stage is the presentation of an example for the application of the present methodology. Figures 5 and 6 show the variation of the (f) factor for different operating conditions with n-pentane as the working medium. This factor is the ratio of the ORC efficiency to the respective Carnot cycle efficiency. In Figure 5, there are three different subfigures that show the results for different low temperatures, while in every figure there are different superheating degrees and variable heat source temperatures. Figure 6 shows results for superheating at 10 K, different low temperatures and variable heat source temperatures. It can be concluded from these figures that the rise of the high temperature decreases the value of (f). Moreover, the rise of the superheating degree in the expander inlet leads to higher values for the ratio (f). The rise of the lower temperature increases the values of (f) in small high-temperatures (T_H), but after T_H=445 K, the reverse result is obtained. Generally, in low values of high temperatures, the (f) is close to 90%, while in increased values of high temperatures it is closer to 70%. This interesting result indicates that the ORC is the most appropriate cycle in applications with relatively low-temperature heat sources.



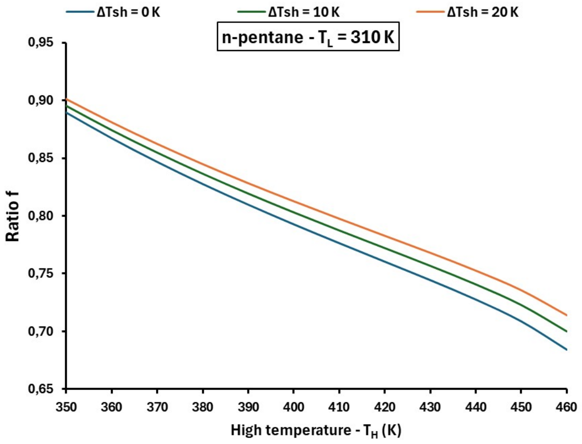
ΔTsh = 0 K
ΔTsh = 10 K
ΔTsh = 20 K
n-pentane - T_L=290 K
Ratio f
0,90
0,85
0,80
0,75
0,70
0,65
350
360
370
380
390
400
410
420
430
440
450
460
High temperature - T_H (K)

(a)



ΔTsh = 0 K
ΔTsh = 10 K
ΔTsh = 20 K
n-pentane - T_L=300 K
Ratio f
0,90
0,85
0,80
0,75
0,70
0,65
350
360
370
380
390
400
410
420
430
440
450
460
High temperature - T_H (K)

(b)



ΔTsh = 0 K
ΔTsh = 10 K
ΔTsh = 20 K
n-pentane - T_L = 310 K
Ratio f
0,95
0,90
0,85
0,80
0,75
0,70
0,65
350
360
370
380
390
400
410
420
430
440
450
460
High temperature - T_H (K)

(c)

Figure 5. Ratio (f) for n-pentane with variable high cycle temperatures, 0-10-20 K superheating degrees and low cycle temperature at (a) 290 K, (b) 300 K, and (c) 310 K

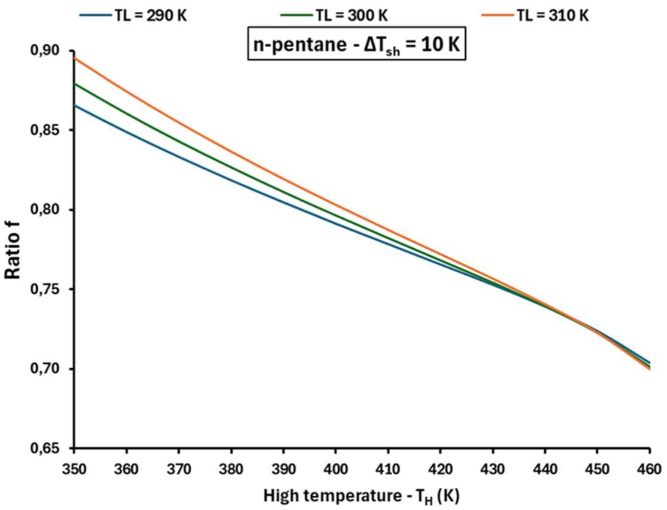


Figure 6. Ratio (f) for n-pentane with low cycle temperatures at 290-300-310 K, 10 K superheating degree and variable high cycle temperatures

3.3. Development of analytical equations

The development of some linear regression equations regarding the factor (f) is something very interesting and it is presented in this subsection. The following approximation equations were developed for three working fluids, and they can be used for the prediction of the cycle's performance.

A) n-pentane ($\mathbf{0.50\%}$ mean deviation and $\boldsymbol{R^2} = \mathbf{99.09\%}$)

$$f = 1.2401 - 0.001554 \cdot T_H + 0.000566 \cdot T_L + 0.001012 \cdot \Delta T_{sh} \tag{28}$$

Applicability for: $350\ K \leq T_H \leq 460\ K,\ 290\ K \leq T_L \leq 310\ K,\ 0\ K \leq \Delta T_{sh} \leq 20\ K$

B) Toluene ($\mathbf{0.40\%}$ mean deviation and $\boldsymbol{R^2} = \mathbf{98.60\%}$)

$$f = 1.0821 - 0.001024 \cdot T_H + 0.000605 \cdot T_L + 0.000496 \cdot \Delta T_{sh} \tag{29}$$

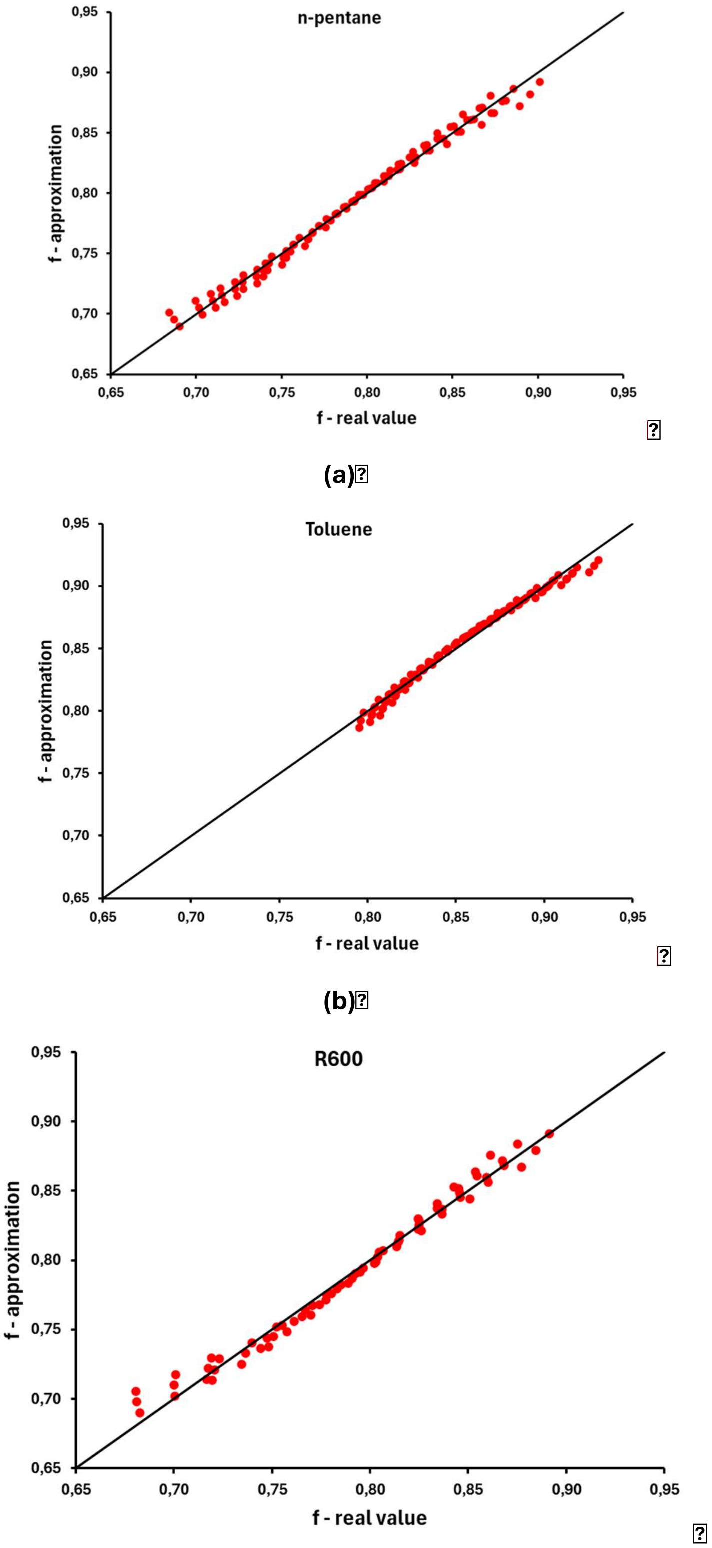
Applicability for: $350\ K \leq T_H \leq 460\ K,\ 290\ K \leq T_L \leq 310\ K,\ 0\ K \leq \Delta T_{sh} \leq 20\ K$

C) R600 ($\mathbf{0.67\%}$ mean deviation and $\boldsymbol{R^2} = \mathbf{98.46\%}$)

$$f = 1.4362 - 0.002312 \cdot T_H + 0.00077 \cdot T_L + 0.00121 \cdot \Delta T_{sh} \tag{30}$$

Applicability for: $350\ K \leq T_H \leq 420\ K,\ 290\ K \leq T_L \leq 310\ K,\ 0\ K \leq \Delta T_{sh} \leq 20\ K$

Figure 7 shows the results regarding the accuracy of the model for the (f) compared to the real analytical model values for n-pentane, toluene and R600 in the **subfigures 7a, 7b and 7c** respectively. It is clear that the suggested formulas have high accuracy, and they can easily be applied to thermodynamic calculations. Therefore, these formulas can be adopted as valid approximations.



(a)

(b)

(c)

Figure 7. Comparison of the approximation model for the (f) ratio for working fluids: (a) n-pentane, (b) toluene, and (c) R600

4. Conclusions

The present work presents an analytical model for predicting the thermodynamic efficiency of a basic ORC. This model uses the low cycle and the high cycle temperatures, the superheating degree, the specific heat capacities of the fluid in liquid and vapor phases, as well as the latent heat in the high-temperature level. This model can determine the efficiency of the cycle with a 5% accuracy compared to the traditional thermodynamic models. Moreover, this model can be used for quick simulations regarding ORCs, for example, in dynamic simulations and in optimization procedures.

Furthermore, in the present work, extra studies were conducted by exploiting the developed model. Specifically, the (f) factor, which is the ratio of the ORC to the respective Carnot cycle was calculated for different operating conditions with n-pentane. Also, linear approximation equations were developed for the (f) for n-pentane, toluene and R600. The results of this investigation can be used for the analysis of systems that include ORC aiming to simplify the calculations and decrease the computational time. The incorporation of such simplified models in optimization procedures is an important application of them. Also, in the future, the present model will be extended to other ORC architectures and to other working fluids.

Nomenclature

$c_{p,l}$	Specific heat capacity of the liquid, kJ/kgK
$c_{p,v}$	Specific heat capacity of the vapor, kJ/kgK
f	Factor for ORC efficiency ratio
h	Specific enthalpy, kJ/kg
q	Specific heat, kJ/kg
q_{in}	Specific heat input in the heat recovery system, kJ/kg
q_{out}	Specific heat rejection in the condenser, kJ/kg
T	Temperature, K
T_H	High cycle temperature, K
T_L	Low cycle temperature, K
T_m	Mean temperature, K
s	Specific entropy, kJ/kgK
w	Specific work, kJ/kg
w_{net}	Net specific work production, kJ/kg
w_{pump}	Specific work demand by the pump, kJ/kg
w_{exp}	Specific work production by the expander, kJ/kg
Greek Symbols	
Δs	Specific entropy variation, kJ/kgK
ΔT_{sh}	Superheating degree in the expander inlet, K
ΔT_{cool}	Superheating degree in the expander outlet, K
η_{carnot}	Carnot cycle efficiency
$\eta_{th,ORC}$	ORC efficiency with the suggested model
$\eta_{real,ORC}$	ORC efficiency with the traditional thermodynamic analysis
Abbreviation	
ORC	Organic Rankine Cycle

Acknowledgment

This research was financially supported by the Ministry of Education, Science and Technological Development of the Republic of Serbia (Contract No. 451- 03-9/2021-14/200109). This research has been supported by the Ministry of Science, Technological Development and Innovation (Contract No. 451-0365/2024-03/200156) and the Faculty of Technical Sciences, University of Novi

Sad through the project “Scientific and Artistic Research Work of Researchers in Teaching and Associate Positions
at the Faculty of Technical Sciences, University of Novi Sad” (No.01-3394/1).

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Paper submitted: 17.12.2024
Paper revised: 24.01.2025
Paper accepted: 31.01.2025