BULETINUL INSTITUTULUI POLITEHNIC DIN IAŞI
Publicat de
Universitatea Tehnică „Gheorghe Asachi" din Iaşi
Volumul 62 (66), Numărul 1, 2016
Secţia
CONSTRUCŢII DE MAŞINI

INFLUENCE OF WORKING FLUID, EXTERNAL AND INTERNAL PARAMETERS ON THE ORGANIC RANKINE CYCLE PERFORMANCE

BY

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Received: May 10, 2015
Accepted for publication: September 25, 2015

Abstract. In this paper the effect of external and internal parameters on the Organic Rankine Cycle (ORC) performance depending on the working fluid is investigated. The pump efficiency , expander efficiency and ambient temperature are the parameters used. The working fluids considered in the present study are Toluene, n-pentane, R600, HFE7100, HFE7000, R11, R141b, R123, R113 and R245fa. In this study the heat source is waste heat applied from diesel engine. The results show that increasing of ambient temperature has bad effect on thermal efficiency and power of ORC system. As well as with increasing the pump and expander efficiency the thermal efficiency and the power of ORC is increased. The largest exergy loss occurs in evaporator, followed by the condenser, expander and pump, so the focus must be on the evaporator section more than the remaining parts. The exergy rate or irreversibility for pump,

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expander and condenser increase slightly with inlet turbine temperature while the increase in exergy rate for evaporator is higher. The results were compared with result by other authors and the agreement was good.

Keywords: Waste heat recovery–Organic Rankine cycle–Performance.

1. Introduction

One of the methods to improve the thermal efficiency of an internal combustion engine is the usage of, Organic Rankine cycles (ORCs) to recover the waste heat. The available heat which is called as waste heat is transferred to the organic working fluid by an evaporator in an ORC, where the organic working fluid changes from a liquid state to a vapour state under a high pressure. Then, the organic working fluid, which has a high enthalpy, is expanded in an expander, and power is generated. Therefore, the evaporator is an important part of the ORC for an engine waste-heat recovery system. Many studies analysing the ORC performances have been conducted recently (Gewald *et al.*, 2012; Kang, 2012; Vélez *et al.*, 2012). Therefore in this study the effects of external and internal parameters on the (ORC) performance are studied. The internal parameters such as pump efficiency and turbine efficiency while ambient temperature is considered as external parameter. The working fluids considered in the present study are Toluene, n-pentane, R600, HFE7100, HFE7000, R11, R141b, R123, R113 and R245fa (He *et al.*, 2012).

2. Mathematical Model

The working fluid leaves the condenser as saturated liquid and then it is pumped from point (1) to point (2) in isentropic process theoretically while to the point (2r) actually as shown in Fig. 1 (Sun and Li, 2011; Wang *et al.*, 2011; Rentizelas *et al.*, 2009). The pump power can be expressed as (Mago *et al.*, 2007):

$$\dot{W}_{P,\,actual} = \frac{\dot{W}_{P,\,ideal}}{\eta_P} = \frac{\dot{m}_{ref}\left(h_1 - h_{2s}\right)}{\eta_P} = \dot{m}_{ref}\left(h_1 - h_{2r}\right) \qquad (1)$$

where: $W_{p,ideal}$ is the ideal power of the pump, $\dot{m}_{ref}$ − the working fluid mass flow rate, η_p − the isentropic efficiency of the pump, h_1 − enthalpy of the working fluid at the inlet and h_{2s} and h_{2r} − the isentropic and actual enthalpies of the working fluid at the outlet of the pump, respectively.

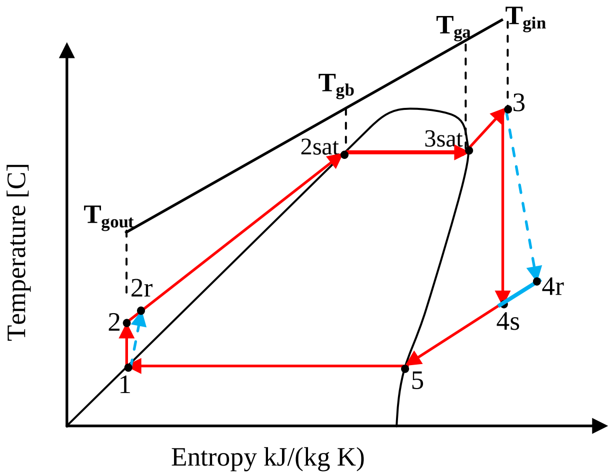


Fig. 1 – T-s diagram for ORC.

The irreversibility rate for uniform flow conditions can be expressed as (Mago *et al*., 2007):

$$\dot{I} = \dot{m}_{ref} T_o \left[\sum_{outlet} s - \sum_{inlet} s + \frac{ds_{system}}{dt} + \sum_k \frac{q_k}{T_k} \right] \tag{2}$$

where: T_k − temperature of each heat source, q_k − heat transferred from each heat source to the working fluid and T_0 − the ambient temperature. In Eq. (2), the contributions of internal or external irreversibilities occurring inside the system or components of the system, a control mass for the system or control volume for each component is taken into account in totality.

Since the system is steady state that is mean ds_{system}/dt=0 then the Eq. (2) reduced to:

$$\dot{I} = \dot{m}_{ref} T_o \left[\sum_{outlet} s - \sum_{inlet} s + \sum_k \frac{q_k}{T_k} \right] \tag{3}$$

Assuming only one inlet and one outlet for a single component, for steady-state steady flow processes the Eq. (3) reduces to:

$$\dot{I} = \dot{m}_{ref} T_o \left[(s_{out} - s_{in}) + \frac{q_k}{T_k} \right] \tag{4}$$

For the pump component the heat transferred to the working fluid (q_k) = 0 substitute in Eq. (4) then the final equation of the exergy destruction rate in the pump (irreversibility rate) is:

$$\dot{I}_p = \dot{m}_{ref} T_o \left[(s_{2r} - s_1) \right] \tag{5}$$

where: s_1 and s_{2r} are the specific entropies of the working fluid at the inlet and exit of the pump for the actual conditions, respectively.

The absorbed energy at the evaporator is converted to useful mechanical work by an expander or a turbine. The turbine power is given by Eq. (6):

$$\dot{W}_t = \dot{W}_{t,ideal}\ \eta_t\ = \dot{m}_{ref}\ (h_3\text{-}h_4)\ \eta_t = \ \dot{m}_{ref}\ (h_3\text{-}h_{4r}) \tag{6}$$

where: $W_{t,ideal}$ is the ideal power of the turbine, η_t − the turbine isentropic efficiency, and h_3 and h_{4r} − the actual enthalpies of the working fluid at the inlet and outlet of the turbine. To calculate the exergy destruction rate in the turbine can be expressed as Eq. (7):

$$\dot{I}_t = \dot{m}_{ref} T_o \left[(s_{4r} - s_3) \right] \tag{7}$$

where: s_3 and s_{4r} are the specific entropies of the working fluid at the inlet and exit of the turbine for the actual conditions, respectively. To determine the exergy destruction rate in the evaporator as shown in Eq. (8):

$$\dot{I}_e = \dot{m}_{ref} T_o \left[(s_3 - s_{2r})\text{-}\frac{(h_3 - h_{2r})}{T_H} \right] \tag{8}$$

Then, the exergy destruction rate in the condenser is calculated from Eq. (9):

$$\dot{I}_c = \dot{m}_{ref} T_o \left[(s_1 - s_{4r})\text{-}\frac{(h_1 - h_{4r})}{T_L} \right] \tag{9}$$

3. Results

Based on the mathematical model presented above a program has been developed in Engineering Equation Solver (F-chart software) for different refrigerants. The aim is to show the influence of ambient temperature, pump and expander efficiency on the performance of the ORC (thermal efficiency and power output) for same heat input. The working fluids considered in the present study are Toluene, n-pentane, R600, HFE7100, HFE7000, R11, R141b, R123, R113 and R245fa .The choosing of these fluids depending on the type of fluid is dry or isentropic. Six working fluid is dry (R113, R600, HFE7100, HFE7000, n-pentane and Toluene) and four working fluid is isentropic (R245fa, R123, R11 and R141b) and leave it the wet fluid because the disadvantages of this type.

3.1. Effect of Ambient Temperature

Effect of ambient temperature on turbine power and thermal efficiency is shown in Figs. 2 and 3 respectively. It can be observed for all fluids that the turbine power and thermal efficiency decrease with the ambient temperature and from this point the increasing in ambient temperature shows a bad effect on the ORC performance. One can notice that the highest power output is obtained for Toluene while the lowest one for HFE7100. The highest efficiency of the ORC is obtained for toluene and R11 while the lowest is obtained for HFE7100.

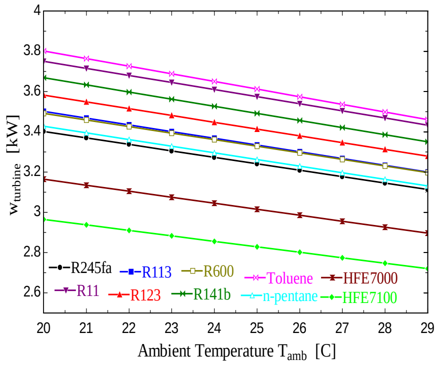


Fig. 2 – Effect of T_{amb} on the Turbine power at Δt_{sup} = 10°C, t_{ev} = 120°C.

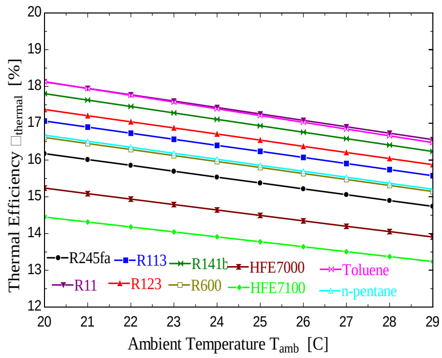


Fig. 3 – Effect of T_{amb} on the thermal efficiency at Δt_{sup} = 10°C, t_{ev} = 120°C.

3.2. Effect of Expander Efficiency and Pump Efficiency

Figs. 4 and 5 show the effect of pump efficiency and expander efficiency on the thermal efficiency and net power at the same time for R245fa. It can be observed from these figures that thermal efficiency and net power increases with the increase in pump and expander efficiency.

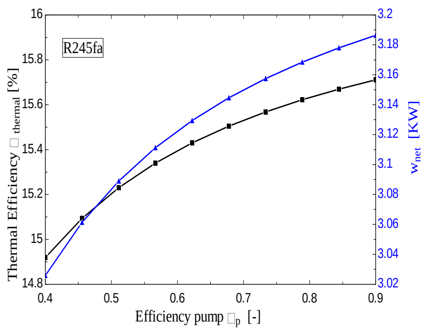


Fig. 4 − Effect of pump efficiency on the thermal efficiency and net power at t_c = 27°C t_{ev} = 120°C.

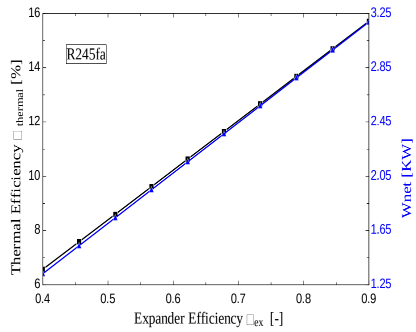


Fig. 5 − Effect of turbine efficiency on the thermal efficiency and net power at t_c = 27°C t_{ev} = 120°C.

From Fig. 6 can be seen the effect of inlet turbine temperature on the irreversibility of components for R245fa as example and from the figure can be seen clearly the largest exergy loss occurs in evaporator, followed by the condenser, expander and pump, therefore it is better to focus on the evaporator section more than the remaining parts. The exergy rate for pump, expander and condenser are increasing slightly while the increase in exergy rate for evaporator is higher. Fig. 7 shows the exergy components for different working fluids.

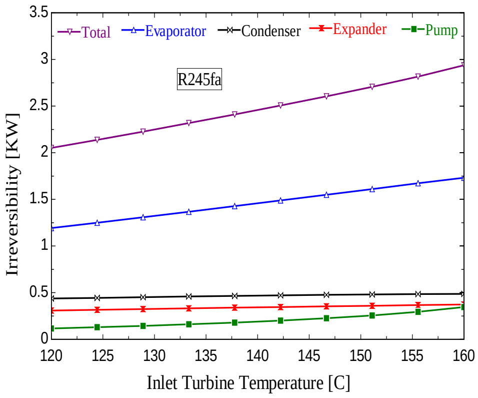


Fig. 6 – Effect of ITT on the components Irreversibility for R245fa.

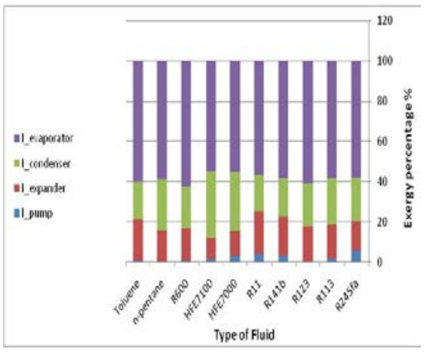
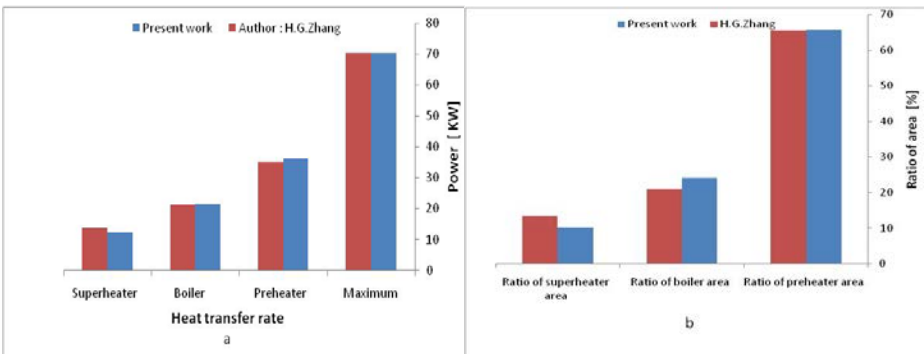


Fig. 7 – Destributed exergy of components Irreversibility for different fluids.

3.3. Comparison Between Present Work and Other Author

To check the validity of the results, it was necessary to compare the results with literature review. The present model and calculation procedure were successfully validated by comparing their results with corresponding results from the literature especially with author (Zhang, 2013) and the comparison were shown in Figs. 8 *a-d*. It can be observed from this figure that paper results are reliable and the agreement is very good.



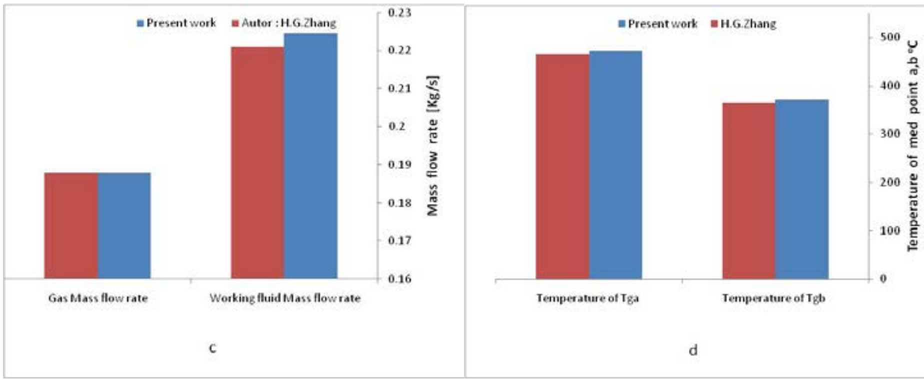


Fig. 8 − Comparison between present work and author: Zhang, 2013.

4. Conclusions

1. The irreversibility of evaporator was the high value while the irreversibility of pump was the smallest and due to that must be focus on the evaporator section more than the remaining parts.

2. It can be observed for all fluids that the turbine power and thermal efficiency decrease with the ambient temperature as example approximately the losses in turbine power is 0.289 KW and 1.43% in thermal efficiency for R245fa because of 9°C increasing in ambient temperature and this is bad effect.

3. From the working fluids under study the Toluene is the best performance and the HFE7100 is the bad performance.

Acknowledgements. One of the authors (M.H.K. Aboaltabooq) acknowledges support from the Ministry of higher education and scientific research of Iraq through grant and the Romanian government through Research grant, "Hybrid micro-cogeneration group of high efficiency equipped with an electronically assisted ORC", 1st Phase Report, 2nd National Plan, Grant Code: PN-II-PT-PCCA-2011-3.2-0059, Grant No.: 75/2012.

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INFLUENŢA AGENTULUI DE LUCRU ŞI A PARAMETRILOR EXTERNI ŞI INTERNI ASUPRA PERFORMANŢEI CICLULUI ORGANIC RANKINE

(Rezumat)

Lucrarea prezintă un studiu termodinamic privind influenţa parametrilor externi (temperatura ambiantă) şi interni (eficienţa izentropică a pompei şi a detentorului) aupra performanţei ciclului organic Rankine (COR) în funcţie de natura şi tipul agentului de lucru. Ciclurile organice Rankine utilizează ca agent termodinamic substanţe de tipul agenţilor frigorifici şi reprezintă o soluţie tehnică avantajoasă pentru recuperarea căldurii reziduale şi creşterea eficienţei energetice a sistemelor termice. Agenţii de lucru consideraţi în această lucrare sunt toluen, n-pentan, R600, HFE7100, HFE7000, R11, R141b, R123, R113 şi R245fa. Sursa de caldură considerată este căldura reziduala provenită de la gazele de ardere ale unui motor cu ardere internă cu aprindere prin comprimare. Studiul termodinamic s-a realizat pe baza unui program elaborat în Engineering Equation Solver (EES). Rezultatele obţinute arată că puterea furnizată şi eficienţa termică a COR scad cu creşterea temperaturii ambiante şi cresc cu creşterea eficienţei pompei şi a detentorului. Un rezultat important al acestei lucrări arată că pierderea de exergie la nivelul vaporizatorului este cea mai mare fiind urmată de cea la nivelul condensatorului, detentorului şi pompei. Acest rezultat arată că în analiza sistemelor COR o atenţie deosebită trebuie acordată vaporizatorului. De asemenea, pierderile de exergie la nivelul pompei, condensatorului şi detentorului cresc mai puţin pronunţat cu creşterea temperaturii agentului frigorific la intrare în detentor decât la nivelul vaporizatorului. Rezultatele obţinute sunt în concordanţă cu alte date disponibile în literatură pentru aplicaţii similare.