Parametric Analysis of Geothermal Organic Rankine Cycle Working on Zeotropic Mixture of R600a/DME

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Abstract : Organic Rankine cycle (ORC) is preferred to convert low temperature geothermal energy to electricity. Selection working fluids and optimization system parameters are two main approaches to improve the performance of ORC system. Zeotropic mixtures show promise as ORC working fluids owing to better match between the working fluid and the heat source/sink temperatures. This study optimizes the evaporator inlet temperature of mixture R600a/ Di-Methyl-Ether (R600a/DME) for various mass fractions to maximize the net work output and compare the thermal efficiency and exergetic efficiency of system for geothermal water temperature of 393K. Irreversibility in each component of system and mass flow rate of mixture are also found in this analysis by varying mass fraction of R600a. It was found that R600a/DME (0.8/0.2) gives the maximum net work output corresponding to 343K inlet temperature to evaporator. Among all selected proportions, R600a/DME (0.6/0.4) has both maximum thermal efficiency and maximum exergetic efficiency corresponding to 373k inlet temperature to evaporator. Irreversibility present in evaporator is minimum for mass fraction 0.6 of R600a. Mass flow rate of mixture increases with mass fraction of R600a up to 0.6 and then become approximately constant.

Keywords: ORC, DME, Mass flow rate, Thermal Efficiency, Exergetic Effeciency.

1. INTRODUCTION

Geothermal energy is thermal energy generated and stored in the Earth. Thermal energy is the energy that determines the temperature of matter. The geothermal energy of the Earth's crust originates from the original formation of the planet and from radioactive decay of materials (in currently uncertain but possibly roughly equal proportions). The geothermal gradient, which is the difference in temperature between the core of the planet and its surface, drives a continuous conduction of thermal energy in the form of heat from the core to the surface.With the cost of energy constantly increasing and the world turning towards more environment friendly and energy efficiency, there has been a need for numerous alternative and renewable sources of energy. One of such sources is an Organic Rankine cycle (ORC). An ORC converts low temperature waste heat into useful work and emits absolutely no carbon dioxide or pollutants. ORC falls under the category of micro turbines. Although these cycles are only approx 10% to 15% efficient they are utilizing heat energy of exhaust gases, which is otherwise wasted and expelled into the atmosphere. An ORC is similar to Rankine cycle but operates on an organic based fluid, an organic fluid is defined as a fluid that contains carbon. The main organic fluids are refrigerants or hydrocarbon chain fluids. An ORC is a basic power producing cycle that utilizes low temperature waste heat to produce electricity. An organic Rankine cycle is similar to a Rankine cycle in the sense that it turns a fluid to a vapour and forces the vapour through a turbine which spins a shaft connected to a generator which in turn produces electricity. The vapour is then forced through a condenser that changes the vapour back to a liquid which then enters a pump and starts the cycle again. An ORC is different from a Rankine cycle because it uses a fluid that is organic based, meaning it contains carbon, and has a lower boiling point. Since the fluid has a lower boiling point the cycle can utilize lower temperature heat to cause the fluid to change phase from a liquid to a vapour.
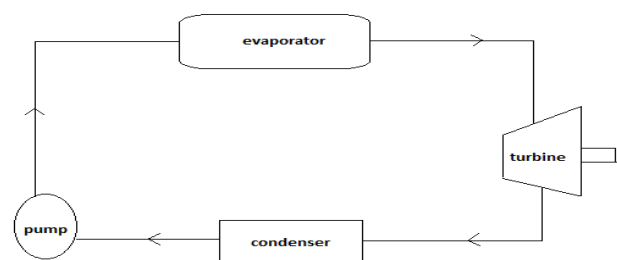

Fig.1 Schematic diagram of basic Rankine cycle

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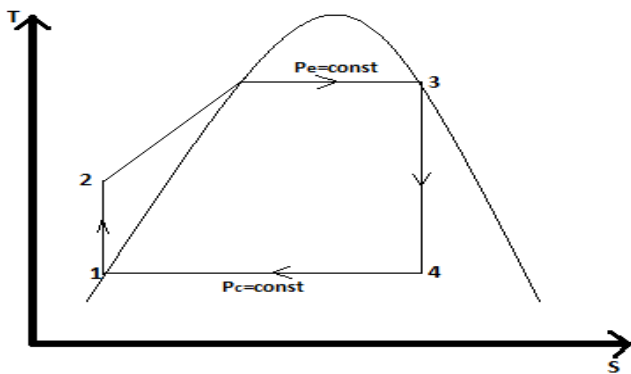

Fig.2 T-S Diagram of Basic Rankine Cycle

An ORC is similar to Rankine cycle but operates on an organic based fluid, an organic fluid is defined as a fluid that contains carbon. The main organic fluids are refrigerants or hydrocarbon chain fluids. An ORC is a basic power producing cycle that utilizes low temperature waste heat to produce electricity. An organic Rankine cycle is similar to a Rankine cycle in the sense that it turns a fluid to a vapour and forces the vapour through a turbine which spins a shaft connected to a generator which in turn produces electricity. The vapour is then forced through a condenser that changes the vapour back to a liquid which then enters a pump and starts the cycle again. An ORC is different from a Rankine cycle because it uses a fluid that is organic based, meaning it contains carbon, and has a lower boiling point. Since the fluid has a lower boiling point the cycle can utilize lower temperature heat to cause the fluid to change phase from a liquid to a vapour. Several geothermal provinces in India categorized by high heat flow (78.468 MW/m^2) and thermal gradients (47100°C/km) discharge about 450 thermal springs. These provinces are capable of generating 10,600 MW of power. Though geothermal power production in Asian countries like Indonesia, Philippines has gone up by 1800 MW in 1998, India with its 10,600 MW geothermal power potential is yet appear on the geothermal power map of the world.

The thermal efficiency of geothermal electric plants is low, around 10 to 23%, because geothermal fluids do not reach the high temperatures of steam from boilers. The laws of thermodynamics limit the efficiency of heat engines in extracting useful energy. Exhaust heat is wasted, unless it can be used directly and locally, for example in greenhouses, timber mills, and district heating. System efficiency does not materially affect operational costs as it would for plants that use fuel, but it does affect return on the capital used to build the plant. A very large amount of geothermal energy is stored in the earth, but 70% of the geothermal source is low-enthalpy geothermal water at temperatures lower than 150℃. The International Geothermal Association (IGA) has reported that 10,715 megawatts (MW) of geothermal power in 24 countries is online, which was expected to generate 67,246 GWh of electricity in 2010. This represents a 20% increase in online capacity since 2005. IGA projects growth to 18,500 MW by 2015, due to the projects presently under consideration, often in areas previously assumed to have little exploitable resource [3, 50-54].

2. LITERATURE REVIEW

Angelino and Paliano [4] examined the performance of a geothermal ORC with mixtures of n-butane and n-hexane. Their results showed that the ORC with a mixture of n-butane and n-hexane produced 6.8% more electricity than with just n-pentane. **Heberle et al. [7]** investigated the exergy efficiency of subcritical ORCs with zeotropic mixtures (isobutane/isopentane and R227ea/R245fa) as the working fluids for conversion of low-enthalpy geothermal sources. Their results showed that the exergy efficiencies increased by 4.3–15% using mixtures compared to the most efficient pure fluid for geothermal source temperatures below 120℃. They pointed out that the temperature glide during condensation should be fit to the cooling water temperature difference. **Baik et al. [8]** investigated the power enhancement potential of a transcritical ORC with R125 based HFC mixture working fluids using a low-temperature geothermal heat source of about 100℃. Their results showed that the optimized transcritical ORC with an R125/R245fa mixture working fluid yielded 11% more power than the optimized subcritical ORC with just R134a. **Liu et al. [9]** investigated the method to determine the mixture condensation pressure and the effect of the condensation temperature glide on the geothermal ORCs performance with zeotropic mixtures as working fluids. Their results showed two maxima in the cycle thermal efficiency, exergy efficiency and net power output when the condensation temperature glide matches the cooling water temperature rise. Use of zeotropic mixtures can also increase the thermodynamic performance of ORCs driven by solar energy or high temperature heat sources. **Papadopoulos et al. [10]** proposed a holistic approach for fluid selection. By the use of computer aided molecular design in conjunction with process optimization physical, chemical, environmental, safety and economic properties of pure ORC fluids were evaluated. Schuster et al. considered different working fluids in supercritical cycles. **Demuth [11]** evaluated in a case study for a geothermal power plant two-component mixtures of natural hydrocarbons for certain compositions at 137℃ and 182℃ geothermal water temperatures. For subcritical cycles efficiency increases up to 14% compared to the most efficient pure fluid propane. **Heberle and Brüggemann [12]** investigated both series and parallel CHP system

configurations for thermal utility temperature up to 90℃. Their analysis showed higher second law efficiency for the CHP series concept. They reported an increase of 20% in second law efficiency of the CHP system when compared to the stand-alone power generation ORC. The system's second law efficiency reached about 54% for the 'series' concept and about 50% for the 'parallel' concept. **Kim et al. [13]** analyzed the performances with seven different fluids of a regenerative ORC with heat supplied in series driven by a low-temperature heat source. They found that higher turbine inlet pressure leads to lower second law efficiency of ORC, but anyway higher than the CHP system. Also the optimal working fluids vary with the heat source temperature. **Gozdur and Nowak [14]** studied Rankine cycles with heat source temperature of 80℃ to 115℃ using natural and synthetic working fluids, as well as mixtures. They found that highest values of power obtained have been for the natural working fluid-propylene and single component synthetic fluid R227ea; highest values of efficiency obtained have been for the natural working fluid-propylene and single-component synthetic fluid R245fa. **Habka and Ajib [15]** investigated the effect of the heating demand parameters of the cogenerative section on the overall power plant behavior for two connections of CHP systems (parallel plant and CHP integration according to NueStadt Glewe plant in Germany) operating with R134a and fuelled by a geothermal resource at 100℃. They investigated the exergy efficiency, net output power of ORC, irreversibility related to the exhausted geothermal water and the total heat exchangers surface areas. They concluded that the performances of such CHP configurations are compromised when working at high cogeneration heating parameters (i.e. temperature and heat demand of the utilities). They also found that the parallel connection is more economical and that the series connection is energetically more efficient and that, on the contrary, the integration according to NueStadt Glewe power plant does not provide any significant optimization. They also noticed that the maximum optimized mechanical power in all of the investigated CHP configurations is not higher than 50% of the maximum power produced by the corresponding stand-alone ORC. **Guo et al. [16]** investigated a novel cogeneration system consisting of low temperature geothermal-powered ORC, an intermediate heat exchanger and a heat pump subsystem at same time identifying a suitable working fluid. The results indicated that the optimized fluids based on each screening criteria are not the same and there exist optimum evaporation temperatures maximizing the P_{net} value and minimizing the A/$P_{net.}$ **Hung et al. [17]** studied an ORC using different fluids among wet, dry and isentropic fluids. Dry and isentropic fluids showed better thermal efficiencies and moreover, they did not condense during expansion in the turbine thus less damage in the machine was obtained. **Tchanche et al. [18]** analyzed thermodynamic characteristics and performances of 20 fluids in a low-temperature solar organic Rankine cycle and R134a was recommended. **Heberle et al. [19]** studied the second law efficiencies of zeotropic mixtures as the working fluids for a geothermal ORC. The results showed that the efficiency was increased up to 15% compared to that of pure fluid for heat source

temperature below 120°C. **Deethayat et al. [20]** studied a basic ORC using R245fa/R152a as the working fluids and the irreversibility at the evaporator and the condenser were found to be less than those of the unit using single R245fa. Anyhow, there was a limit of R152a composition due to its high flammability when the value was over 30%. In this study, performance analysis of a 50KW ORC with internal heat exchanger was studied when the working fluid was a mixture of R245fa/R152a. A hot water stream at 115°C was taken as a heat source at the evaporator and a cool water stream fixed at 27°C was conducted as a heat sink at the condenser. The effects of evaporating temperature, mass fraction of R245fa and effectiveness of internal heat exchanger on the ORC performances following the first law and the second law of thermodynamics were considered. **Fiaschi et al. [21]** investigated possibility of using an absorption heat transformer to enhance low-enthalpy geothermal water temperature for producing electricity throughout ORC power plant. **Gozdur and Nowak [22]** found that the cycle efficiency is not a sufficient criterion for assessment of the ORC. Regarding the ORC–CHP systems energized by geothermal water, few activities and researches have been conducted for different assumptions and evaluations. **Li et al. [23]** analyzed the series and parallel circuit geothermal systems (100–150℃) in oil field using ORC under consideration of various working fluids. The results showed that R601a has the highest cycle performance within the scope of that study and the series circuit with a pre heater has higher efficiencies than that without. **Tempesti et al. [24]** presented a thermoeconomic analysis of a micro-Combined Heat and Power (CHP) plant operating through an ORC using the geothermal (80–100℃) and solar energies. The results showed that R245fa allows the lowest price of electricity production and the lowest overall cost of the CHP plant. **Khennich et al. [25]** modeled two CHP systems with ORC and R134 a as working fluid. The both systems generated less mechanical power than the heat delivered to the heating load and a higher fraction of the heat source was used as the heating load increases. **Mago et al. [26]** analysed the exergy destruction in Organic Rankine Cycle. Visual representations using an exergy wheel clearly show the exergy accounting for each thermodynamic process. The results show that the evaporator has by far the highest exergy destruction rate, followed by the turbine. Therefore, it seems that cycle modifications, of which the aim is to reduce exergy destruction in the evaporator, have a major potential to increase the power output of the ORC. **Roy et al. [27]** studied the output power, the second and first law efficiency and irreversibilities of an ORC using R12, R123 and R134a as working fluids. The ORC was driven by flue gas waste heat at 140℃. Their results show that the point of maximum thermal efficiency and maximum power output do not coincide. Furthermore the second law efficiency is strongly affected by the pinch point temperature difference in the evaporator. **Heberle et al. [28]** investigated the second law efficiency of an ORC with zeotropic mixtures of isobutene/isopentane and R227ea/R245fa as working fluids. The results show that for temperatures below 120℃ the second law efficiencies increased in the range of 4.3–15%.

The optimal second law efficiency was achieved when the temperature glide of condensation and cooling water matched. **Ho et al. [29]** compared the Organic Flash Cycle (OFC) to an optimized basic ORC cycle, a zeotropic Rankine cycle with a binary ammonia–water mixture and a transcritical CO_2 cycle. A distinction is made between utilization efficiency and second law internal efficiency. The former definition assumes that the exergy which is left in the waste heat stream is discarded or unused, while the latter discards exergy destruction due to heat transfer in the evaporator. The definition of second law efficiency is therefore not unique; it is based on a carefully selected set of chosen input and output streams. **Liu et al. [30]** analyzed the influence of the temperature glide during the zeotropic condensing process on the thermal efficiency, exergy efficiency and output work of ORC system. Based on the experimental result of the exhaust gas under varying operating conditions. **Yang et al. [31]** studied the system performance of eight zeotropic mixtures as working fluids in a waste heat recovery system of vehicle engine. **Wang and Zhao [32]** compared three different compositions (0.9/0.1, 0.65/0.35 and 0.45/0.55) of R245fa/R152a to pure R245fa at a low temperature solar ORC. In order to investigate the second law efficiency of subcritical cycles. **Garg et al. [33,34]** respectively used isopentane/R-245fa, CO_2/isopentane and CO_2/propane as working fluids, and evaluated the system performance. A technique of identifying the required source temperature for a given output of the plant and the maximum operating temperature of the working fluid is developed by the authors. For the heat source temperature of 150℃ and 250℃, when using mixtures as the working fluids of ORC systems. **Chys et al. [35]** found a potential increment of 16% and 6% in system efficiency respectively. The power generation at optimal condition can be increased by 20% for the low temperature heat source comparing with the pure working fluids. **Venkatarathnam et al. [36]** considered that there were certain limits for the temperature glide of the heat transfer fluid in the evaporator and condenser to avoid pinch point, which could be used to evaluate the suitability of zeotropic mixtures for glide matching. **Chen et al. [37]** proposed a supercritical Rankine cycle using zeotropic mixtures for the low grade heat. The result showed that thermal efficiencies of the cycles using mixtures (0.7R134a/0.3R32) were 10–30% higher than the cycle with pure R134a. **Saleh et al. [38]** examined 31 pure fluids with ORC cycle operating temperature of less than 100℃. The results show that the thermal efficiency ranges between 3.6% and 13% depending on choice of working fluid and rises with increasing critical pressure of the fluid. **Tchanche et al. [39]** evaluated 20 working fluids for a solar ORC micro-power system. For the chosen boundary conditions R134a, R152a and R600 are the most suitable fluids. **Papadopoulos et al. [40]** proposed a holistic approach for fluid selection. By the use of computer aided molecular design in conjunction with process optimization physical, chemical, environmental, safety and economic properties of pure ORC fluids were evaluated. **Bliem [41]** investigated the use of R114/R22 for geothermal power generation. The mixture shows between 3% and 8% higher efficiency

compared to R114. It should be noted that these fluids are forbidden by law nowadays. **Gawlik and Hassani [42]** demonstrated that levelized equipment costs can be reduced by using mixtures instead of pure fluids in geothermal binary plants. **Borsukiewicz [43]** analyzed different pure fluids and a propane/ethane mixture for low-temperature ORC. They observed a higher power output at similar thermal efficiency for the equimolar propane/ethane mixture compared to pure propane. **Lakew et al. [44]** stated that when a heat source with temperature ranging from 80 to 160℃, using R227ea resulted in a maximum power output; when heat source temperature ranged from 160 to 200℃, using R245fa could obtain a maximum power.

Through the literature survey it was recognized that the parametric study of the mixture of R600a and Di-Methyl-Ether used in the ORC has not been done. Efficient operation of an Organic Rankine Cycle is primarily a function of two parameters: the working conditions of the cycle and the thermodynamic properties of the working fluid used in the cycle. Three main types of working fluids exist that can be used in ORC. These types are classified by their slope on the vapour side of the saturation curve. The plot in fig.2.1 is a simple saturation dome that has been cut off after the inflection point in order to display the different type of fluid slopes. Fig.2.1 below shows a comparison of the types of fluids classified by their slopes on a T-s diagram. Note that this diagram does not directly reflect any specific fluids it is simply a graphic display of the possible types of fluids and their slopes [45,46].
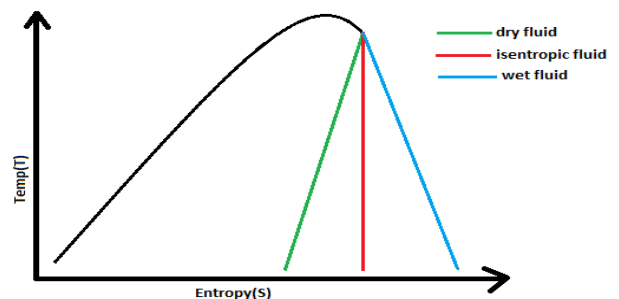

Fig.3 T-S Diagram for Different Fluid Types

A fluid that has a negative slope, such as the blue line in fig.3, is called a wet fluid. The most common wet fluid is water but other wet fluids include: ethanol, methane, hydrogen, DME and most fuels. A fluid that has a positive slope, such as the green line in fig.2.1, is called a dry fluid. Common dry fluids include: toluene, n-pentane, isopentane, R-245fa, Isobutane and many refrigerants. A fluid that has an infinite slope, such the red line in fig.2.1, is called an isentropic fluid. Common isentropic fluids include: benzene, R-11, R-123, and hexane. The different types of fluids can vary the thermal efficiency of the ORC and each type can offer a different advantage when it is used within the cycle.

3. SYSTEM DISCRIPTION
The working principles for the ideal organic Rankine cycle are similar to the ideal Rankine cycle. The condensate working fluid is pumped from the condenser where the

pressure is low to the evaporator where the pressure is high. The process takes place at constant entropy. The high pressure liquid enters the evaporator and absorbs the thermal energy from heat source at constant pressure. In this process the refrigerant changes the phase from saturated liquid to saturated or superheated vapor. The external heat source can be waste heat from industry, geothermal heat, solar heat, biomass etc. The high pressure saturated or superheated vapor leaves the evaporator and expands through an expander at constant entropy to produce mechanical work. Under the expansion process, the pressure decreases to condenser pressure. After expansion process the working fluid leaves the expander and enters the condenser as unsaturated, saturated or superheated vapor depending on working conditions and the type of used working fluid. In the condenser, the working fluid condensates and changes phase to saturated or undercooled liquid with the help of a heat sink, and then the cycle is repeated.
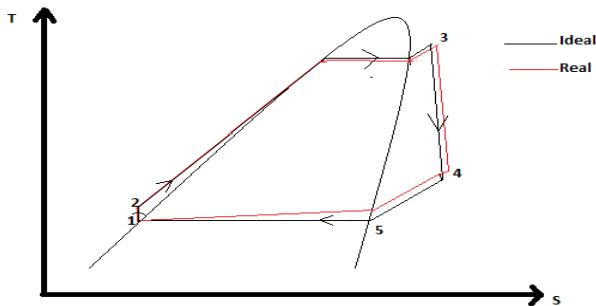
Fig.4 Ideal and Real Organic Rankine Cycle

In the real cycle the compression and expansion processes are not isentropic and there are always some losses in the pump and the expander. The heat addition and heat rejection processes are not isobaric and there are always pressure losses in the piping system. The irreversibility affects very much the performance of the thermodynamic system.

In a real cycle, there are two main sources for entropy generation and these sources are external and internal. The internal entropy generation occurs due to

- Pressure drop because of friction in the system associated pipes
- Un-isentropic compression and expansion in the compressor or expander
- Internal transfer of energy over a finite temperature difference in the components.

And the external entropy generation occurs due to
- The mechanical losses during work transfer
- Heat transfer over the finite temperature difference

Organic Rankine cycle has the same working principles and main components (evaporator, condenser, expander and pump) as the steam Rankine cycle. The main difference between the two cycles is the working fluid utilized. Fig.5

shows the T-S diagram for a basic organic Rankine cycle and fig.6 shows the cycle layout[55-56].
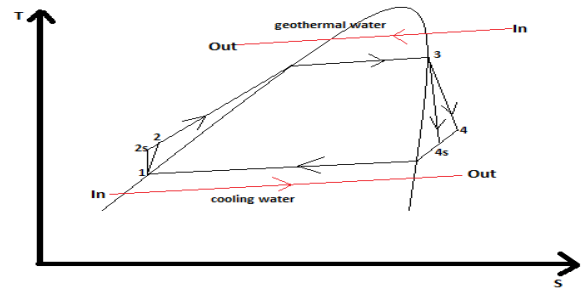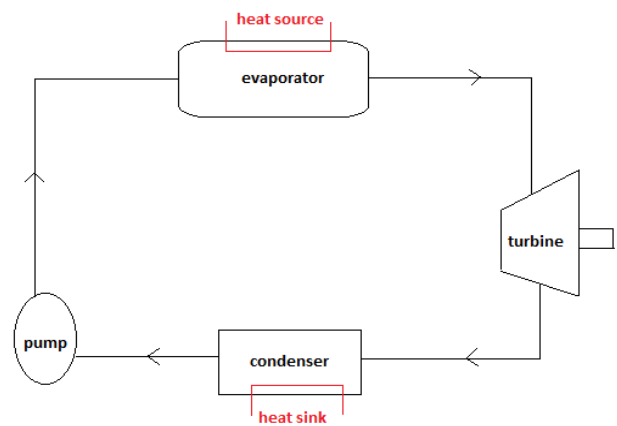

Fig.5 T-S diagram of actual organic Rankine Cycle

Fig.6 Basic layout of organic Rankine cycle

Table 1 Input Parameters

Ambient temperature (T_o)	298K
Inlet temperature of geothermal heat source (T_{in})[49]	393K
Mass flow rate of geothermal water (m_2)[50]	5 kg/s
Inlet temperature of cold source (T_{ci})	293K
Outlet temperature of cold source (T_{co})	303K
Evaporating temperature (T_e)[50]	323K-373K
Temperature difference at pinch point (ΔT_{pp})	10K
Condensing temperature (T_c)	308K
Isentropic efficiency of pump (η_p)	75%
Isentropic efficiency of turbine (η_t)	85%
Const. pressure specific heat of geothermal water (C_{pw})[51]	4.31kJ/kgK

Table2. Thermal physical properties of selected organic working fluids [45]

Name	NBP(℃)	T_c(℃)	P_c(kPa)	ODP	GWP (100 yr)	Safety Group	Expansion State
R600a	-11·7	134.70	36·3	0	20	A3	Dry
DME	-24·8	127.23	53·4	0	20	A3	Wet

NBP is normal boiling temperature of fluid
T_c is critical temperature of fluid
P_c is critical pressure of fluid
ODP is ozone depletion potential of fluid
GWP is global warming potential of fluid
Expansion state is the state after the expansion in turbine
The thermodynamics analysis of geothermal organic Rankine cycle using zeotropic mixture for the condition stated above, have been carried out by using software simulations, many thermal physical properties of zeotropic mixture at various state points have been obtained.

4. RESULTS AND DISCUSSION
The organic Rankine cycle (ORC) using geothermal water as heat source is analysed for temperature ranges of zeotropic mixture of organic fluids at the inlet to the evaporator by developing simulations on softwares. The performance parameters selected for the analysis are thermal efficiency, exergetic efficiency, net work output, irreversibility present in each component of system, mass flow rate of organic fluid etc. The performance of geothermal organic Rankine cycle using zeotropic mixture of R600a/DME is analysed on the basis of First law of thermodynamics (Energy analysis) and Second law of thermodynamics (Exergy analysis). By this analysis, several graphs are drawn by varying different parameters of ORC and after this comparison is made among them also.The analysis provide a useful picture that clearly shows the best possible thermal efficiency, exergetic efficiency, net work output etc for ORC for various evaporator inlet temperature and for different fraction of mass of Isobutane (R600a) and Dimethylether (DME).
Fig.7 to fig.12 show a variation of net work output of system against temperature of zeotropic mixture at the inlet to the evaporator for six different proportions. Fig.4.13 shows comparison of net work output of system against evaporator inlet temperature of mixture of R600a/DME for six different proportions. By using mixture in ratio of R600a/DME (0.8/0.2), the system gives 100.2kW net work output corresponding to 343K inlet temperature to the evaporator, which is maximum output among different proportions. Analysis revealed that for each different proportion of mixture, system gives the maximum net work output for temperature range of 343K-353K. After 353K net work output of system starts decreasing.
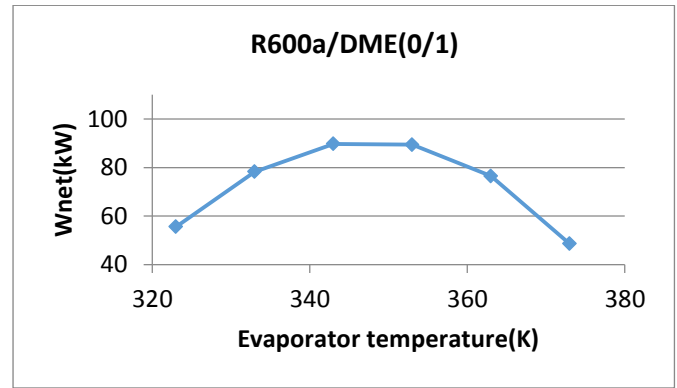

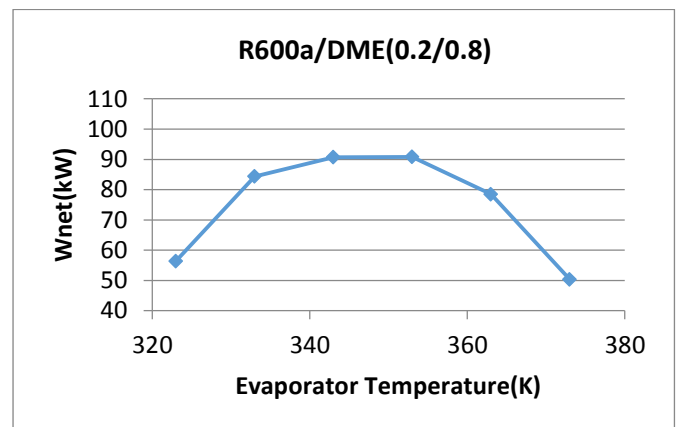Fig.7 Net work output of system with temperature of mixture R600a/DME (0/1) at inlet to evaporator

Fig.8 Net work output of system with temperature of mixture R600a/DME (0.2/0.8) at inlet to evaporator
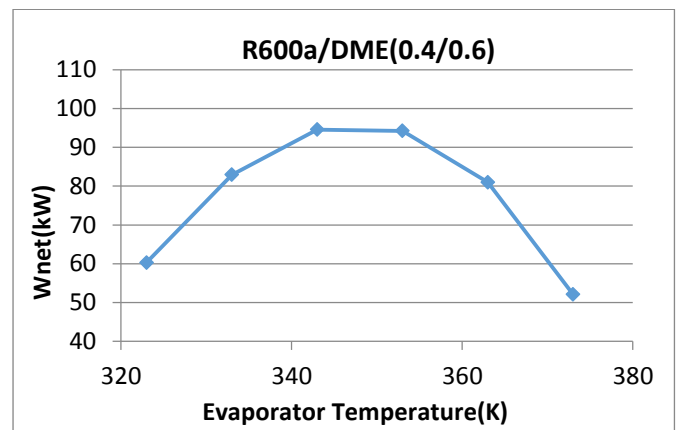

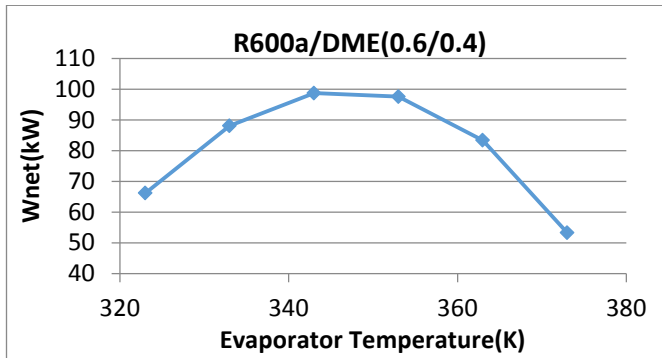Fig.9 Net work output of system with temperature of mixture R600a/DME (0.4/0.6) at inlet to evaporator

Fig.10 Net work output of system with temperature of mixture R600a/DME (0.6/0.4) at inlet to evaporator
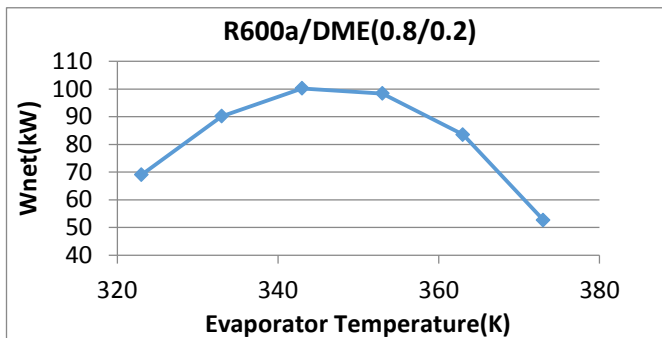

Fig.11 Net work output of system with temperature of mixture R600a/DME (0.8/0.2) at inlet to evaporator
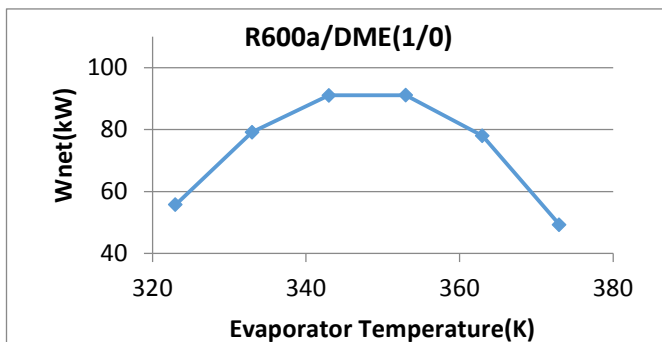

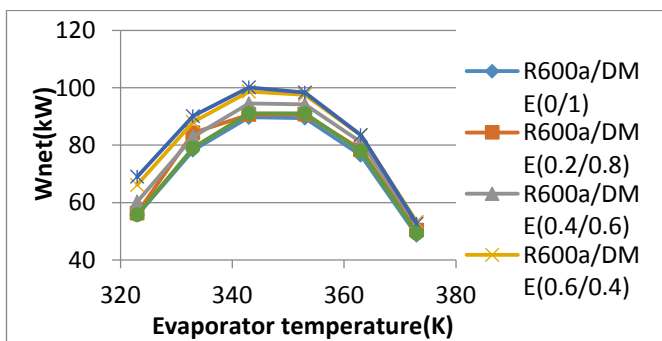Fig.12 Net work output of system with temperature of mixture R600a/DME (1/0) at the inlet to the evaporator

Fig.4.13 Comparison of variation of net work output of system against inlet temperature to evaporator of mixture for six different proportions

Table 3. Comparison of net work output with different inlet temperature to evaporator for each proportion of mixture of R600a/DME

Inlet temperature To evaporator(K)	Net work output (kW)					
	Proportions of mixture of R600a/DME					
	0/1	0.2/0.4	0.4/0.6	0.6/0.4	0.8/0.2	1/0
323	55.61	56.37	60.28	66.19	69	55.76
333	78.3	84.35	82.95	88.13	90.17	79.16
343	89.75	90.69	94.56	98.75	100.2	91.07
353	89.45	90.81	94.24	97.61	98.39	91.1
363	76.52	78.49	81.01	83.41	83.53	77.98
373	48.63	50.37	52.09	53.27	52.67	49.3

Fig.14 to fig.4.19 show a variation of thermal efficiency against inlet temperature of zeotropic mixture to the evaporator for six proportions of mass of R600a/DME (0/1, 0.2/0.8, 0.4/0.6, 0.6/0.4, 0.8/0.2, 1/0) respectively. By increasing inlet temperature of zeotropic mixture to the evaporator, heat rejection reduces and output of turbine increases, due to which thermal efficiency of system continuously increases. Fig.4.20 shows the comparison of thermal efficiency against evaporator temperature for all six proportions. Amongst all the selected proportions, R600a/DME (0.6/0.4) has the maximum thermal efficiency about 12.81% corresponding to 373K temperature of zeotropic mixture at the inlet to evaporator. R600a/DME (0/1) has second best thermal efficiency about 12.74% among these six proportions corresponding to 373K evaporator temperature. R600a/DME (1/0) has least 3.82% thermal efficiency corresponding to 323K temperature at inlet to the evaporator.
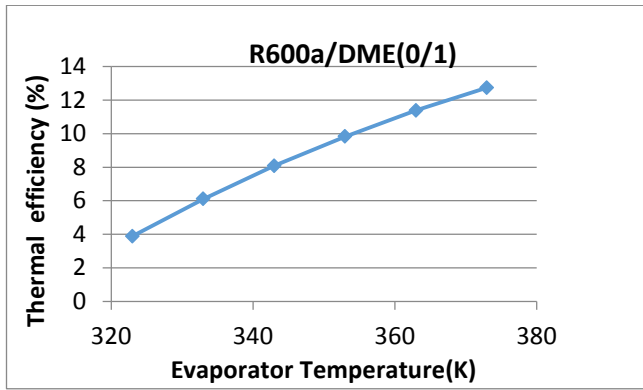
Fig.14 Variation of thermal efficiency with temperature of mixture R600a/DME (0/1) at inlet to evaporator
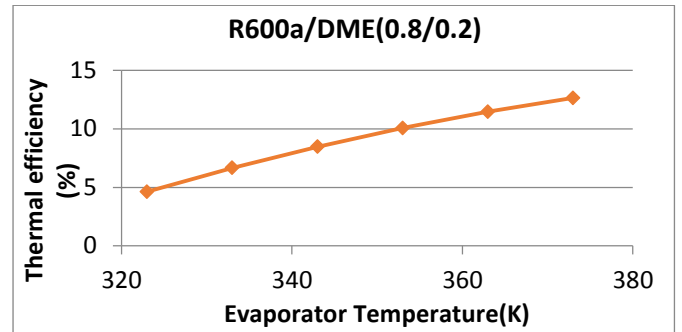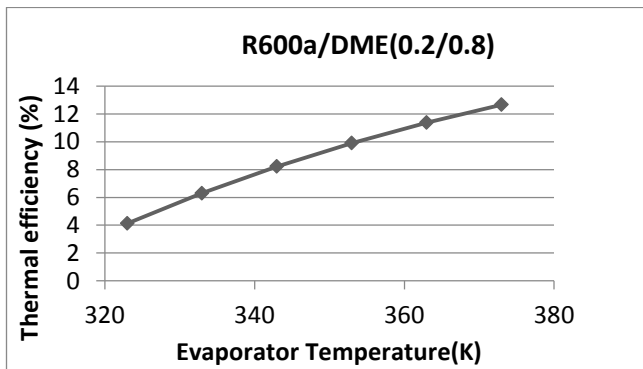

Fig.15 Variation of thermal efficiency with temperature of mixture R600a/DME (0.2/0.8) at inlet to evaporator

Fig.16 Variation of thermal efficiency with temperature of mixture R600a/DME (0.4/0.6) at inlet to evaporator

Fig.4.17 Variation of thermal efficiency with temperature of mixture R600a/DME (0.6/0.4) at inlet to evaporator

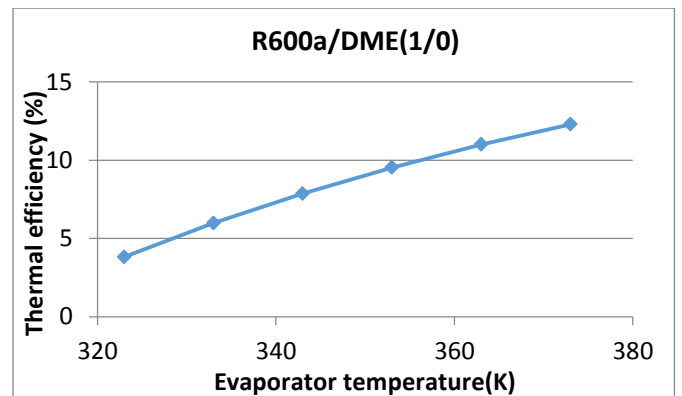Fig.18 Variation of thermal efficiency with temperature of mixture R600a/DME (0.8/0.2) at inlet to evaporator

Fig.19 Variation of thermal efficiency with temperature of mixture R600a/DME (1/0) at inlet to evaporator
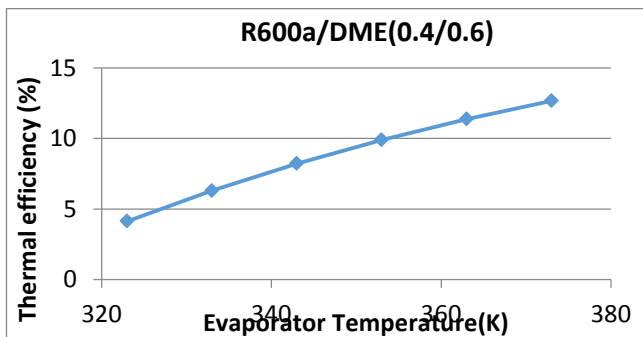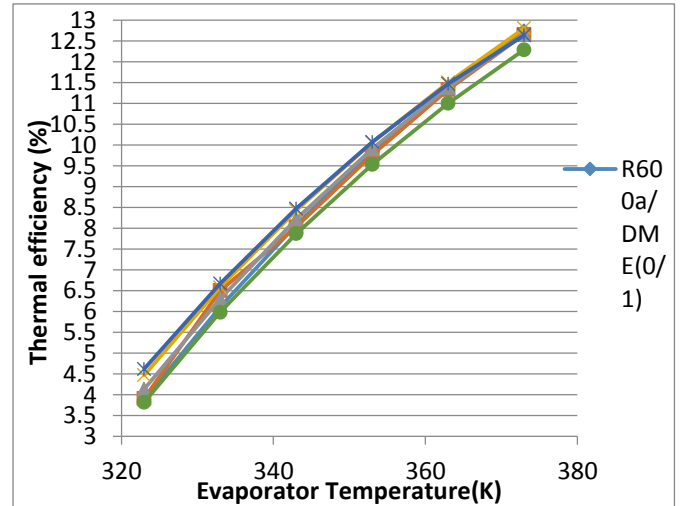

Fig.20 Comparison of variation of thermal efficiency with temperature at inlet to the evaporator of mixture R600a/DME for different proportions
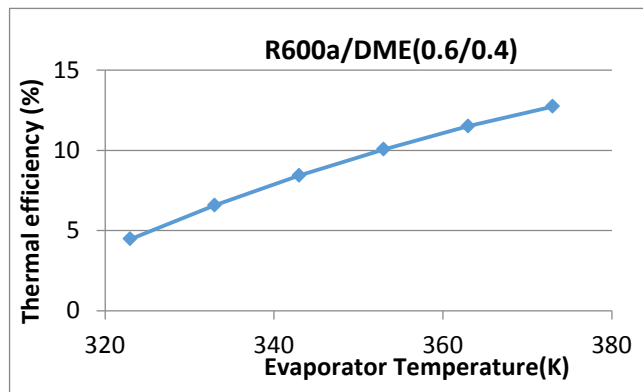
Table4. Comparison of thermal efficiency with different inlet temperature to evaporator for each proportion of mixture of R600a/DME

Inlet temperature To evaporator(K)	Thermal efficiency (%)					
	Proportions of mixture of R600a/DME					
	0/1	0.2/0.4	0.4/0.6	0.6/0.4	0.8/0.2	1/0

323	3.888	3.935	4.135	4.472	4.621	3.822
333	6.108	6.168	6.3	6.581	6.671	5.983
343	8.085	8.128	8.222	8.433	8.475	7.869
353	9.835	9.839	9.905	10.06	10.07	9.531
363	11.39	11.33	11.38	11.5	11.47	11
373	12.74	12.59	12.67	12.81	12.64	12.39

Fig.21 to fig.26 show a variation of exergetic efficiency of system against temperature of mixture (R600a/DME) at the inlet to the evaporator for six different proportions. Fig.27 shows comparison in exergetic efficiency for all six proportions of R600a/DME against evaporator inlet temperature. By increasing inlet temperature to evaporator, exergetic efficiency of system continuously increases upto an optimum temperature. Amongst all selected proportions, R600a/DME (0.6/0.4) has the maximum exergetic efficiency about 54.3% corresponding to 373K inlet temperature to the evaporator. R600a/DME (0.4/0.6) has second best exergetic efficiency about 53.91% corresponding to 373K inlet temperature to the evaporator. R600a/DME (1/0) has least exergetic efficiency of system corresponding to 323K temperature of mixture at the inlet to the evaporator.
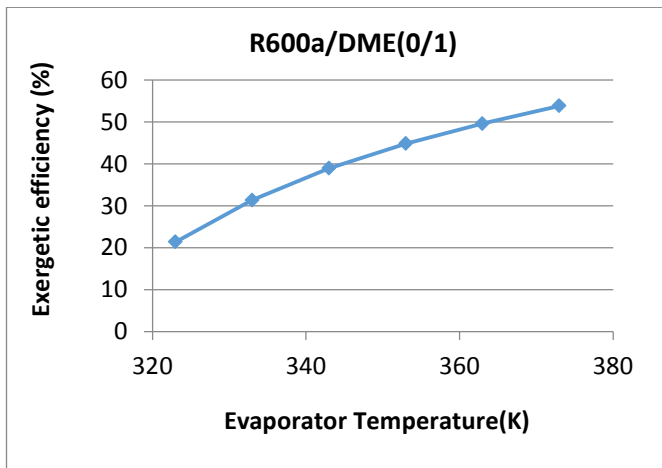
R600a/DME(0/1)

Fig.21 Variation of exergetic efficiency with temperature of mixture R600a/DME (0/1) at inlet to evaporator

R600a/DME(0.2/0.8)

Fig.22 Variation of exergetic efficiency with temperature of mixture R600a/DME (0.2/0.8) at inlet to evaporator

R600a/DME(0.4/0.6)

Fig.23 Variation of exergetic efficiency with temperature of mixture R600a/DME (0.4/0.6) at inlet to evaporator

R600a/DME(0.6/0.4)

Fig.24 Variation of exergetic efficiency with temperature of mixture R600a/DME (0.6/0.4) at inlet to evaporator
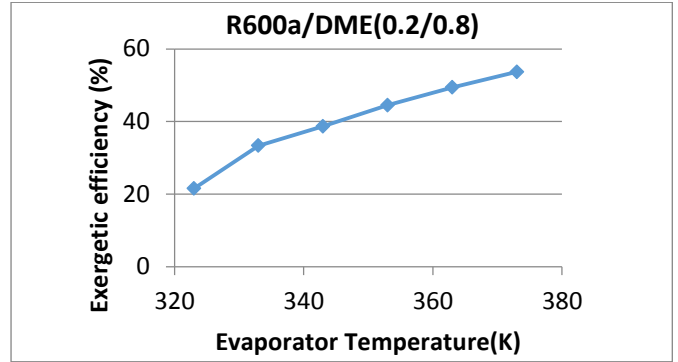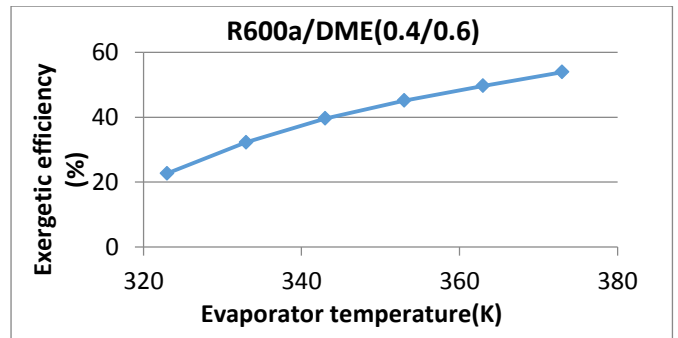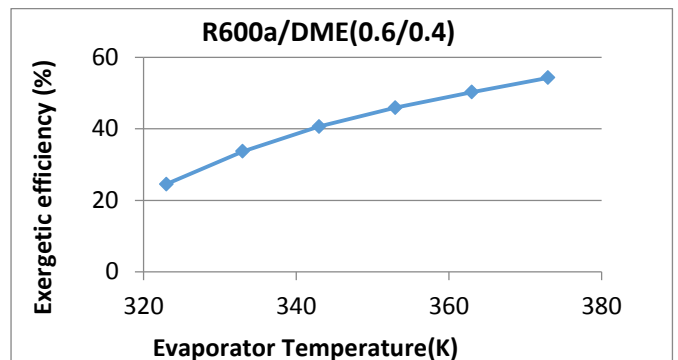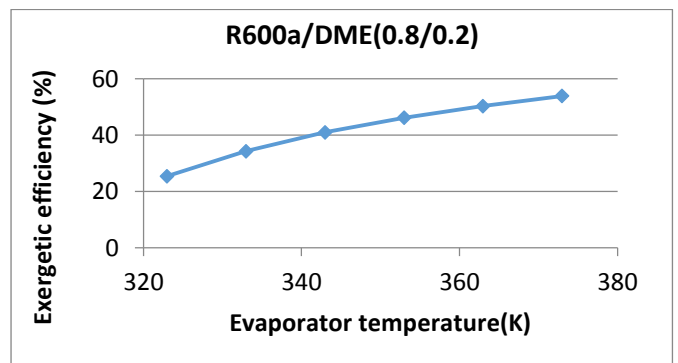
R600a/DME(0.8/0.2)

Fig.25 Variation of exergetic efficiency with temperature of mixture R600a/DME (0.8/0.2) at inlet to evaporator

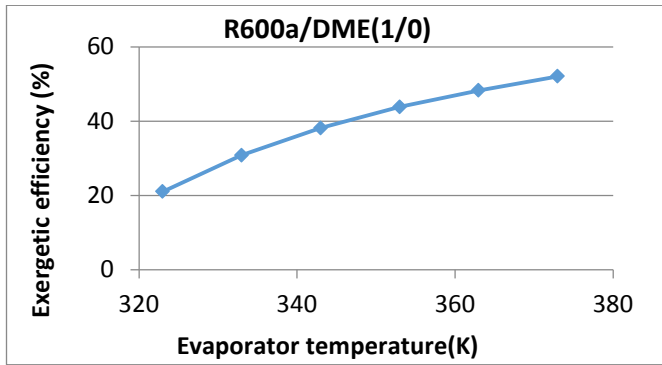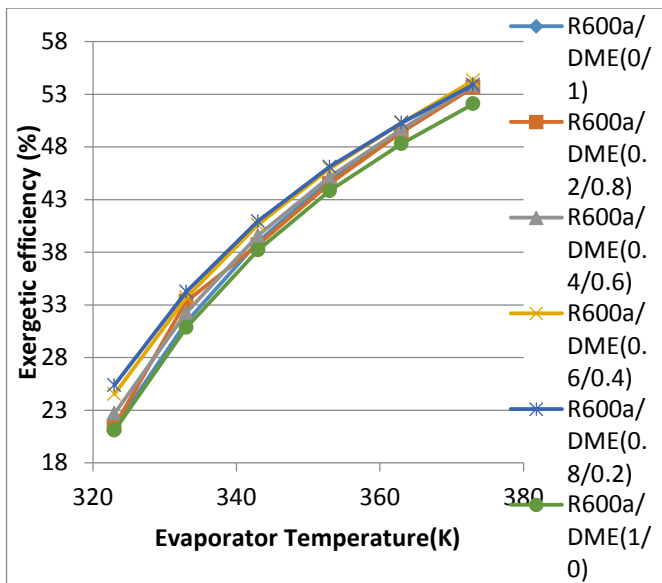Fig. 26 Variation of exergetic efficiency with temperature of mixture R600a/DME (1/0) at inlet to evaporator

Fig.4.28 to fig.4.33 show a variation of irreversibility present in each component of system and total irreversibility present in system against mass fraction of more volatile component, which is Isobutane (R600a) for a range of inlet temperature (323K to 373K) to the evaporator. From the mathematical analysis graphs show that total irreversibility present in the system isapproximately constant for each using mass fraction of isobutane, but irreversibility present in evaporator is minimum, when we select the mass fraction of isobutane is around 0.6 and irreversibility present in condenser is maximum for same mass fraction (around 0.6) of isobutane.If we talk about irreversibility present in pump and turbine, which is approximately constant throughout for each mass fraction of R600a.

Fig.27 Comparison of variation of exergetic efficiency with temperature at inlet to the evaporator of mixture R600a/DME for different proportions

Fig.28 Variation of irreversibility present in system for 323K inlet temperature to evaporator against mass fraction of R600a

Table 5. Comparison of exergetic efficiency with different inlet temperature to evaporator for each proportion of mixture of R600a/DME

Inlet temperature To evaporator(K)	Exergetic efficiency (%)					
	Proportions of mixture of R600a/DME					
	0/1	0.2/0.4	0.4/0.6	0.6/0.4	0.8/0.2	1/0
323	21.39	21.51	22.7	24.53	25.38	21.08
333	31.36	33.34	32.27	33.7	34.24	30.84
343	38.99	38.7	39.58	40.62	40.95	38.19
353	44.87	44.5	45.14	45.94	46.11	43.84
363	49.62	49.39	49.67	50.29	50.3	48.3
373	53.86	53.67	53.91	54.3	53.89	52.09

Table 6. Irreversibility rate in each component of system for different proportions of mixture (R600a/DME) at 323K inlet temperature to evaporator

Different proportions of mixture (R600a/DME)	Irreversibility rate (kW)				
	Total	Evaporator	Condenser	Turbine	Pump
0/1	204.3	148.3	44.60	9.496	1.898
0.2/0.8	205.2	134.7	58.97	9.576	2.019
0.4/0.6	207.4	117.6	78.10	9.507	2.134
0.6/0.4	209.3	111	86.46	9.573	2.213
0.8/0.2	209.2	129	68.54	9.62	2.031
1/0	208.8	151.9	45.75	9.429	1.643

Fig. 29 Variation of irreversibility present in system for 333K inlet temperature to evaporator against mass fraction of R600a

Fig. 29 Variation of irreversibility present in system for 333K inlet temperature to evaporator against mass fraction of R600a

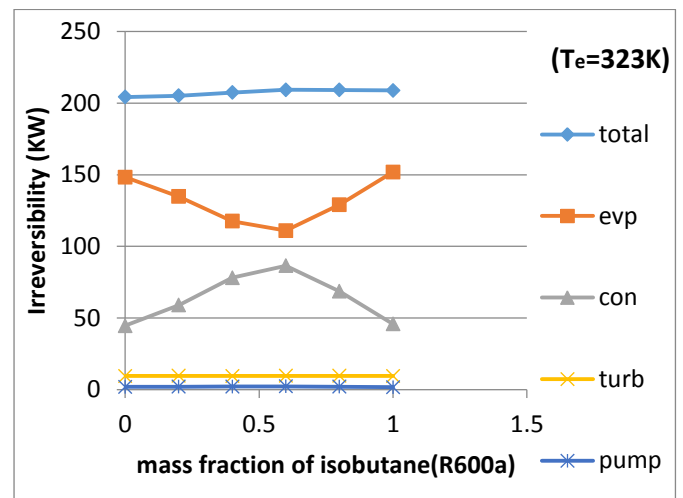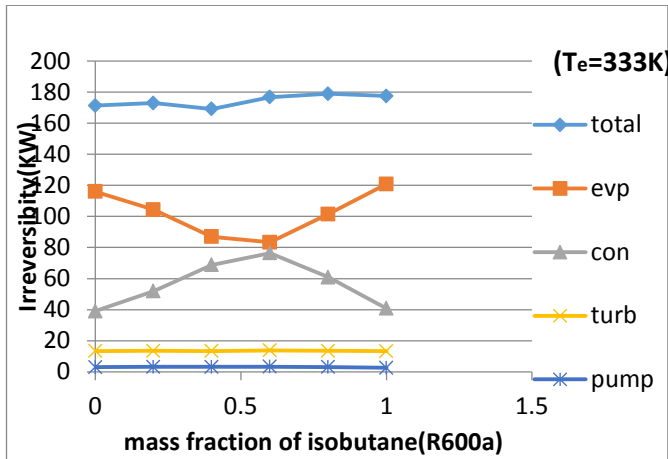| Different proportions of mixture (R600a/DME)	Irreversibility rate (kW)				
	Total	Evaporator	condenser	Turbine	Pump
0/1	171.4	116	39.05	13.37	2.917
0.2/0.8	173	104.4	51.88	13.52	3.140
0.4/0.6	169.2	86.92	68.78	13.39	3.143
0.6/0.4	176.8	83.32	76.4	13.83	3.297
0.8/0.2	179	101.5	60.86	13.52	3.093
1/0	177.5	120.9	40.82	13.30	2.515
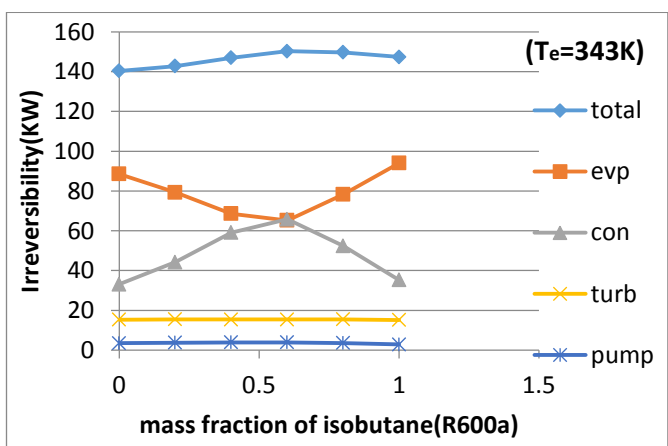
Fig.30 Variation of irreversibility present in system for 343K inlet temperature to evaporator against mass fraction of R600a

Table 8. Irreversibility rate in each component of system for different proportions of mixture (R600a/DME) at 343K inlet temperature to evaporator

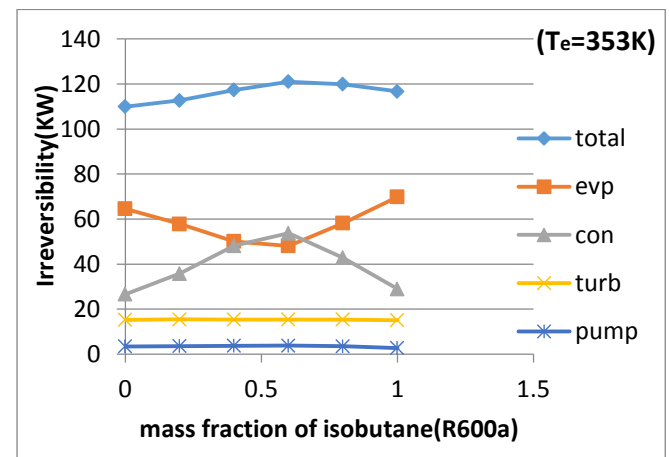| Different proportions of mixture (R600a/DME)	Irreversibility rate (kW)				
	Total	Evaporator	condenser	Turbine	Pump
0/1	140.4	88.52	33.1	15.33	3.495
0.2/0.8	142.8	79.29	44.23	15.51	3.726
0.4/0.6	147	68.58	59.12	15.42	3.889
0.6/0.4	150.3	65.2	65.76	15.44	3.939
0.8/0.2	149.7	78.22	52.4	15.49	3.593
1/0	147.4	94.02	35.32	15.19	2.897

Fig. 31 Variation of irreversibility present in system for 353K inlet temperature to evaporator against mass fraction of R600a

Table9. Irreversibility rate in each component of system for different proportions of mixture (R600a/DME) at 353K inlet temperature to evaporator

Different proportions of mixture (R600a/DME)	Irreversibility rate (kW)				
	Total	Evaporator	condenser	Turbine	Pump
0/1	109.9	64.59	26.6	15.27	3.433
0.2/0.8	112.7	57.83	35.8	15.49	3.62
0.4/0.6	117.3	50.09	48.14	15.39	3.71
0.6/0.4	121	48.09	53.67	15.40	3.827
0.8/0.2	119.9	58.18	42.84	15.42	3.497
1/0	116.7	69.87	28.97	15.10	2.761

Fig. 32 Variation of irreversibility present in system for 363K inlet temperature to evaporator against mass fraction of R600a

Table10. Irreversibility rate in each component of system for different proportions of mixture (R600a/DME) at 363K inlet temperature to evaporator

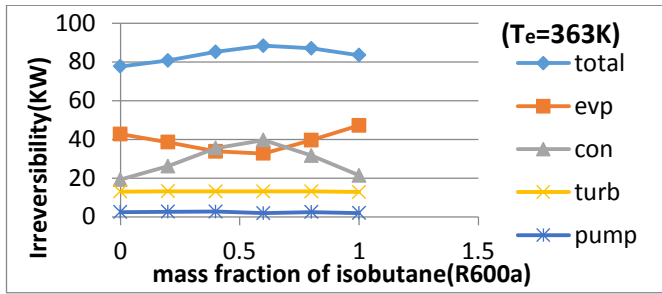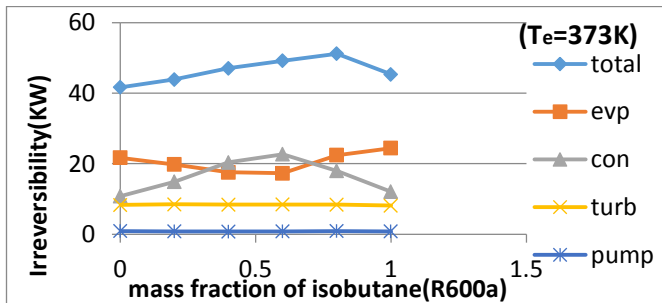| Different proportions of mixture (R600a/DME)	Irreversibility rate (kW)				
	Total	Evaporator	condenser	Turbine	Pump
0/1	77.69	42.77	19.32	13.07	2.532
0.2/0.8	80.7	38.54	26.24	13.28	2.636
0.4/0.6	85.23	33.78	35.56	13.17	2.721
0.6/0.4	88.35	32.74	39.68	13.19	1.932
0.8/0.2	87.01	39.64	31.64	13.17	2.548
1/0	83.45	47.17	21.40	12.85	2.023

Fig.33 Variation of irreversibility present in system for 373K inlet temperature to evaporator against mass fraction of R600a

Table11. Irreversibility rate in each component of system for different proportions of mixture (R600a/DME) at 373K inlet temperature to evaporator

Different proportions of mixture (R600a/DME)	Irreversibility rate (kW)				
	Total	Evaporator	condenser	Turbine	Pump
0/1	41.66	21.68	10.81	8.304	0.872
0.2/0.8	43.91	19.77	14.87	8.486	0.783
0.4/0.6	47.05	17.57	20.38	8.368	0.731
0.6/0.4	49.19	17.26	22.69	8.421	0.811
0.8/0.2	51.21	22.42	17.96	8.361	0.911
1/0	45.34	24.40	12.06	8.093	0.791

5. CONCLUSION

An extensive First law (energy) and Second law (exergy) analysis of geothermal organic Rankine cycle using zeotropic mixture of R600a and DME in different mass proportions has been presented.
Conclusions are summarised as follows:

1. Thermal efficiency (First law efficiency) and Exergetic efficiency (Second law efficiency) of organic rankine cycle increases with temperature of mixture at inlet to the evaporator.
2. Mixture ratio R600a/DME (0.8/0.2) gives maximum net work output corresponding to 343K inlet temperature to evaporator.
3. Irreversibility present in evaporator is minimum at optimal mass proportion of mixture R600a/DME (0.6/0.4) and irreversibility in condenser is maximum corresponding to mass proportion R600a/DME (.6/.4).
4. By this analysis, it is found that total irreversibility present in system is approximately constant for each mass ratio of mixture.
5. By this analysis, also found that mass flow rate of mixture goes on increasing with inlet temperature to evaporator due to which size of turbine increases. Therefore R600a/DME (0.6/0.4) can select as optimal mixing ratio for each inlet temperature to evaporator.
6. R600a/DME (0.8/0.2) can be considered as best option among six different mass proportions because this mass ratio gives the maximum net work output corresponding to an optimal evaporating temperature. Amongst all selected proportions, R600a/DME (0.6/0.4) has maximum thermal efficiency corresponding to 373K inlet temperature to evaporator and R600a/DME (0.6/0.4) has maximum exergetic efficiency corresponding to 373K inlet temperature to evaporator also.

Further investigation should be carried out of regenerative and superheated ORC by using this zeotropic mixture. Actual organic Rankine cycle should be studied for this system by considering pressure drop in condenser and evaporator. By varying the evaporator and condenser pressure of system, the First law and Second law analysis should be studied.

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