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Original Research Paper

Engineering

Optimization of Organic Rankine Cycle for different working fluids operated by low grade waste heat by using Genetic Algorithm

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ABSTRACT This research work deals with optimization of Organic Rankine Cycle (ORC) system driven by waste heat using R-11, R-141b and R-245fa as working fluids. Thermodynamic performance analysis has been performed. Design of experiment was performed on Genetic Algorithm. Thermal efficiency and network done of system are maximized and exergy destruction rate of evaporator and overall system are minimized by using Genetic Algorithm. The result shows for same power output of turbine R-11 has maximum mass flow rate. Work output for R-245fa is 104.94% greater than R-11, 57.53% greater than R-141b, work output for R-141b is 30.097% greater than R-11 at optimized point. R-245fa has maximum and R-11 has minimum thermal efficiency at optimized point. Ambient temperature has significant effect on exergy destruction rate of overall system, when ambient temperature is too high then exergy destruction rate of overall system is high. In summer weather conditions, exergy destruction rate exceeds 25% more than the winter conditions. Maximum utilization of exhaust heat is a good way to improve work output of system. Overall result shows that R-245fa is best working fluid for inlet temperature of turbine from 390 K to 425K.

KEYWORDS : waste heat, optimization, Genetic Algorithm, Organic Rankine Cycle

INTRODUCTION

Industrial waste heat is the energy which is discharged to the environment after achieving the process; such heat is not put in practical use. Sources of waste heat are industrial hot gases exhaust, exhaust gases after combustion, heat lost from surface of equipment, combustion chamber, furnace etc. Exact quantity of waste heat is unknown but waste heat generated in industry is not utilized in any process and it is directly discharged to atmosphere. Waste heat affects the environment as well as economy. Waste heat is of low quality and have less economic value and lower temperatures, but available in large quantities. New technologies are developing for waste heat recovery. Waste heat can be utilized for drying granulates conversion to electricity, in paper mill, to pre-heat the inlet air to the combustion chamber, energy wheel, heat pump etc. Utilizing waste heat can reduce burden on fossil fuel, thus reducing environment pollution. Recovered waste heat can be converted into electricity by using Organic Rankine Cycle (ORC) which can operate from 353K to 623K.

The main difference between ORC and simple Rankine cycle lies in the working fluid. In simple Rankine cycle water is used as working fluid but in ORC hydrocarbon (butane, pentane hexane etc.) and low boiling point fluids are used. These organic fluids have higher molecular weight and low boiling point which allow recovering waste heat from low boiling point temperature. ORC can be coupled with wide range of heat sources, biomass energy, industrial waste heat, fuel cell and ocean energy. ORC can also couple with thermodynamic cycle like internal combustion engine, thermoelectric generator and Brayton cycle etc. The selection of working fluid is important for ORC; the selection of organic fluid depends on type of heat source. Three categories of Organic fluids type-1, type-2 and type- 3 are used in ORC. The classification of organic fluid is done on slope of saturation vapour curve on T-S diagram. Type -1 fluid are those fluids whose saturation vapour curve is negative. For example R-41, R-116 and water have low critical temperatures therefore they need superheating. Type -2 fluids are dry fluid, slope of saturation vapor curve is positive for example R-600, R-601. Type -3 fluids are isentropic fluids, slope of saturation vapor curve is constant for example R-245fa.

Large quantity waste heat is generated in process industry which is not utilized in any process. Therefore there is need of utilization of waste heat so that dependency on conventional energy reduces. When this waste heat is discharged to environment then causes serious environmental problems. Therefore there is need of utilization of waste heat one of best of option is to convert waste heat into electricity. ORC uses organic (hydrocarbon) fluids with low boiling points and high molecular weights which help to recover heat from low temperature.

The various advantages of ORC include simple design of turbine because of low temperatures and low evaporating pressures, compactness of cycle due to higher fluid density, high heat recovery potential of cycle from low temperature waste streams, no requirement of water treatment system, low condensing pressure and no superheating requirements.

Optimization techniques are used to find design parameters, for this fitness function has to minimize or maximize depends upon nature of problem. In this research work maximization power output of cycle, minimization of exergy destruction rate and maximization of thermal efficiency is done.

Constrained and unconstrained problem can be optimized by genetic algorithm which is based on natural selection. Fitness function can be subjected to constraints in form of equality constraint, inequality constraint or parametric bounds. Genetic algorithm drives biological evaluation. Genetic algorithm modifies population and produce children for next generation. Fitness function continuously optimized in successive generation at last gives optimum solution. Here fitness can be optimized maximum up to 100th generation. To produce children from current population genetic algorithm uses three types of rules and these rules are selection rules, crossover rules and mutation rules. In selection rule: select individual called parents and used to produce children in next generation. In crossover rules: combine two parents to produce children for next generation. In mutation rules: random changes in individual parents so that children are generated.

Most of researchers optimize ORC for very small range of turbine inlet temperature; this research paper is done to optimize for wide range of turbine inlet temperature (390 – 425K). Literature Review no work is found about optimization of working fluid R-11, R-141b, R-245fa by using genetic algorithms. Here optimized thermal efficiency, net power output of system, exergy destruction rate of overall system and evaporator for R-11, R-245fa, and R-141b are compared.

This research paper is organized in 5 sections. Section 2 detailed literature review is done to find research gap. Literature review is done in two fields first is for organic fluids second different methodology to optimize ORC. Section 3 presents thermodynamic analysis of ORC is done and different equations are derived for different component of ORC and cycle description is done. Section 4 presents different plots are discussed, different results is drawn. Section 5 concludes the paper.

CYCLE DESCRIPTION

Basic Components' of ORC are turbine, condenser, evaporator, pump similar to Simple Rankine Cycle and organic fluid. ORC is operated by waste heat shown in fig. 1[9].

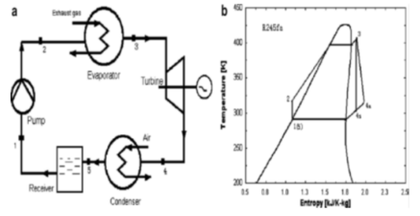


Figure 1(a). schematic of ORC operated by exhaust gases fig 1(b) T-s diagram for ORC on R-245fa

There are five processes to complete a cycle. These proceses described as:

Process 1-2 (pumping)

This is the first process to initialize the cycle and started when pump pushes the working fluid to the evaporator. This process can be considered isentropic *i.e.* S_1

$W_{pideal} = m\,(h_1 - h_2)$ -----(1)

$W_{pactual} = {}^{W}(p\,ideal)/{}^{\eta}p$

$W_{pactual} = m\,(h_2 - h_1)\,{}^{\eta}p$ -----(2)

Where W_p ideal is power needed for ideal pump, $w_{p\ actual}$ is power needed by real pump, m is mass flow rate of organic fluid, ηp is isentropic efficiency of pump, h_1 and h_2 are enthalpies at inlet and outlet of the pump respectively.

Exergy destruction rate for uniform condition is

$$\dot{X} = T_0 \left(\frac{dS}{dT}\right) \text{ -----(3)}$$

Where S refers to entropy, T0 refers to ambient temperature,
For pump

$Xp = 0\ (s1 \approx s2)$ -----(4)

Where X_p refers to exergy destruction rate for pump
Process 2-3 (constant pressure heating)

Constant pressure heating of organic fluid take place and state of organic fluid at outlet or turbine inlet is either superheated vapour or saturated vapour

$\dot{q} = \dot{m}(h_3 - h_2)$

Where $\dot{q}$ refers to specific heat transfer rate from exhaust gases to organic fluid, h_3 and h_2 refers to corresponding enthalpies at evaporator outlet and evaporator inlet. Exergy destruction rate for evaporator is given by

$X_e = mT_0\,[(s_3\neg - s_2) - ((h_3 - h_2)/T_H)]$ -----(5)
S_3 and S_2 refers to specific entropies at inlet and exit of evaporator and TH refers to average temperature of exhaust gases, Xe refers to exergy destruction rate of evaporator

Process(3 to4a;)
Expansion of fluid take place in turbine and mechanical power is generated. Isentropic efficiency of turbine neverbe 100%,

$$\eta_t = \frac{h_{4a} - h_3)}{h_{4s} - h_3)}$$

$\dot{W}t = \dot{m}\,(h_3 - h4_a)\,\eta_m$ ------(6)

Here η_t refers to isentropic efficiency of turbine and ηm refers to mechanical efficiency of turbine, h_{4a}, h_3, h_{4s}, are enthalpies corresponding point here we consider 80% mechanical efficiency of turbine, exergy destruction rate for turbine is

$\dot{X}t = \dot{m}T_0\,(s_{4a} - s_3)$ -----(7)

Process (4-5;)
Cooling of organic fluid take place in air cooled condenser at constant pressure. Arithmetic mean temperature of air is

$T_L = (T_0 + T_{outair}/2)$-----(8)

$\dot{X}_c = \dot{m}T_0\,[(s_5 - s_{4a}) - ((h_3 - h_{4a})/T_L)]$-----(9)

$\dot{X}_c = \dot{m}$ is exergy destruction rate in condenser, s_4 and s_5 refers to specific entropies of organic fluid at inlet and outlet of condenser, T_{outair} is temperature of air at outlet of condenser.

Process (5-1;)
The organic fluid is received in receiver, ignoring static hydraulic pressure and flow resistance process is isenthalpic and isentropic process, exergy destruction rate is zero.

$\dot{X}_c \approx 0$ and $h_5 \approx h_1$ -----(10)

Net power generated in system is equal to turbine power minus pump power,
$W_{net} = W_t - W_{pactual}$

$$W_{net} = \dot{m}\,(h_3 - h_{4a})\,\eta_m - \dot{m}\,\frac{(h_2 - h_1)}{\eta p}$$

$$W_{net} = \dot{m}\left[(h_3 - h_{4a})\,\eta_m - \frac{(h_2 - h_1)}{\eta p}\right]$$

------(11)

Thermal efficiency of cycle is ratio of net power generated in cycle to the heat supplied in evaporator by exhaust gases

$$\eta_{th} = \frac{\dot{m}\left[c_{p,vap.}(T_3 - T_{4a})\,\eta m - c_{p,liq}\,\frac{(T_2 - T_1)}{\eta p}\right]}{\dot{m}_{ex}(h_{in} - h_{out})} \text{ -----(12)}$$

Where η_{th} is thermal efficiency of system, m ex is exhaust mass flow rate (kg/s), T1, T2, T3and T4a are temperature of respective points, hin and hout are enthalpies of exhaust gases at inlet and outlet to evaporator.

Equation 12 can be written in alternate way

$$W_{net} = \dot{m}\left[c_{p,vap.}(T_3 - T_{4a})\,\eta_m - c_{p,liq}\,\frac{(T_2 - T_1)}{\eta p}\right] \text{ -----(13)}$$

Total exergy destruction rate of system obtained by combining equations (4), (5), (7), (9) and (10)

$$X\square_{sys} \approx m\square\ T_0\left[\frac{h_2 - h_3}{T_H} + \frac{h_{4a} - h_1}{T_{air}}\right]$$
$$\approx m\square\ T_0\left[\frac{c_{p,liq}T_2 - c_{p,vap}T_3}{T_H} + \frac{c_{p,vap}T_{4a} - c_{p\ liq}T_1}{T_L}\right] \text{ ----(14)}$$

Where TH refers to average temperature of exhaust gases and TL is arithmetic mean temperature of air, Xsys is exergy destruction of overall system with different assumptions: steady state, uniform condition, considered each component as control volume, equations (12), (14) and (11) are optimized i.e. equations (11) and (12) are maximized and equation (14) is minimized by genetic algorithm tool in MATLAB, equations (12), (14) and (11) are converted into fitness functions, considering isentropic efficiency of pump 60%, equality constraint turbine outlet temperature(T_{4a}) 293 K, exhaust mass flow rate 8 kg/s, outlet temperature of exhaust gases 373 K.

There are two bounds one is lower bound (lb) one is upper bound(ub) these are different for different parameters. These are:

Table 1: Parametric bounds

Variable or parameter	Lower bound (lb)	Upperbound (ub)
Mass flow rate of organic fluids (kg/s)	4	6
Turbine inlet temperature (K)	390	425
Pump outlet temperature (K)	295	300
Pump inlet temperature (K)	290	292
Exhaust inlet temperature (K)	643	673

DISCUSSION

To plot Figure (2) average temperture of refrigerent (K) at inlet of turbine is taken on x axis and mass flow rate of refrigerent taken on y axis, for this variation of temperature in the range 353 to 428 K figure (2) shows that as the average temperture of refrigerent at inlet of the turbine which is producing constant power 140 kW, mass flow rate of refrigerent needed to produce constant power continuesly decreses , to produce same power less amount of mass flow rate needed, but one of the intersting behaviour of R-245 is that first decrese in mass flow rate upto 413K after more mass flow rate needed upto critical temperture , after critical temperture decomposition and deteriotation of organic fluid going to take palce,R-11 needed more mass flow rate than R-245fa and R-141b, upto 403 K for producing same power less mass flow rate of R-245 needed then R-141b but after this R-245fa needed more mass flow rate of refrigerent than R-141b.

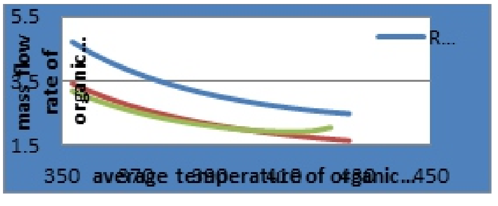


Figure 2: mass flow of refrigerent – average temperture of orgnic fluids at turbine inlet

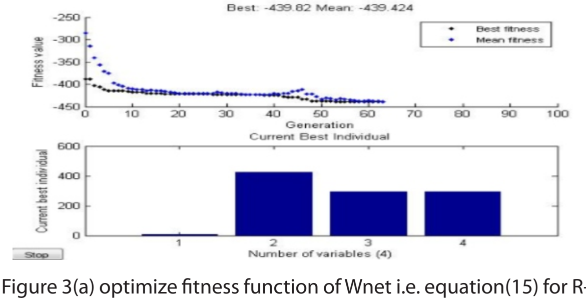


Figure 3(a) optimize fitness function of Wnet i.e. equation(15) for R-11

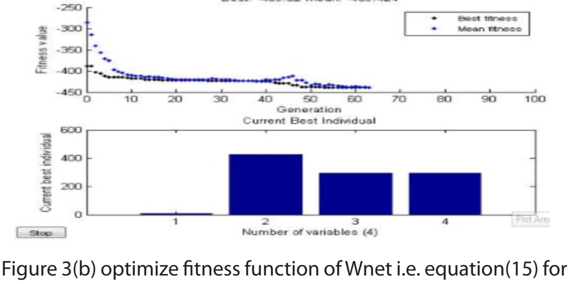


Figure 3(b) optimize fitness function of Wnet i.e. equation(15) for R-141b

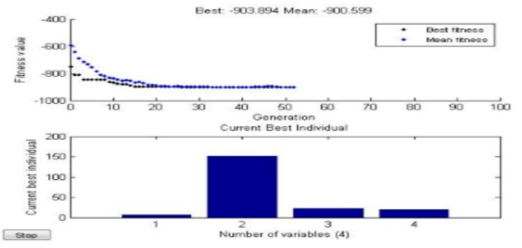
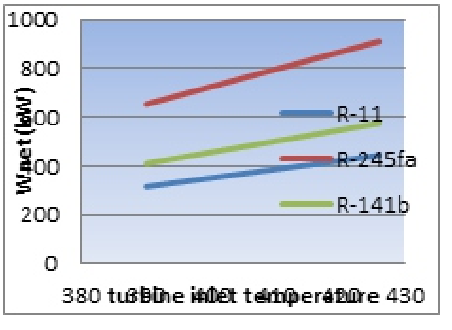
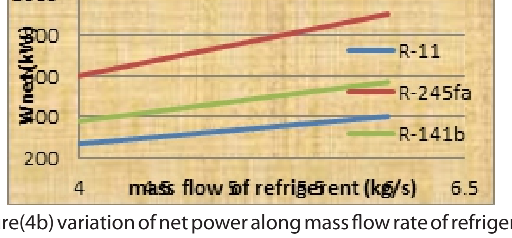


Figure 3© optimize fitness function of Wnet i.e. equation(15) for R-245fa

Figure (3a),(3b),3© gets after optimization of equation(15) for R-11, R-141b and R-245fa by palcing values of parameteric bound in genetic algorithm tool of MATLAB, 4 parametric bounds or varibles are taken, variable 1 refres to mass flow rate of refrigerent and variable 2 refers to turbine inlet temperture , variable 3 refers to pump outlet temperture and varible 4 refers to pump inlet temperture, these figures there is two plots in one figure above plot is for values of fitness function at each iteration and bottom plot is for value of input varible in form of bar chart at optimized point , figure 3(a), 3(b) and 3(c) shows that variable 2 has maximum value means turbine inlet temperture has maximum Value, variable 1 has minmum value, for figurure (3a) i.e. for R-11 upto 10th genertion gape between values of fitness function of equation (15) for R-11 of is too high, after 10th genertion gape between values of fitness function slowly decreses finally after 63rd genertion get optimized point [5.996 423.869 295.012 291.999] and fitness function value 439.82 kW, figure (3b) shows that gape between values of fitness function up to15th genertion of equation(15) for R-141b is too high after 15th generation gape between values of fitness function slowly decreases and fitness function optimized at 55th generation and optimized point is [6 424.999 295.001 291.998] and value of fitness function 571.972 kW, figure(3c) shows that gape between values of fitness function of equation (15) for R-245fa upto 17th genertion is too high after 17th genertion gape between valuse of fitness function continously decreases and finally get optimized point [6 424.907 295 291.998] at 52th generation and fitness function value at this is 903.894 kW, at optimized point Wnet in kW for R-245fa is 104.94% greater than R-11 and 57.53% greater than R-141b, Wnet for R-141b is 30.097% greater than R-11. figure (4a) is variation of net power of system against temperature(K) at inlet of turbine powe Wnet is taken on Y axis and temperature (K) taken on X axis , temperture of variaiation is 390 K to 425 K corresponding power of cycle is presented on Y axis,results shows that power of cycle (kW) variying linerly with turbine inlet temperature.



Figure(4a) variation of net power along turbine inlet temperture



Figure(4b) variation of net power along mass flow rate of refrigerent

With increase in turbine inlet temperature from 390 K to 426K increase in Wnet for R11 is 40.28%, for R-245fa is 39.39% and for R-141b is 40.24%, at turbine inlet temperature 426K net power generated for R245fa is 105.61% greater than R11 and 58.31% greater than R141b, For highest turbine inlet temprature 426K, power for R141b is 29.87% greater than R11. Figure (4b) shows variation power of cycle (kW) against mass flow rate of refrigerent, mass flow rate of refrigerent taken on X axis net power of cycle (kW) taken on Y axis, varitation of mass flow rate of organic fluids for plot is 4 to 6 kg/s, results for figure (4b) shows that Wnet varies linearly with mass flow of refrigerent as mass flow rate increases Wnet of system also increases, just slight of increse in mass flow rate from 4 to 6 kg/s increase in Wnet for R11 is 49.99%, for R245fa is 50%, forR141b is 50% hence trends in increses in W_{net} for R11,R245fa and R141b is same but at highest mass flow rate of refrigerent 6(kg/s) R245fa produce 124.24% more power than R11 and 58.174% more power than R141b, R141b produce 41.74% more power than R11.

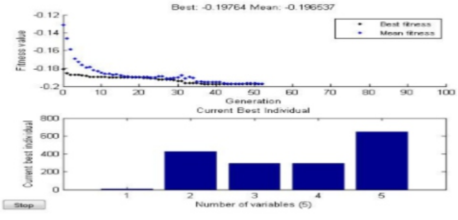


Figure (5a) for maximization of efficiency for R-11

For maximizing effeciency of ORC: fitness function for effeciency i.e. equation(16) optimized in genetic algoritm tool in MATLAB, 5 parametric bounds are taken, variable 5 has maximum value i.e. exhaust gas temperature, second highest value is for varible (2) i.e. turbine inlet temperature, figure (5a), (5b), (5c) shows that varible (5) has maximum value and varible (1) has minimum value, for R-11 upto 15th genertion gape between values of fitness function is too high after that gape decreses continously and fitnees function optimized at point [5.997 423.54 295.024 291.985 646.535] 52th generation, value of fitness fuction at this point is 19.74%, figure (5b) is for optimization of effeciency of R-141b, upto 18th generation or iteration gape between values of fitness function is too high, after it gape between values fitness function slowly decreases and gets optimized value 25.52% at 52nd generation and optimized point is [5.97 424.937 295.122 291.994 649.467], figure (5c) is for optimization of effeciency of R-245fa: upto 15th gape between values of fitness function is too high after that gape decreases and gets optimized value at a point [5.998 421.787 295 291.994 649.576] at 51 generation or iteration and value of fitness function is 39.6304%, the exhaust inlet temperture increses thermal effeciency of oragnic rankine cycle increases, higher the tempearture of organic fluids higher the the thermal effeciency , condenser temperture should be lower, work required for pump should be lower then thermal effeciency going to improve.

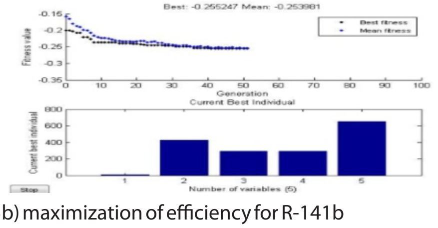
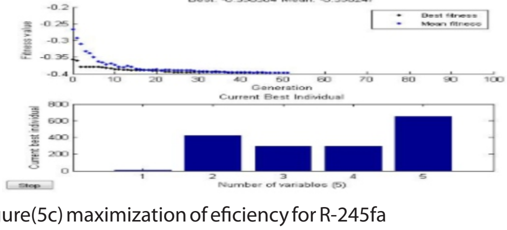
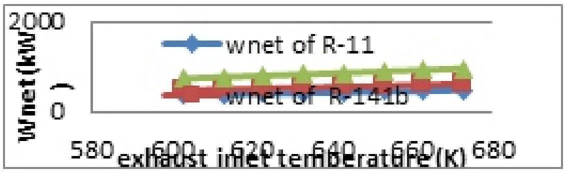


Figure (5b) maximization of efficiency for R-141b

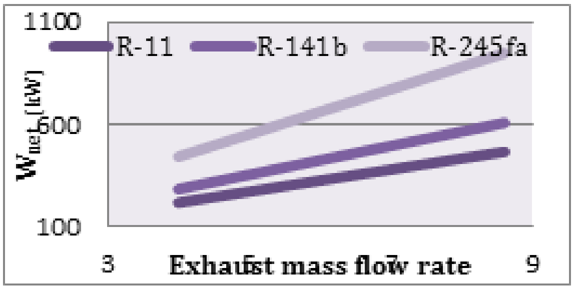


Figure(5c) maximization of eficiency for R-245fa

Figure (6a) shows the variation of net power of cycle against exhaust inlet temperture, for this exhust inlet temperture taken on x-axis and net power of cycle taken on Y axisWnet or net power of cycle(kW) increases linearly with exhaust inlet temperature, at highest exhaust inlet temperture 673K: R245fa produce 101.6% more power than R11, 56.01% more power than R141b, at highest exhaust inlet temperture 673K R141b produce 29.22more power than R11.



Figure(6a) variationbetween exhaust inlet temperature and $W_{net,}$



Figure(6b) shows variation between exhaust mass flow rate and W_{net}

Figure (6b) shows variation between net power generated in cycle and exhaust mass flow rate, for this purpose exhaust mass flow rate taken on X- axis and net power generated taken on Y –axis, plot shows that net power of cycle increases linearly with exhaust mass flow rate.

W_{net} increases linearly with exhaust mass flow rate, for highest exhaust mass flow rate of 8.5 kg/s power produce by R245fa is 103.85% more than R11, 56.075% more power than R141b, for highest exhaust mass flow rate of 8.5 kg/s power produce by R141b is 30.61% more than R11.

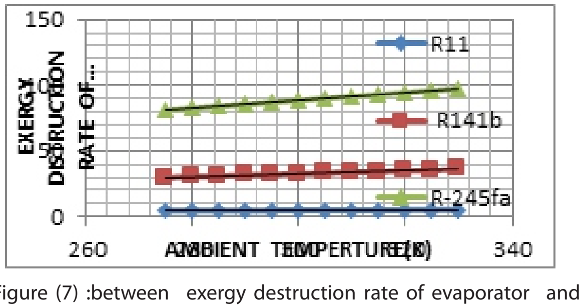


Figure (7) :between exergy destruction rate of evaporator and ambient temperature

Figure (7) is plot between exergy destruction rate of evaporator and ambient temperature, for this purpose ambient temperature (K) is taken on X- axis and exergy destruction rate of evaporator(kW) is taken on Y-axis, figure (7) shows that as the ambient temperature increases exergy destruction rate increases, this indicate that under high temperature condition exergy destruction arte of evaporator is also high, summer condition system performance is poor than in winter condition therefore choosing a proper nominal or ambient condition is also important for better performance of system, exergy destructions rate in summer is more than 25% than in winter condition.

CONCLUSIONS

More mass flow rate of R-11 than R-141b and R-245fa needed for producing same mechanical power through turbine, 403 K turbine inlet temperature R-245fa has less mass flow rate than R-141b after 403 K R-245fa needs more mass flow rate. As the temperature at

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turbine inlet increases required mass flow rate decreases. Higher the turbine inlet temperature and lower condenser pressure results in higher mechanical power of system as well as thermal efficiency. At optimized point net power generated for R-245fa is higher than R-141b and R-11. At optimized point net power generated for R-141b is higher than R-11. At optimized point W_{net} in kW for R-245fa is 104.94% greater than R-11 and 57.53% greater than R-141b, W_{net} for R-141b is 30.097% greater than R-11. Net power generted in system is increses linerly with turbine inlet temperture and mass flow rate of refrigerent. At turbine inlet temperture 426K net power generated for R245fa is 105.61% greater than R11 and 58.31% greater than R141b, For highest turbine inlet temp 426K, power for R141b is 29.87% greater than R11. At highest mass flow rate of refrigerent 6(kg/s) R-245fa produce 124.24% more power than R11 and 58.174% more power than R-141b, R-141b produce 41.74% more power than R11. Net power generated varies linearly with exhaust inlet temperature. Optimized point for thermal efficiency and $W_{ork\ output}$ is different. Effective utilization of heat source can improve system performance. Ambient temperature affects the exergy destruction rate of overall system, more than 25% exergy destruction rate in summer than winter. This approach is very helpful for optimizing the parameters easily so that there is no need to perform number of experiment for optimizing the cycle, so that time going to save. Similar types of optimizing technique can perform for larger number of refrigerants and variables

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