

OPTIMUM OPERATION CONDITIONS AND BEHAVIOR OF ORGANIC RANKINE CYCLE SYSTEM UNDER VARIABLE HEAT INPUT WITH CONTROL ON REFRIGERANT MASS FLOW RATE

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In this study, the heat transfer characteristics of an ORC evaporator and condenser applied on diesel engine were analyzed using measured data such as flue gas mass flow rate and flue gas temperature. A mathematical model was developed with regard to the preheater, boiler and the superheater zones of a counter flow evaporator but desuperheater and condenser zones of a counter flow condenser. The performance of the ORC system was evaluated under variable heat input. Three steps applied in this paper starting with determine optimum pressure for working fluid followed by design the ORC system and finally investigate the variable heat input on the ORC system. Can be seen from result that there are optimum evaporation pressure to obtain maximum net power depending on working fluid type. The maximum net power output for R141b (3.361 kW) followed by R123 (3.198 kW) followed by R245fa (2.454 kW) and finally R600a (2.112 kW). The design area of evaporator and condenser proportional to net power output that's mean we need big area for maximum net power output. As well as can be seen from our analysis the total area of evaporator is very close from the total area of condenser for all cases and this is easy for manufacturer. The refrigerant mass flow rate is a very important control parameter on the maximize the net power output as well as to be sure that the vapor at inlet turbine is superheated to avoided dangerous of the droplets in expander. Most cases larger part of heat add go to preheater zone and the remaining part subdivided on boiler and superheater zones. The preheater heat input for R123 it is the largest between working fluids but for R245fa it is smallest that's mean we need more heat to convert R123 from liquid state to saturation liquid but need less heat for R245fa. In the same way we need more heat to convert R141b from saturation liquid to saturation vapor but need less heat for R600a. Finally we need more heat to convert R600a from saturation vapor to be superheated but need less heat for R245fa.

Keywords: Internal combustion engine, heat recovery, organic Rankine cycle.

1. Introduction

The amount of fuel combustion energy which is converted to useful work to drive vehicle or to produce electricity in internal combustion engine or generator about 30% and the remainder energy is consider as a waste energy distributed in

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many sectors such as exhaust gas, coolant system, and convection as well as radiation from the engine block. About 40% from waste heat is dissipated by engine exhaust gases [1]. If just this portion of waste heat could be recovering it, the energy efficiency will be enhanced and the fuel consumption is decreased. Organic Rankine Cycle (ORC) could be used to recover low-grade waste heat from engine exhaust gas [2-10].

The objective of this paper is to create and develop the mathematical model for ORC combining with ICE in steady state condition to design ORC system depending on the optimum evaporation pressure for maximum power of ORC under variable heat input with control on refrigerant mass flow rate. The design including calculate the area and length of tube of evaporator and condenser as well as modelling the pump and expander. Variable heat input meaning that different heat input to the ORC system.

The mathematical model is simulated by a code developed using Engineering Equation Solver (EES). Results of simulations perform with:

- (1) Different of flue temperature gas at evaporator inlet.
- (2) Different inlet turbine temperature (superheating degree).
- (3) Different refrigerant mass flow rate.
- (4) Different load of engine (50 %, 75% and 100%).
- (5) Different working fluids (dry and isentropic type).

The strategy aims to adjust the effect of many factors in order to achieve maximum output power from expander such as refrigerant mass flow rate as a control parameter as well as maximum efficiency after design ORC.

2. System description

In this paragraph the description of the overall system which is used in this paper is presented. The system description is divided in to two parts the diesel engine and organic Rankine cycle description.

2.1 Diesel engine description

In this study, a commercial engine applied in vehicles was used for heat recovery. This is a 3.3 litters (0.83 litter/cylinder), compression ignition, and four-cylinder engine. The studies were performed with different engine load (50%, 75% and 100%) and engine speed about 1500 rpm that consumes diesel fuel. The important parameter in the engine is the temperature of exhaust gases which is measures at the end of silencer is between (300 – 480) ° C depending on the load and the second parameter is water jacket temperature inlet which is between (71 – 74) ° C while water jacket temperature outlet (33 -54) ° C for load (50 % to 100 %). The two major primary sources of waste heat from an ICE are the engine exhaust (medium-grade) and engine coolant (low-grade). The technical

characteristics and the fuel specifications of engine were presented in table (1) [11].

Table 1

Main engine characteristics [11]

Parameter	Specification	Units
Engine type (model)	4-Cylinder	
Power	37.7	kW
Piston stroke, S	110	mm
Compression ratio	18:1	-
Fuel LHV	43	MJ/kg
Exhaust gas temperature	213-477	°C
Cylinder diameter, D	98	mm
Rotation speed	1500	rot/min
Flue gas mass flow rate	192.2-192.6	kg/h
Water jacket flow rate	327.6-1105.5	kg/h

2.2 Organic Rankine cycle description

As shown in Figure 1, the ORC heat recovery power plant used in this report is composed of an expander, condenser, an evaporator (heat recovery unit), a working fluid pump and other auxiliary equipments. Heat from heat source such as waste heat from internal combustion engine is pumped into evaporator (process 1-2) where refrigerant is evaporated, then delivered to expander inlet (process 2-3). The refrigerant drives the expander to generate electricity (process 3-4) along with pressure and temperature decreasing. The low pressure and temperature refrigerant is cooled to refrigerant liquid when passing through the condenser (process 4-1) and collected into a receiver. The working fluid pump lifts the liquid refrigerant back into evaporator (process 1-2) to absorb the heat again then the above process repeats. T-s diagram of ORC is shown in Figure 2.

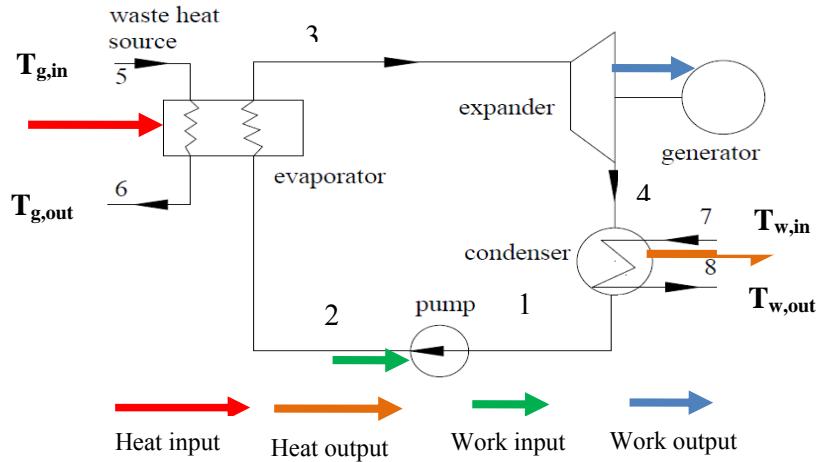


Fig. 1. Schematic diagram of the ORC system.

3. Mathematical modelling for ORC system

Before create the mathematical model of this system to simplify the analysis, some general assumptions are formulated as follows:

- 1) Steady-state and steady-flow condition.
- 2) No pressure drops in heat exchangers and connecting pipes.
- 3) The external heat loss from pipes and equipments are negligible and the exhaust gas heat carrier is non-condensable and non-corrosive during the heat transfer process. The input data used in analysis is explained in table 2.

The mathematical model is a complex work and details regarding it are available in [12]. The present paper focuses on the most important and interesting results obtained by applying the mathematical model for given experimental data.

Table 2

Main input data for code.

Parameter	Specification	Value	Units
Flue gas mass flow rate (100 % Load)	m_g	0.053 [2]	kg/s
Flue gas mass flow rate (75 % Load)	m_g	0.0494 [2]	kg/s
Flue gas mass flow rate (50 % Load)	m_g	0.0475 [2]	kg/s
Flue gas temperature (100 % Load)	T_{gin}	477 [2]	°C
Flue gas temperature (75 % Load)	T_{gin}	403 [2]	°C
Flue gas temperature (50 % Load)	T_{gin}	327 [2]	°C
Ambient temperature	T_{amb}	20	°C
Inlet temperature of cooling system	T_{win}	20	°C
Outlet temperature of cooling system	T_{wout}	30	°C
Pump efficiency	η_p	0.8	—
Expander efficiency	η_s	0.7	—
Inside diameter of inner heat exchanger tube	d_i	0.03	m
Outside diameter of inner heat exchanger tube	d_o	0.04	m
Outside diameter of outer heat exchanger tube	D	0.06	m

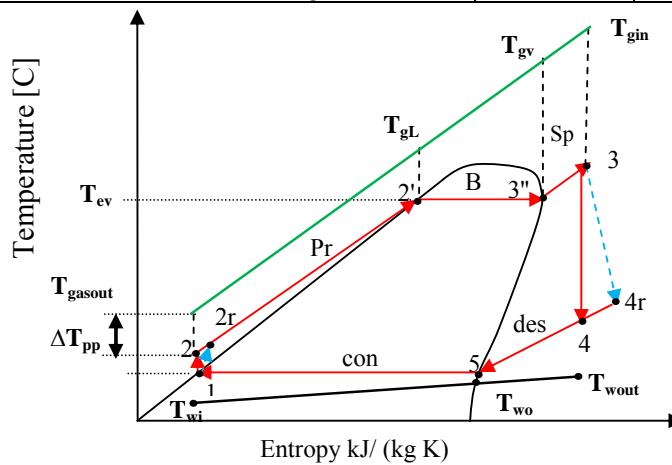


Fig. 2. T-S diagram of the ORC system

4. Result

After create and developed model for our four cases the code will be built in Engineering Equation Solver (EES) program [13] and the result will divided into three steps as presented below.

4.1 Optimum evaporator pressure

First of all, can be start with determine the optimum evaporator pressure. The net power output increases first and then reaches maximum value and then decreases for each working fluids. Figure 3 shows the variations of net power output with various evaporation pressures for R245fa as isentropic fluid for different superheating temperatures. It is obvious that as the evaporation pressure increases, the net power output for R245fa increase to reach maximum and then decreases for different superheating temperatures. There are optimum evaporation pressures for different superheating temperatures and there is maximum net power from these values corresponding to a superheating degree of 30°C. There are optimum evaporation pressure and optimum superheating temperature which give us maximum net power.

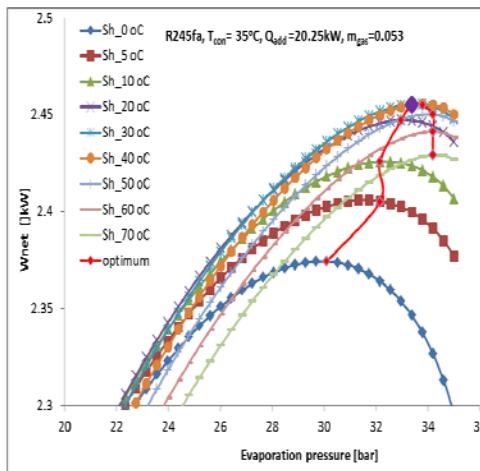


Fig. 3. Effect of evaporation pressure on net power for different superheating temperature for R245fa.

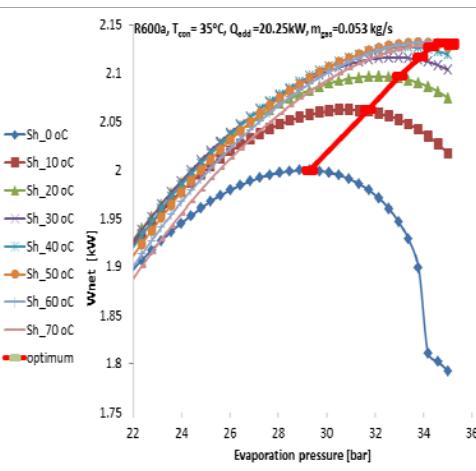


Fig. 4. Effect of evaporation pressure on netpower for different superheating temperature for R600a.

Also for dry working fluid type as shown in Fig. 4 can be seen the variations of net power output with various evaporation pressures for R600a for different superheating temperatures. It is obvious also that as the evaporation pressure increases, the net power output for R600a increase to reach maximum and then

decreases for different superheating temperatures. There are optimum evaporation pressures for different superheating temperatures and there is maximum net power corresponding to these values at a superheating degree of 30°C.

Also Figs. 5 and 6 show us the variations of net power output with various evaporation pressures for R123 and R141b respectively for different superheating temperatures.

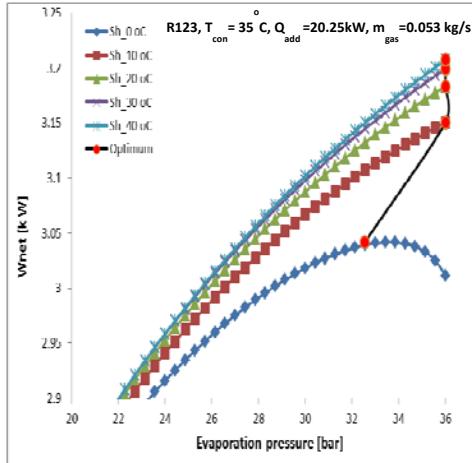


Fig. 5. Effect of evaporation pressure on net power for different superheating temperature for R123.

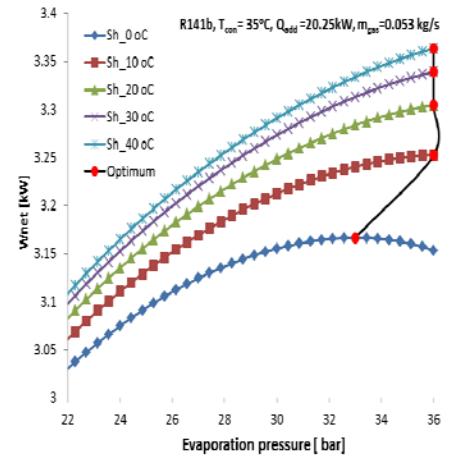


Fig. 6. Effect of evaporation pressure on net power for different superheating temperature for R141b.

After the optimum evaporation pressure is determined can be move to the next step to design organic Rankine cycle system to determine the area of evaporator and condenser as well as the length of tube for evaporator and condenser and all specifications about ORC system.

4.2 Design of ORC system

Four cases used in present paper for two working fluid types as shown in table 3 as follows:

1. Using R245fa with superheating degree 30°C.
2. Using R141b with superheating degree 40°C.
3. Using R123 with superheating degree 30°C.
4. Using R600a with superheating 30°C.

Table 3

Four cases for two types of working fluids.

Case	Fluid	P _{optimum} (bar)	T _{ev} (C)	ΔT _{gpp} (C)	Q _{add} (kW)	W _{net} (kW)	η _{th} (%)	T _{gasin} (C)	T _{gasout} (C)	m _f (kg/s)
1	R245fa Isentropic	33.37	149.3	102.2	20.24	2.454	12.13	477	140.3	0.0659
2	R141b Isentropic	36	193.9	103.9	20.24	3.361	16.61	477	140.3	0.05613
3	R123 Dry	36	182.5	101.2	20.24	3.198	15.8	477	140.3	0.07274
4	R600a Dry	34.71	130.6	101.4	20.24	2.112	10.44	477	140.3	0.04086

Table 4 shows values of preheater, boiler, superheater, desuperheater, condenser and total area of evaporator and condenser for all cases as well as length of condenser and evaporator.

Table 4

Design Area and length of evaporator and condenser.

Case	Total Area A _{evaporator} (m ²)	Pre-heater Area (m ²)	Boiler Area (m ²)	Super-heater Area (m ²)	Total Area A _{condenser} (m ²)	Desuperheater Area (m ²)	Condenser Area (m ²)	L _{ev} (m)	L _{con} (m)
1	2.189	1.687	0.2365	0.2657	2.458	0.3319	2.126	17.6	19.57
2	2.731	2.039	0.3653	0.3271	2.444	0.2689	2.175	21.75	19.46
3	2.406	1.973	0.1347	0.2979	2.513	0.2969	2.216	19.15	20
4	2.03	1.544	0.1014	0.3848	2.417	0.3762	2.095	16.16	19.67

Table 5 shows values of preheater, boiler, superheater, desuperheater, condenser and total heat input and heat rejected for all cases as well as temperature of exhaust gases inside evaporator.

Table 5

Heat balance of evaporator and condenser.

Case	Q _{con} (kW)	Q _{des} (kW)	Q _{rej} (kW)	Q _{sp} (kW)	Q _b (kW)	Q _{pr} (kW)	Q _{add} (kW)	T _{gL} (C)	T _{gv} (C)	m _w (kg/s)
1	12.24	5.468	17.71	4.218	3.613	12.4	20.24	350.3	409.3	0.4231
2	12.46	4.343	16.81	3.801	4.462	11.97	20.24	343.2	416	0.4015
3	12.18	4.821	17	4.353	1.885	14	20.24	376.4	407.1	0.4061
4	12.66	5.357	18.02	5.991	1.589	12.65	20.24	354.4	380.5	0.4304

4.3 Variable heat input

In the current step will be use a data from previous step as example area of evaporator, condenser, refrigerant mass flow rate and all data about ORC but will be change the heat input to the evaporator and can be seen the reaction of ORC system according to variable heat input.

4.3.1 Refrigerant mass flow rate Control

After finding the optimal components for the ORC, the next step is to maximize the net power output in part load regime in other words when use variable heat input to the evaporator with same ORC operation conditions as example area of evaporator and condenser as well as refrigerant mass flow rate and cooling mass flow rate the reaction of this variations is appear on the point 3 Point 3 will be shift in to new location. The point 3 is very important for test the working fluid in superheated zone or in liquid zone toward inlet to the expander. We can see clearly from Figures (7a to 7e) the stages of control for refrigerant mass flow rate on the point 3 to obtain on maximum net power output for ORC in all cases. The variable that can be controlled is the mass flow rate of refrigerant aiming to maximize the net power output in each case.

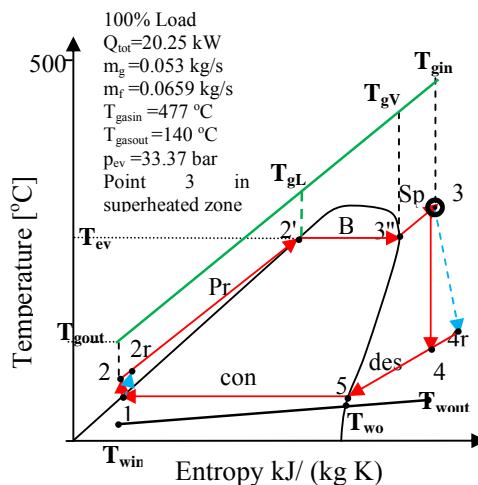


Fig. 7a. T-S diagram for R245fa

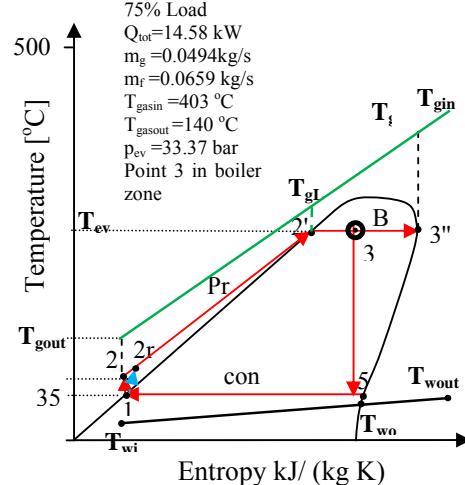


Fig. 7b. T-S diagram for R245fa

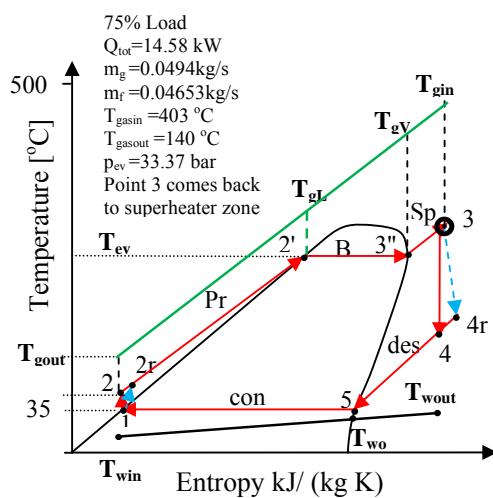


Fig. 7c. T-S diagram for R245fa

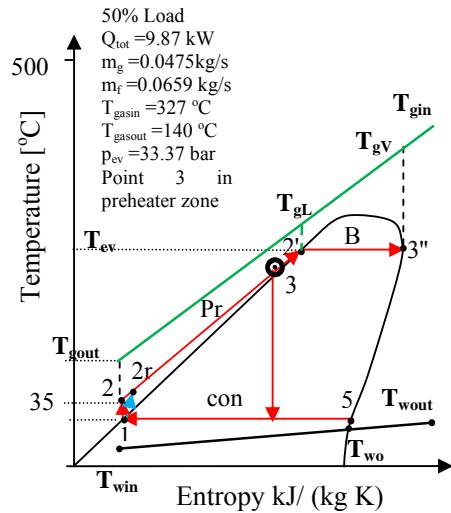


Fig. 7d. T-S diagram for R245fa

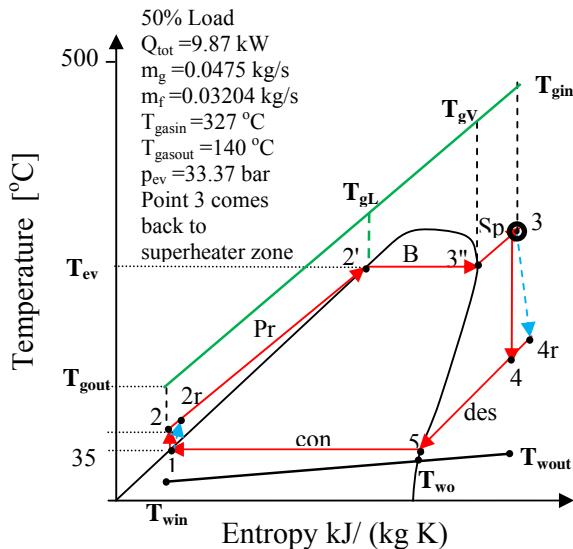


Fig. 7e. T-S diagram for R245fa

4.3.2 Optimum Refrigerant mass flow rate

Figures 8-13 show us the optimum value of refrigerant mass flow rate for R123 and R245fa to obtain maximum net power output from ORC system under variable heat input (variable engine load).

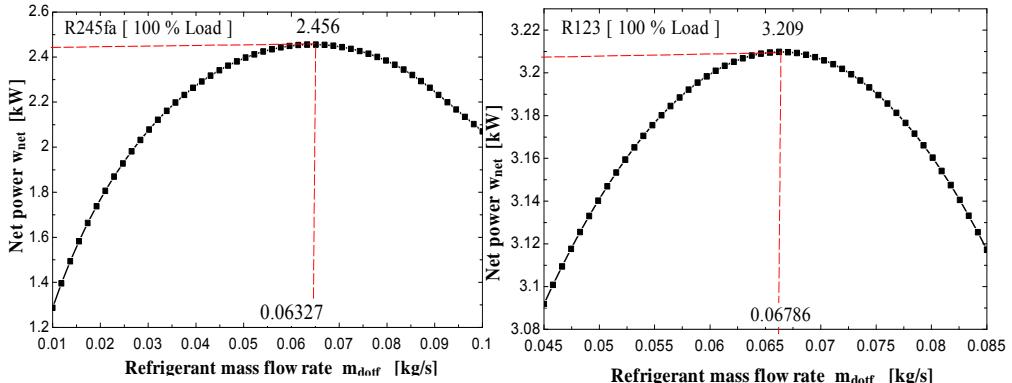


Fig. 8. Effect of Refrigerant mass flow rate on net power for R245fa at 100 %

Fig. 9. Effect of Refrigerant mass flow rate on net power for R123 at 100 %

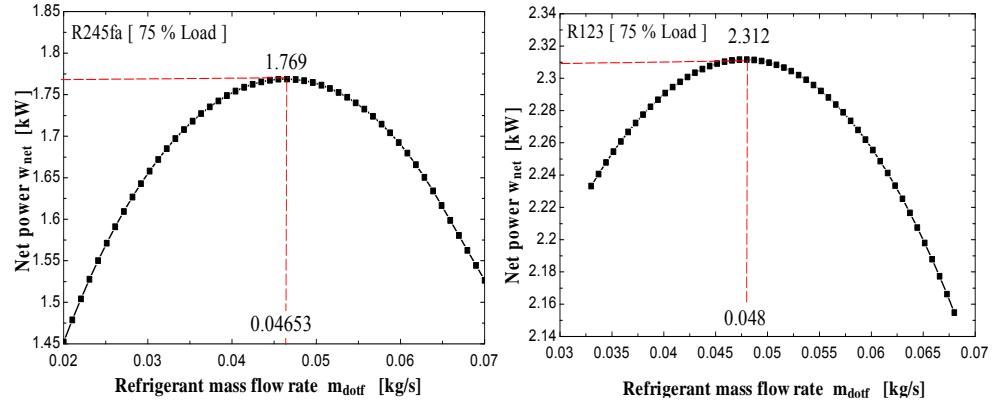


Fig. 10. Effect of Refrigerant mass flow rate on net power for R245fa at 75 % Load

Fig. 11. Effect of Refrigerant mass flow rate on net power for R123 at 75 % Load

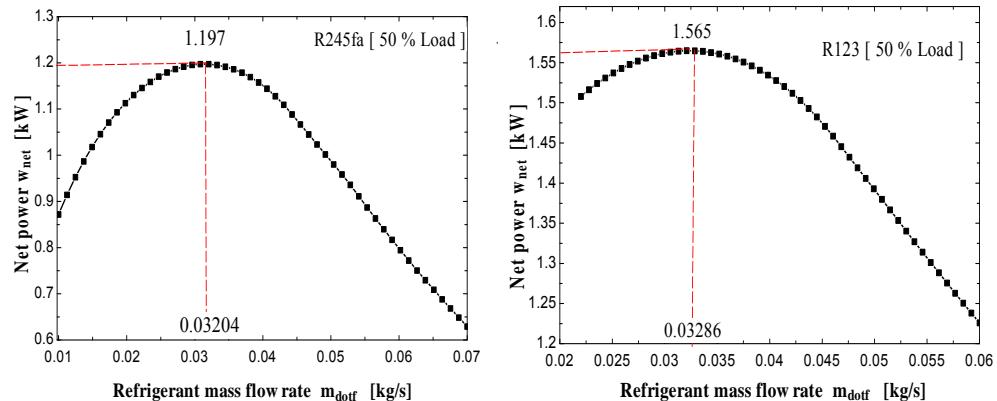


Fig. 12. Effect of Refrigerant mass flow rate on net power for R245fa at 50 %

Fig. 13. Effect of Refrigerant mass flow rate on net power for R123 at 50 %

5. Conclusions

In this study, the heat transfer characteristics of an ORC evaporator and condenser applied on diesel engine were analysed using measured data such as flue gas mass flow rate and flue gas temperature. A mathematical model was created with regard to the preheater, boiler, and the superheater zones of a counter flow evaporator and desuperheater zone, condenser zone of a counter flow condenser. The performance of the ORC system was evaluated under variable heat input. Based on our analysis, the followings can be concluded:

1. There is optimum evaporation pressure to obtain maximum net power depending on working fluid type and this is the first step for design ORC system.
2. For our cases the optimum evaporation pressure for R245fa is 33.37 bar and for R600a is 34.17 bar but there is maximum evaporation pressure for R123 and R141b is 36 bar.
3. There is optimum superheating degree for our working fluids which is between 30 to 40°C.
4. From results it is clear that the maximum net power output between our cases and it is for R141b (3.361 kW) followed by R123 (3.198 kW) followed by R245fa (2.454 kW) and finally R600a (2.112 kW).
5. The design area of evaporator and condenser proportional to net power output in other words we need large area for maximum net power output.
6. The refrigerant mass flow rate is a very important control parameter to maximize the net power output as well as to be sure that the vapour at inlet turbine is superheated to avoid dangerous of the droplets in expander.
7. It is clear from the results that the pinch point temperature difference (ΔT_{pp}) in the evaporator exits (ΔT_{out}) because it is smaller temperature difference between all temperatures difference.
8. Can be seen from our analysis the total area of evaporator is very close from the total area of condenser and this is easier for manufacturer.
9. Most cases larger part of heat adds go to preheater zone and the remaining part subdivided on boiler and superheater zones.
10. The preheater heat input for R123 it is the largest between working fluids but for R245fa it is smallest that's mean we need more heat to convert R123 from liquid state to saturation liquid but need less heat for R245fa. In the same way we need more heat to convert R141b from saturation liquid to saturation vapour but need less heat for R600a. Finally we need more heat to convert R600a from saturation vapour to be superheated but need less heat for R245fa.

Acknowledgments

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