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ABSTRACT

In numerous Industrial setups such as Power Plants, Steel Plants, Food Processing, etc., a considerable amount of waste heat is available which goes into the environment to contribute to Global Warming and Thermal Pollution. In this research investigation Heat Recovery from Gas Power Cycle (GPC) exhaust has been objectified to operate Ejector Refrigeration Cycle (ERC) through the superconductor of heat i.e., Loop Heat Pipes (LHP). In LHP fluids like Acetone, Ethanol, Methanol & Water, and in ERC new eco-friendly refrigerants such as R236ea, R1224yd (Z), R1233zd (E), R245fa, R365mfc along with R718 have been selected for study as the working fluids. Results obtained from the software-based mathematical modeling have been presented and an assessment of this novel system for industrial viability has been done. It has been observed that the boiler temperature of ERC can be maintained near or above the critical temperature of the Eco-friendly refrigerants working fluids. COP & refrigeration capacity of the combined system has been obtained in the range of 0.25-0.28 and 10.35kW-315kW respectively for various combinations of fluids. Based on the eco-friendliness, compactness & industrial viability the system with Water (LHP)-R1224yd (Z) (ERC) has been recommended as the mass flow required for working fluids is the least & utilization of Heat Input Available to ERC is maximum for the operations.

KEYWORDS


INTRODUCTION

The world has seen numerous methods of refrigeration across the millenniums. From the use of Ice & low temperatures at night to the modern huge HVAC plants, there has been extraordinary development in the field of refrigeration technology. The most common and most efficient refrigeration system is the Vapour Compression Refrigeration System for it works on High-Grade Energy & has the highest Coefficient of Performance (COP) of all. However, to generate this high-grade energy extremely high costs are incurred in the power plants along with wastage of Energy, Exergy, Fuel, and the Generation of all sorts of pollution. It can be simply stated that everything comes at a cost. One of such power plants is the Gas Power Plant (GPP) working on Brayton Cycle which we all are well acquainted with. Even after the regenerator of the GPP, a considerable amount of energy remains in form of the exhaust gases. Hence, a Highly Flexible & Highly Efficient Superconductor Loop Heat Pipe Heat Exchanger (LHP HEx) can be incorporated to harness the waste heat and utilize it in other systems which can satisfactorily work on Low-Grade Energy reducing thermal pollution and global warming.

From Figure 1, the LHP can be explained as a device that can be used to transfer heat from one point to another point over a long distance with the help of an insulated vapor line and liquid line. It works on the phenomenon of Evaporation-Condensation. It can be simplified to be similar to Power Cycle less the Turbine & Pump; the remaining components of the power cycles such as the evaporator & the condenser are connected through thermally insulated lines called vapor lines & liquid lines and the evaporator is in porous wicked construction. A
working fluid is filled under pressure within the LHP system, which after receiving heat evaporate and moves out of the evaporator through the porous structure, it enters the transfer lines (Vapour Line & Liquid Line) to move to the condenser part, where the fluid from evaporator condenses and returns to the evaporator owing to the buoyancy, hence, completing the cycle. Hence no input work is required here to transfer heat from one point to another distant point. More than one LHP Ex. May also be suggested for one system to take the complete heat load.

In search of such a system that can work on waste heat and generates desired effects, we came across the Ejector Refrigeration System (ERS) working on new eco-friendly refrigerants which have a relatively low boiling point, hence, the waste energy at relatively low temperature can be harnessed. It has a complicated operation in which converging-diverging nozzles, flash evaporation and refrigeration can be observed to take place. Flash evaporation of ERS & flash evaporation in VCRS both occur due to sudden pressure drop, however, the causes of pressure drops are different for the two.

In the following, advancements have been studied on ERS & LHP:

Owing to the requirement of heat at low temperatures only, ERS has been attached with many systems including solar power sources. Through the literature review followings, advancements have been studied on ERS & LHP: Untea et. al. [1] performed energy and exergy investigation of an ejector refrigeration system working on...
different working fluids. For 4 working fluids water, methanol, ammonia, and R134a, the best performances were achieved for water. The optimum value was reached for Boiler Temperature 140°C, Condenser Temperature 30°C, Evaporator Temperature 5°C which resulted in a COP of 0.48 and η_ex of 0.085. Ebadollahi et. al. [2] used multi-parallel ejector and low-temperature heat sources in the Ejector Refrigeration Cycle (ERC) to evaluate the first and second law analysis keeping the system running at optimum conditions under various working conditions. The maximum and minimum coefficient of performance (COP) was recorded at 0.344 for R152a and 0.285 for R236fa, respectively, whereas the maximum and minimum exergy efficiencies were 33.98% for R152a and 27.63% for isobutene, respectively. Besagni et. al. [3] performed an overall review of the ERS system working on low GWP Refrigerants.

Akkurt et. al. [4] analyzed a solar-assisted ejector cooling system having varying ejector area ratios such as Ar = 6.56, Ar = 7.17, and Ar = 7.86. Exergy destruction proportions found were 42.9%, 44.7%, and 45.2% in the ejector, 10.9%, 9.1%, and 10.1% for generator, 7.6%, 7.7%, and 8.7% for condenser, 5.9%, 5.6%, and 5.8% for evaporator of total cooling subsystem for different ejector area ratios. Besagni et. al. [5] performed another comprehensive study and review of the ERS. An overview of ejector technology, relationships among the working fluids, and the ejector performances, with a focus on past, present, and future trends, were presented.

Liu [6] studied a performance improvement potential analysis for a booster-assisted ejector refrigeration system. Outcomes showed that 61.6% of the system exergy destruction occurs due to the components. Also, 55.5% of the overall exergy destruction may be avoided by improving component efficiencies. Memet et. al. [7] investigated theoretically an Ejector Refrigeration System, started with a typical ejector design. For the chosen input conditions, it was revealed that the Coefficient of Performance increased with the increase of the boiler temperature with the best COP being 0.178. Taleghani et. al. [8] analyzed the exergy of a CO2 (R744) two-phase ejector using a 1D model for single and double choking conditions.

It was observed that the transiting exergy flow had an important effect on the exergy analysis of the system. It was also observed that the Grassmann exergy efficiency was not the suitable standard for evaluating the performance of a trans-critical CO2 ejector. Al-Sayyab et. al. [9] analyzed a compound PV/T waste heat driven ejector-heat pump system for simultaneous data center cooling and waste heat recovery for district heating. The compressor contributed the largest exergy destruction source at 26%, the boiler was the lowest source at 2%. Moreover, advanced exergy analysis indicated that 59.4% of the whole system may be avoided. The condenser has the highest & the ejector has the lowest exergoeconomic factors. Sharma et. al. [10] applied exergy analysis to each component of the system. The refrigerant R1234yf had been used and the results exhibited that maximum exergy destruction in the generator followed by ejector and other components. Kumar et. al. [11] investigated the performance of the ejector refrigeration system working on R-134a. It was observed that an optimum area ratio of the ejector was required for better performance.

Moreover, a higher value of area ratio gives higher COP for a lower value of critical condenser pressure. The critical condenser pressure for the area ratio 10.08 was 778.9 kPa, whereas, it was 916.72 kPa for the ejector of area ratio 6.451 while boiler and evaporator temperature were kept at 80 °C and 15 °C respectively. Ebadollahi et. al. [12] conducted a theoretical analysis of the triple-evaporator ejector refrigeration cycle (TEERC) for triples applications of cooling, freezing, and ventilation, with Nine appropriate working fluids i.e., R717, R152a, R134a, R290, cis-2-butene, butane, isobutene, isobutane, R236fa. The maximum & minimum COP were observed for R717 and R236fa at 0.333 and 0.268, respectively. The maximum & minimum exergy efficiencies were considered for R717 and isobutene at 21.43% and 12/51 %, respectively. Ventilation, cooling, and freezing capacities recorded were 11.68 kW, 3.86 kW & 1.904 kW, respectively. Reddy [13] presented a numerical analysis of ejectors optimizing operating evaporator temperature, condenser temperature, and generator temperature with R245fa as the working fluid. Li et. al. [14] chose refrigerants R1234yf and R1234ze(E). R1234yf had a greater entrainment ratio than R134a and R1234ze(E).

Besana et. al. [15] proposed screening of refrigerants based on an integrated Computational Fluid Dynamic (CFD) - Lumped Parameter Model (LPM) approach. Performances for different refrigerants were obtained through a validated CFD approach, whereas, the cycle was modeled by a Lumped Parameter Model. For Different refrigerants, the energy performances were calculated and the effects of the “component-scale” on the
“system-scale” were investigated. Mishra [16] used HFO refrigerants in the ejector refrigeration system and comparisons with other HFC and HCFC refrigerants were presented. Dwivedi et. al. [17] performed investigation with eco-friendly refrigerants R-404A, R-410A, R-407C, R-423A, R-500, R-502 and R-507C. The maximum first Law Efficiency ($\eta_1$) was in the range of 1.5-1.8 with fluid R-404A and at condenser temperature 328K. At 253K evaporator temperature, the Second Law Efficiency ($\eta_2$) was 37% for R-404A. The highest COP was associated with R-404A with subject temperature variations. He et. al. [18] designed and experimented with neon Cryogenic LHP with infrared point-to-point heat transfer elements in future space applications. The results from the experiments exhibited that the supercritical startup was realized effectively at cases of 1.5 W secondary evaporator power, unsuccessful when 0.5 and 1W heat load was subjected to the second evaporator.

The maximum heat transport capacity of the primary evaporator was in the range of 4.5and 5 W with the proper auxiliary heat load. Hossain et. al. [19] performed an experimental investigation on the loop performance at different heat loads. Thermocouples were installed on LHP to gauge the temperature record at various locations. The minimum thermal resistance of LHP recorded was 0.78 °C/W for a heat load of 100 W, whereas, the maximum was 3.1 °C/W for a heat load of 20 W. The maximum heat transfer coefficient in the evaporator was recorded as 14114 W/m2 for a heat load of 100 W. Setyawan et. al. [20] aimed to manufacture a loop heat pipe uses capillary wick copper sintered with centrifugal casting. The filling ratio was also varied for the investigation FR: 50%, 60%, and 80%, with 60% giving the best performance. Martvovová et. al. [21] worked on increasing the efficiency of fireplace inserts by preheating the combustion air with the heat from the flue gases through an LHP. Anikivi et. al. [22] executed an experimental investigation and to evaluate the performance of Flat loop heat pipe (LHP) with stainless-steel mesh wick. Different fill ratios such as 40%, 50%, 60% were used subject to different heat loads of 20 W, 40 W, 60 W, 80 W, and 100 W, using De-Ionized water as fluid. Results exhibited that with forced water-cooling conditions, the LHP can transfer a maximum heat load of 100 W.

The evaporator temperature was recorded at 92°C, with minimum thermal resistance at 0.5332 °C/W. Gai et. al. [23] established that the temperature hysteresis of the LHP was connected to the gas-liquid distribution in the compensation chamber (CC). It relied on the interaction between the heat leak of the evaporator and the reflux liquid from the condenser. The temperature of the LHP evaporator increased as the vapor phase in the compensation chamber began to increase. Shukla [24] presented a thermo-fluid dynamic model for the transient operations and analyzed the thermo-fluid dynamic feature of a Stainless Steel/Ammonia. Buz et. al. [25] prepared a mathematical model and performed computational analysis on the processes in the loop heat pipe to avoid auto-oscillation and increasing of the efficiency and temperature. Shioga et. al. [26] conducted an experimental study measuring the temperature of each division of the thin LHP aligned with the subjected heat input evaluating the heat transfer capability. The Thermal resistance between the evaporator and the condenser was found to be 0.11 K/W for horizontal orientation. For a bottom heat orientation, it was 0.03 K/W and 0.28 K/W for a top heat orientation at 20W. Nagano et. al. [27] performed a comprehensive test program with start-up, power step-up power cycle, and low power. Korn [28] reported the major working principles and the most important potential to calculate heat. Dwivedi et. al. [29] studied the viability of LHP to be used for intra-cycle heat exchange for the refrigeration cycles. Hirase et. al. [30] investigated the thermal performance of a loop heat pipe having two evaporators and two condensers applying a lumped network model. The Results exhibited that the vapor and liquid flow rates and the thermal conductance of the heat pipe varied considerably based on the distribution ratio of the heating rate of the evaporators.

**GAP IN THE LITERATURE**

The review of the previous investigations suggests that there is a huge scope of using both LHP & ERC for utilizing the waste heat from power plants or steel plants. Hence, an industrially viable eco-friendly refrigeration unit can be objectified which benefits the environment and well as the industry in cost savings on multiple fronts. In this research analysis, the combining of GPC to ERC is done through LHP HEx. The limits of the cycle have been fixed along with the selection of working fluids. The objective of this investigation is not to study the GPC, LHP, or ERS for sufficient search that has already been published on the First Law & Second Law analysis of these systems. The sole objective of this research work is to study the industrial feasibility of
such an eco-friendly novel combination of cycles through a software-based equation solver. Moreover, mass flow ratio in the ERS, Pressure of Condenser of ERS, and others have been chosen based on the available literature and standard texts. Peak Temperature of GPC, Pressure of the Boiler of ERS, the mass ratio of ERS have been made input parameters and other performance parameters have been recorded through the equation solver and have been presented.

MATERIALS AND METHODS

As summarized in the previous section, in this research investigation a Gas Power Cycle (GPC) has been combined with an Ejector Refrigeration Cycle (ERC) through a Loop Heat Pipe Heat Ex. In this section, a basic overview of the operation of this cycle has been presented regarding Fig 3. The Topping cycle is the GPC working of the standard working fluid (Air) which has a Reheater & Regenerator in addition to other compulsory parts such as Compressor, Generator & Gas Turbine. The exhaust gas from the regenerator has been routed through the Evaporator of the LHP Ex. back to the Compressor. The heat of the exhaust is absorbed by the LHP fluid while evaporating and the vapors through the wicked structure move to the condenser, from where the liquid generated after vapor releasing heat to the boiler of the ERC returns to the Evaporator. The working fluids for the LHP HEx have been chosen from the easily available fluids such as Water, Ethanol, Methanol, and Acetone. It has been studied how these working fluids affect the performance of the components as well as the combined cycle.

Figure 3. Schematic of Ejector Refrigeration System Combined with Organic Vapour Power Cycle through Loop Heat Pipes

From Figure 3 is can be seen that the vapor generated in the boiler is expanded in the Ejector Nozzle (Fig 3a) at 11, causing the sudden drop in of the pressure at the converging section which causes the liquid inside the flash chamber to evaporate and move into the ejector nozzle at 15 resulting into the cooling of the remaining liquid in the flash chamber. This cool liquid at 16 is throttled down to the Evaporator at 17 which maintains the desired low temperature. The fluid coming out of the evaporator at 18 is pumped back to the flash chamber in form of
spray at 19 to cool down the liquid further. The fluid coming out of the mail ejector nozzle at 12 is pumped back to the boiler at 10 through the condenser at 13. Hence, 2 parallel cycles operate in the ERC.05 New Eco-friendly refrigerants namely R236ea, R1224yd (Z), R1233zd (E), R245fa, R365mfc along with 1 established refrigerant Water (R718) for standardization purpose have been chosen for the investigation so that the objective of environmental protection and conservation has been strengthened. The corresponding T-s plot can be seen in Fig 4.

![Figure 3a. The ejector of the ERS [32]](image)

![Figure 4. T-s Plot for Gas Power Cycle Combined with Organic Vapour Power Cycle through Loop Heat Pipes.](image)

Tables 1 & 2 contain the 10-working fluids being used in the combined system. Table 1 has all the refrigerants being considered for the ERS whereas table 2 has the working fluids being considered for the LHP Hex. It needs to be noticed that the GWP & GDP for R1224yd (Z) is 0, hence, the most attractive from the list. Moreover, it can be seen that the critical temperatures for the refrigerants fall in the same line whereas for water it is quite high. Hence, it needs to be pointed out that ERS working on these refrigerants can work on even low-temperature heat sources. Hence, the peak temperature of the GPC may be kept even in the range of 500K. The chiller temperature has not been altered keeping because of the operating temperature of the Water.
Table 1. Combination of working fluids for ORC

<table>
<thead>
<tr>
<th>ORC Working Fluid</th>
<th>R236ea</th>
<th>R1224yd(Z)</th>
<th>R1233zd(E)</th>
<th>R245fa</th>
<th>R365mfc</th>
<th>R718</th>
</tr>
</thead>
<tbody>
<tr>
<td>Critical Temperature (K)</td>
<td>412.4</td>
<td>428.7</td>
<td>438.8</td>
<td>427</td>
<td>460</td>
<td>674.14</td>
</tr>
<tr>
<td>Ozone Depletion Potential (ODP)</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>Global Warming Potential (GWP)</td>
<td>1200</td>
<td>&lt;1</td>
<td>4.5</td>
<td>1030</td>
<td>794</td>
<td>0</td>
</tr>
</tbody>
</table>

Table 2. Combination of working fluids for LHP-Ex

<table>
<thead>
<tr>
<th>LHP Working Fluid</th>
<th>Acetone</th>
<th>Ethanol</th>
<th>Methanol</th>
<th>Water</th>
</tr>
</thead>
<tbody>
<tr>
<td>Critical Temperature (K)</td>
<td>508K</td>
<td>516.25</td>
<td>512.6K</td>
<td>674.14</td>
</tr>
</tbody>
</table>

METHODOLOGY AND EQUATIONS USED

The GPC Peak temperature has been varied suitably from 600 K to 1000K. The pressure ratio has been kept at 5 with the assumption of perfect inter-cooling & reheating. The values of the Efficiencies & Effectiveness for the different components of the complete combined cycle such as Compressors, Turbines, Nozzles, Diffusers, etc. have been suitably assumed with the help of referred previous research works and textbooks. The condenser temperature has been kept at 323K; the Evaporator Temperature has been kept at 278K. The mass ratio has been varied from 2 to 7. The Boiler temperature will be dependent on the Condenser temperature of the LHP Ex. The state of Fluid out of the boiler has been assumed as Saturated Vapor. The equations available from texts and literature available have been mathematically modeled into the Engineering Equation Solver to run the desired iterations to obtain the results. The results have been displayed in form of comparative graphs based on the eco-friendly working fluid combinations and input pressure & temperature to the cycle to ascertain the industrial feasibility.

Gas Power Cycle [35]

Isentropic compression the work required for the compressor is given by

\[
\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \frac{T_5}{T_6}
\]  

(1)

\[
T_3 = T_1 + \frac{(T_2 - T_1)}{\eta_T} & T_7 = T_5 - \frac{(T_5 - T_6)}{\eta_T}
\]  

(2)

Where \(\eta_T\) can be taken as 0.92 & \(\eta_T\) can be taken as 0.86

\[
W_{\text{InB}} = H_3 - H_1 = \dot{m}_a c_p \left(T_3 - T_1\right)
\]  

(3)

Isobaric heat addition the net heat added is given by

\[
Q_{\text{add}} = H_5 - H_4 = \dot{m}_a c_p \left(T_5 - T_4\right) = \dot{m}_a CV\ (\text{Calorific Value of Fuel}= 43 MJ)
\]  

(4)

Isentropic expansion the work done by turbine is given by

\[
W_{\text{outB}} = H_7 - H_5 = \dot{m}_g c_p \left(T_7 - T_5\right)
\]  

(5)

\[
W_{\text{NetB}} = W_{\text{outB}} - W_{\text{InB}}
\]  

(6)

\[
\eta_B = \frac{W_{\text{NetB}}}{Q_{\text{add}}}
\]  

(7)

Heat Exchanged in the Regenerator

\[
Q_R = H_3 - H_1 = \dot{m}_a c_p \left(T_3 - T_1\right)
\]  

(8)

Where, \(T_4 = T_3 + \epsilon_R \left(T_7 - T_3\right)\)

(9)

Isobaric heat rejection (in a heat exchanger), the net heat rejected is given by

\[
Q_{\text{Rej}} = H_8 - H_1 = \dot{m}_g c_p \left(T_8 - T_1\right)
\]  

(10)

Where \(T_8 = T_7 - \epsilon_R \left(T_7 - T_3\right)\) & \(\epsilon_R\) can be taken as 0.75

(11)
Table 3. List of parameters & respective abbreviations for GPC

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Abbreviations/ Symbols</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Gas Power Cycle</strong></td>
<td></td>
</tr>
<tr>
<td>Inlet Temperature (K) &amp; Pressure (kPa) to Compressor of GPC</td>
<td>$T_1, P_1$</td>
</tr>
<tr>
<td>Outlet Temperature (K) &amp; Pressure (kPa) to Compressor of GPC</td>
<td>$T_3, P_1$</td>
</tr>
<tr>
<td>Inlet Temperature (K) &amp; Pressure (kPa) to Combustion Chamber of GPC</td>
<td>$T_4, P_2$</td>
</tr>
<tr>
<td>Inlet Temperature (K) &amp; Pressure (kPa) to Turbine of GPC</td>
<td>$T_5, P_2$</td>
</tr>
<tr>
<td>Inlet Temperature (K) &amp; Pressure (kPa) to Regenerative Heat Exchanger of GPC</td>
<td>$T_9, P_1$</td>
</tr>
<tr>
<td>Inlet Temperature (K) &amp; Pressure (kPa) to Loop Heat Pipe Connecting the Power Cycles</td>
<td>$T_8, P_1$</td>
</tr>
<tr>
<td>Work Input in the GPC (kW)</td>
<td>$W_{inB}$</td>
</tr>
<tr>
<td>Heat Input in the GPC (kW)</td>
<td>$Q_{add}$</td>
</tr>
<tr>
<td>Work Output in the GPC (kW)</td>
<td>$W_{outB}$</td>
</tr>
<tr>
<td>Net-Work Output in the GPC (kW)</td>
<td>$W_{netB}$</td>
</tr>
<tr>
<td>Heat Exchange between GPC &amp; LHP in kW</td>
<td>$Q_{GP-LHP}$</td>
</tr>
<tr>
<td>Flow Rate of Air in GPC (kg/s)</td>
<td>$\dot{m}_a$</td>
</tr>
<tr>
<td>Flow Rate of Gas in GPC (kg/s)</td>
<td>$\dot{m}_g$</td>
</tr>
<tr>
<td>Pressure Ratio</td>
<td>$r$</td>
</tr>
<tr>
<td>Heat Exchange in Regenerator in kW</td>
<td>$Q_R$</td>
</tr>
<tr>
<td>Turbine &amp; Compressor Efficiency</td>
<td>$\eta_T &amp; \eta_B$</td>
</tr>
</tbody>
</table>

**Loop Heat Pipes** [31]

$Q_{GP-LHP} = m_g c_p (T_8 - T_1)$  \hspace{1cm} (12)

$\dot{E}_{GP-LHP} = m_g (s_8 - h_1) - T_o m_g (s_s - s_1)$  \hspace{1cm} (13)

Where, $s_s - s_1 = c_p (T_8 - T_1) - R_g \ln(P_8/P_1)$  \hspace{1cm} (14)

Effectiveness of LHP Evap-GPC Heat Exchanger $\varepsilon_{LHPEv-GPC} = 0.75$

Effectiveness of LHP Cond-ORC Heat Exchanger $\varepsilon_{LHPCond-ORC} = 0.85$

Figure of Merit for comparing different working fluids for LHP:

$M_{Fh} = \rho \sigma h_\delta / \mu_l$  \hspace{1cm} (15)

The axial heat flow rate due to the sonic limitation was calculated from the following equation

$Q_S = \pi r_p^2 \rho_s h_\delta \sqrt{\frac{\gamma R T}{2(\gamma + 1)}}$  \hspace{1cm} (16)

The maximum heat transfer due to the entrainment limit was determined using the equation

$Q_{Ent} = \pi r_p^2 h_\delta (2 \pi \rho_l \sigma \cos \phi / \lambda)$  \hspace{1cm} (17)

The maximum heat transfer due to the capillary limit was determined using the equation

$Q_{Cap} = (M_{Fh})(A_w k_{hp} / L_{eff}) \{ (2/r_p) - (\rho_l L_{eff} \sin \phi / \sigma) \}$  \hspace{1cm} (18)

The degree of superheat to cause nucleation is given by:

$\Delta T = 3.06 \sigma_{LHHP} / \rho_l h_\delta \delta$  \hspace{1cm} (19)

With reference to mass flow rate of warm and cool air, the heat transfer rates to the evaporator and condenser sections are calculated as follow:

$q_e = m_c (T_{e, in} - T_{e, out}) = \rho_u A c_p (T_{e, in} - T_{e, out})$  \hspace{1cm} (20)
\( q_c = m c_p (T_{c, in} - T_{c, out}) = \rho \ u \ A c_p (T_{c, in} - T_{c, out}) \) \hfill (21)

\[ HTF = 2M F_h A w / r_p \] \hfill (22)

**Table 4.** List of parameters & respective abbreviations for LHP-Ex

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Abbreviations/ Symbols</th>
</tr>
</thead>
<tbody>
<tr>
<td>Loop Heat Pipe HEx.</td>
<td></td>
</tr>
<tr>
<td>Evaporator Temperature LHP Connecting the Power Cycles (K)</td>
<td>( T_{EvLHP} )</td>
</tr>
<tr>
<td>Condenser Temperature LHP Connecting the Power Cycles (K)</td>
<td>( T_{CondLHP} )</td>
</tr>
<tr>
<td>Heat Absorbed in the Evaporator of the LHP Connecting the Power Cycles (kW)</td>
<td>( Q_{EvLHP} )</td>
</tr>
<tr>
<td>Heat Transported to the Condenser of the LHP Connecting the Power Cycles (kW)</td>
<td>( Q_{CondLHP} )</td>
</tr>
<tr>
<td>Heat Input to the Boiler of the ORC Through the LHP Connecting the Power Cycles (kW)</td>
<td>( Q_{inL,R} )</td>
</tr>
<tr>
<td>The surface tension N/m</td>
<td>( \sigma )</td>
</tr>
<tr>
<td>The latent heat of vaporization at heat pipe temperature in kJ/kg</td>
<td>( h_{fg} )</td>
</tr>
<tr>
<td>The viscosity of the liquid in Pa-s.</td>
<td>( \mu_l )</td>
</tr>
<tr>
<td>The Figure of Merit in kW/m²</td>
<td>( MF_h )</td>
</tr>
<tr>
<td>The X-sectional area of wick in m²</td>
<td>( A_w )</td>
</tr>
<tr>
<td>The Permeability of the wick material in m²</td>
<td>( k_{hp} )</td>
</tr>
<tr>
<td>The effective length of the heat pipe in m</td>
<td>( L_{eff} )</td>
</tr>
<tr>
<td>The pore radius in m</td>
<td>( r_p )</td>
</tr>
<tr>
<td>The acceleration due to gravity in m/s²</td>
<td>( g )</td>
</tr>
<tr>
<td>The inclination angle in degrees</td>
<td>( \Phi )</td>
</tr>
<tr>
<td>The density of vapor in heat pipe in kg/m³</td>
<td>( \rho_l )</td>
</tr>
<tr>
<td>Mass Flow Rate in the Evaporator &amp; Condenser of LHP in kg/s</td>
<td>( \dot{m}_{LHP} )</td>
</tr>
<tr>
<td>Heat Exchange in the Evaporator&amp; Condenser of LHP Respectively in kW</td>
<td>( Q_{EvLHP} ) &amp; ( Q_{CondLHP} )</td>
</tr>
<tr>
<td>Enthalpy inlet &amp; outlet of the Evaporator of LHP kJ/kg</td>
<td>( h_{EvL,LHP} ) &amp; ( h_{EvO,LHP} )</td>
</tr>
<tr>
<td>Enthalpy inlet &amp; outlet of the Condenser of LHP kJ/kg</td>
<td>( h_{CondLHP} ) &amp; ( h_{CondoLHP} )</td>
</tr>
<tr>
<td>Mean Temperature Diff. in the Evaporator &amp; Condenser of LHP Respectively in K</td>
<td>( T_{H-LHP} ) &amp; ( T_{L-HLP} )</td>
</tr>
<tr>
<td>The temperature of the vapor in LHP in K</td>
<td>( T_vLHP )</td>
</tr>
<tr>
<td>The characteristic dimension of the liquid/vapor interface (m)</td>
<td>( \lambda )</td>
</tr>
<tr>
<td>Thickness thermal layer (m)</td>
<td>( \delta )</td>
</tr>
<tr>
<td>Heat Transfer in LHP as per Sonic Limitation in kW</td>
<td>( Q_S )</td>
</tr>
<tr>
<td>Heat Transfer in LHP as per Entrainment Limit Limitation in kW</td>
<td>( Q_{Ent} )</td>
</tr>
<tr>
<td>Heat Transfer in LHP as per CapillaryLimit Limitation in kW</td>
<td>( Q_{Cap} )</td>
</tr>
<tr>
<td>X-sectional area of the duct of LHP (m²)</td>
<td>( A )</td>
</tr>
<tr>
<td>Permeability of the Wick Material</td>
<td>( k_{hp} )</td>
</tr>
</tbody>
</table>

Ejector Refrigeration Cycle [32]

The Jet Velocity at the exit of the Nozzle,
\[ C_{11a} = \sqrt{2 \eta_N (h_{11} - h_{12})}, \] where \( \eta_N \) is nozzle efficiency 0.85. \hfill (23)

Continuity Equation at the Mixing Section \( m_{11} + m_{15} = m_{11bc} \), \hfill (24)

Where; point \( 11bc \) shows the throat right before the shock.

Momentum Equation at the Mixing Section \( (m_{11} + m_{15}) C_{11b} = m_{11} C_{11c} - m_{15} C_{15} \) \hfill (25)

Velocity & Mass Flow at the Suction of vapor from flash chamber \( m_{15} \) & \( C_{15} \) are considered to be negligible.

\[ m_{11bc} C_{11bc} = m_{11} C_{11a} \] \hfill (26)

Energy Equation of Mixing Process

\[ m_{11} h_{11} = m_{11bc} (h_{11b} + c_{11a}^2/2) \] \hfill (27)

Continuity Equation for Shock Diffuser Section

\[ m_{11bc} = C_{11b} A/v_{11b} = C_{11c} A/v_{11c}, \] where \( v \) is the specific volume. \hfill (28)

Momentum equation at shock diffuser section:

\[ (p_{11c} - p_{11b}) A = (C_{11b} - C_{11c}) m_{11bc} \] \hfill (29)

Energy equation at shock diffuser section:

\[ h_{11b} + C_{11b}^2/2 = h_{11c} + C_{11c}^2/2 \] \hfill (30)

Energy equation at shock diffuser section:

\[ h_{12} - h_{11c} = C_{11c}^2/2 = h_{11c} - h_{11b}/\eta_D, \] where \( \eta_D = 0.8 \) is the diffuser efficiency \hfill (31)

Neglecting the shock waves,

\[ \eta_m m_{11} C_{12}^2/2 = m_{11bc} C_{11bc}^2/2, \] where \( \eta_m = 0.65 \) is the entrainment efficiency \hfill (32)

Mass Ratio, \( m_R = m_{11}/m_{15} = h_{12} - h_{11bc}/[\eta_N \eta_D \eta_E (h_{11} - h_{11bc}) - (h_{12} - h_{11bc})] \) \hfill (33)

\[ \eta_N \eta_D \eta_E \text{ Overall Efficiency of Ejector, } \eta_O \] \hfill (34)

Refrigeration Effect, \( RE = h_{18} - h_{17} \) \hfill (35)

Heat Supplied, \( Q_{\text{Sup}} = h_{11} - h_{10} \) \hfill (36)

\[ \text{COP} = RE/Q_{\text{Sup}} \] \hfill (37)

**Table 5.** List of parameters & respective abbreviations for ORC

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Abbreviations/ Symbols</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Ejector Refrigeration Cycle</strong></td>
<td></td>
</tr>
<tr>
<td>Temperature (K) &amp; Pressure (kPa) at Different States</td>
<td>( T, P )</td>
</tr>
<tr>
<td>Enthalpy in kW &amp; kJ/kg respectively</td>
<td>( H &amp; h )</td>
</tr>
<tr>
<td>The temperature of the Environment in K</td>
<td>( T_o )</td>
</tr>
<tr>
<td>Nozzle, Diffuser, Entainment &amp; Overall Efficiency</td>
<td>( \eta_N, \eta_D, \eta_E, \eta_O )</td>
</tr>
<tr>
<td>Coefficient of Performance</td>
<td>( \text{COP} )</td>
</tr>
<tr>
<td>Refrigerating Effect kW</td>
<td>( \text{RE} )</td>
</tr>
<tr>
<td>Heat Supplied in the Boiler kW</td>
<td>( Q_{\text{Sup}} )</td>
</tr>
<tr>
<td>Ejector Area Ratios</td>
<td>( A_R )</td>
</tr>
</tbody>
</table>
Mass at Different States kg/s $m$
Velocity at different State’s m/s $C$
Dryness Factor at Ejector Exit $x_{12}$
Ratio of Mass Flow in Boiler to Evaporator $m_{Ratio}$

Table 6 shows the recommended and non-recommended wick material for the LHP-Evaporator with the corresponding working fluids. In this research work Copper, Silica or Stainless Steel can be invariably used based on thermal effectiveness & cost-effectiveness.

Table 6. Heat Pipe working fluids & material compatibility [31]

<table>
<thead>
<tr>
<th>Working Fluid</th>
<th>Recommended Material</th>
<th>Not Recommended Material</th>
</tr>
</thead>
<tbody>
<tr>
<td>Acetone</td>
<td>Copper, Silica, Aluminium, Stainless Steel</td>
<td>Aluminium</td>
</tr>
<tr>
<td>Methanol</td>
<td>Copper, Silica, Stainless Steel</td>
<td>Silica, Aluminium, Stainless Steel, Nickel, Carbon Steel</td>
</tr>
<tr>
<td>Water</td>
<td>Copper, Silica, Aluminium, Stainless Steel, Nickel, Carbon Steel</td>
<td>Silica, Aluminium, Stainless Steel, Nickel, Carbon Steel, Inconel</td>
</tr>
<tr>
<td>Ethanol</td>
<td>NA</td>
<td>NA</td>
</tr>
</tbody>
</table>

Similar to the table above, Table 7 presents the suitable operating temperature range, based on which the working fluid can be decided for the LHP. In this research work, the operating range is about 400-450 K. Hence, the corresponding working fluids i.e., Ethanol, Acetone, Methanol & Water have been chosen. It can be observed that the LHP can be used to extract heat from relatively low temperatures as well as to run the bottoming cycle such as chosen in this investigation.

Table 7. Heat Pipe working fluids useful operating temperatures [31]

<table>
<thead>
<tr>
<th>Working Fluid</th>
<th>Melting Point (ºC)</th>
<th>Boiling Point at ATP (ºC)</th>
<th>Useful Range (ºC)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Acetone</td>
<td>-95</td>
<td>57</td>
<td>0-120</td>
</tr>
<tr>
<td>Methanol</td>
<td>-98</td>
<td>64</td>
<td>10-130</td>
</tr>
<tr>
<td>Ethanol</td>
<td>-112</td>
<td>78</td>
<td>0-130</td>
</tr>
<tr>
<td>Water</td>
<td>0</td>
<td>100</td>
<td>30-200</td>
</tr>
</tbody>
</table>

The next section discusses the results obtained from the simulations through various plots and data obtained.

RESULTS AND DISCUSSIONS

The results obtained from the have been presented in this section. As input parameters the Boiler Pressure ($P_{11}$) of ERS has been varied from 500 kPa to 800 kPa, Peak Temperature of the GPC ($T_5$) from 600 K to 1000 K, 6 Eco-friendly Working Fluids for ERS, and 4 Working Fluids of the LHP Ex. have been used for the investigation. This section has been separated into 3 sub-sections which present the results based on $T_5$ (subsection-3.1), $P_{11}$ (subsection-3.2) & Working Fluids in the LHP (subsection-3.3). These results have been presented for the different working fluids being used in the ERS. In sub-section 3.3, results based on 3 input variables namely ERC working fluid-LHP working fluid-$P_{11}$ and ERC working fluid-LHP working fluid-$T_5$ respectively have been presented. The results have been emphasized for R718 for being the widely available and origin of Steam Jet Refrigeration System as well as all other ancient refrigeration systems and for R1224yd (Z) as it has 0 ODP & 0 GWP and is the most eco-friendly refrigerant of all the chosen refrigerants. By the end of this section, we will be able to assess the feasibility and best operating parameters for such a combined system. The dotted – discontinuous lines/curves may be neglected for they are mere projections.
The Peak Temperature Of Gpc ($T_s$)

As the $T_s$ has been varied from 600K to 1000K, the effect of temperature available for heat transfer has been studied. In fig 5 it can be observed that the Temperature available in the boiler of ERC varies from 380 K to 455 K. It can be reiterated that the critical temperatures of the Eco-friendly refrigerants chosen for the study are in the range of 380K – 450K as well. Hence, the boiler can be operated at high temperatures near critical points low heat consumption for the cycle to work.

![Graph showing temperature profiles of components of combined cycle with $T_s$.]

**Figure 5.** Temperature Profiles of the Components of Combined Cycle with $T_s$

In Fig 6 refrigeration capacity obtained for the temperature range has been presented. It can be seen that owing to the critical temperature limits, all the ERC working fluids except R718 have been studied for the range of 600K to 800K and 800K -1000K for R718. It can be seen R365mfc (i.e., 8.601kW to 10.35kW) &R1233zd (E) (i.e., 7.657kW to 11.05kW) provide consistent high values of the refrigeration capacities. R245fa &R1224yd (Z) (7.87kW-9.903kW) have coinciding values. With the temperature rise, it can be observed that the Refrigeration capacity also increases. For R718 the rise is the steepest from 6.835kW to 31.545kW.

![Graph showing refrigeration capacity of ERS with $T_s$.]

**Figure 6.** Refrigeration Capacity of the ERS with $T_s$
The COP can be seen to be decreasing with the increase in temperature Fig 7. For R365mfc the value of COP has been noted to be the maximum from 0.2856 to 0.249. Whereas, for R718 the COP is more consistent and is in the range of 0.285 to 0.282. The main drawback of R718 being not able to run below 278 K. For R1224yd (Z) the value of COP is close and parallel with R245fa in the range of 0.241 to 0.225.

![Figure 7. COP of the ERS with T₅](image)

The Boiler Pressure of ERS (P₁₁)

Refrigeration capacity and COP have also been presented with the varying Boiler Pressure in Fig 8 & 9 respectively. Unlike the previous sub-section, all the refrigerants have been studied for the same range of pressure 500kPa to 800kPa. A continuous and gradual increase can be seen in the refrigeration capacity with the increase in pressure.

![Figure 8. Refrigeration Capacity of the ERS with P₁₁](image)
It can be seen in Figure 8 that for R718 the refrigeration capacity has been recorded as a maximum of 16.62 to 17.65 kW, whereas, for the refrigerants, R365mfc provides a cooling effect of 10.75 to 12.93 kW under the subject conditions. It must be mentioned that the R1224yd (Z) which has 0 GWP & 0 ODP in a system has recorded RC of 9.86 to 11.41 kW.

**Figure 9. COP of the ERS with P_{11}**

Fig 9 presents the data recorded for COP under the variable Boiler pressure. The COP can be seen gradually increasing with an increase in boiler pressure. R718, R365mfc & R1224yd (Z) have maximum COP of 0.289, 0.275 & 0.241 respectively.

**Working Fluids In The Lhp**

Four easily available and eco-friendly working fluids have been selected for the LHP Ex. As mentioned in Section 2, the working fluids have defined working temperatures as well as suitable materials based on the corresponding thermo-physical properties. This section has focussed on the mass flow rate of the working fluids, availability of Heat at the inlet of LHP & ERC for all the working fluids subject to variable Peak Temperature of GPC, Peak Pressure of ERC, and different working fluids. Smaller the mass flow handling, Compact the whole combined system. It can be seen that mass flow rate and Heat Input for LHP for all the refrigerants except R718 and R718 it’s different. Moreover, as mentioned above the primary focus of this section is on R718 & R1224yd (Z), which has the potential of providing the most compatible eco-friendly system subject to the variable Temperature & Pressure

**Acetone**

In Figure 10 to Figure 13 the results based on Acetone as LHP working fluid have been presented based on variable Pressure & Temperature. The mass flow rate of refrigerants in the ERC is maximum for R1224yd (Z) (1.041kg/s) and least for R718 (0.097kg/s). The mass flow rate in the LHP for the R718 system is 0.639kg/s, whereas, for other refrigerants, it was recorded as 0.4635kg/s.

In Figure 11 heat inputs can be seen to be constant with the varying Pressure. Input to the LHP can be seen to be carried away for transfer with R718 system is at 197.1 kW & for other refrigerants 155.9 kW. Whereas, the heat input to the ERC for R718 is 160.1kW & for other eco-friendly refrigerants is 142kW.
The results for the analysis of Acetone based on Peak Temperature of GPC Fig 12 & 13. From Fig 12 the maximum range of mass flow rate in ERC has been recorded as 0.821-1.283 kg/s for R236ea and for R1224yz (D) it has been recorded as 0.85-1.245 kg/s. For R718 the range of mass flow rate in ERC is 0.096-0.17 kg/s.

The maximum mass flow rate range inside the LHP is recorded for R365mfc at 0.342-0.439 kg/s and for R1224yz (D) at 0.3401-0.498 kg/s. It can also be seen that for R 718 the LHP range of mass flow rate is 0.6396 - 1.321 kg/s.

The heat input range at ERC for R718 can be seen from Fig 13 as 160.1 - 182.6 kW & for LHP it has been recorded as 197.1 - 353.1 kW. Whereas, for other refrigerants, the range has been recorded as 123.8 - 154.3 kW & 127.2 - 181.5 kW respectively for ERC & LHP.

**Figure 12.** Mass Flow Rate of ERS & LHP with $T_3$ for Acetone

**Figure 13.** Heat Available as Input to ERS & LHP with $T_3$ for Acetone

**ETHANOL**

In Figures 14 & 15 the results of the analysis of Ethanol with variable pressure have been presented. It can be observed from Fig 14 that the maximum mass flow in ERC is for R1224yz(D) 1.01 kg/s and the minimum is for R718 at 0.096 kg/s. Maximum mass flow in LHP for refrigerants is 0.244 kg/s and for R718 is at 0.342 kg/s.

**Figure 14.** Mass Flow Rate of ERS & LHP with $P_{11}$ for Ethanol

From Figure 15 it can be observed that heat input available at ERC & LHP for refrigerants are 142 kW & 155.9 kW. For R718 heat input available at ERC & LHP are 160.1 kW & 194.4 kW.

**Figure 15: Heat Available as Input to ERS & LHP with $P_{11}$ for Ethanol**

Similarly, the results of analysis based on the Peak Temperature have been reported in Figure 16 & 17. The range of Maximum mass flow rate in ERC from Fig 16 is for R1224yz(D) at 0.8321 - 1.178 kg/s. The range of mass flow rate for LHP for R1224yz (D) is 0.1893 - 0.3261 kg/s. The range of mass flow rate in ERC & LHP for R718 can be observed as 0.09514 - 0.1877 kg/s & 0.3419 - 0.7967 kg/s respectively.

Furthermore, from Figure 17 it has been studies that the Heat input range available for ERC & LHP for refrigerants is 123.8 - 154.3 kW & 124.5 - 173.7 kW. Moreover, for R718 it has been observed for ERC & LHP is 160.1 - 185.4 kW & 194.4 - 388.9 kW.
METHANOL

For methanol as well Maximum Mass flow for variable boiler pressure can be observed for R1224yz(D) as 1.023 Kg/s from Figure 18. For R718 the maximum mass flow rate is 0.09247 kg/s. The maximum mass flow rate for all the refrigerants has been recorded as 0.1968 kg/s and 0.2618 kg/s for R718.

From Figure 19 it has been studied that the Heat Input available for refrigerants at ERC & LHP is 142 kW & 153.2 kW respectively. Similarly, for R718 Heat Input available is 160.1 kW & 186.7 kW.
For the study based on the Peak GPC Temperature Fig 20 & 21 have been presented. It has been observed from Fig 20 that the range for maximum mass flow rate in ERC is for R236ea as 0.8109 - 1.239 kg/s. For R1224yz (D) the range has been observed to be 0.8389 - 1.186 kg/s. Moreover, mass flow rate range for LHP for R236ea & R1224yz (D) is 0.1512 - 0.2375 kg/s & 0.1512 - 0.2618 kg/s respectively. For R718 mass flow rate range for ERC & LHP are 0.09139 - 0.1621 kg/s & 0.2618 - 0.5451 kg/s respectively.

The ranges of Input Heat available for ERC & LHP for Refrigerants are 123.8 - 154.3 kW & 125.5 - 175.3 kW respectively. Moreover, for the R718 range of Heat Input Available for ERC & LHP are 160.1 - 185.4 kW & 186.7 – 336 kW respectively.

**Figure 20.** Mass Flow Rate of ERS & LHP with T₅ for Methanol

**Figure 21.** Heat Available as Input to ERS & LHP with T₅ for Methanol

**WATER**

In Figure 22 & 23 results for water in LHP have been presented for variable Boiler Pressure. In Fig 22, the Maximum mass flow rate in ERC has been required for R1224yz (D) as 0.8651 kg/s whereas, for R718 the value of the same rounds up to 0.07329 kg/s. Furthermore, the mass flow rate for the LHP for R1224yz (D) & R718 is 0.07071 kg/s & 0.08299 kg/s.

Heat Input available at ERC & LHP for refrigerants is 129.5 kW & 142 kW, while, the heat available at ERC % LHP for R718 is 148 kW & 160.1 kW.
Figure 22. Mass Flow Rate of ERS & LHP with $P_{11}$ for Water

Figure 23. Heat Available as Input to ERS & LHP with $P_{11}$ for Water

Figure 24 & 25 present the study with variable temperature. In the Fig 24, the Range of mass flow rate in ERC & LHP for R1224yz(D) is $0.7475 - 0.9348$ kg/s & $0.05957 - 0.08301$ kg/s respectively, whereas, for R718 the range of mass flow rate $0.07244 - 0.0906$ kg/s & $0.08299 - 0.1128$ kg/s respectively.

From Figure 23, the heat input range available at the ERC & LHP for the refrigerants is $111.9 - 142$ kW & $123.8 - 154.3$ kW, whereas, for R718 it is $148 - 188.7$ kW & $160.1 - 196.7$ kW respectively.

CONCLUSIONS

The results obtained based on various Input parameters have been discussed, especially for R718 & R1224yz (D) (i.e., 0 ODP & 0 GDP). It has also been observed that the Temperature that can be made available for heat transfer with varying the Peak Temperature of GPC can be near or beyond the critical temperature of the eco-friendly refrigerants hence reducing the overall requirement of heat input. Hence, making the combined cycle highly attractive for industrial & eco-friendly usage. Moreover, the followings are the quantitative conclusions to the research investigation.

i. The temperature available at the ERC Boiler Input is in the range of 380 K to 455K which is very close to and above the Critical Temperature of the New Eco-Friendly Refrigerants.

ii. The Maximum Refrigeration capacity for variable Peak GPC Temperature conditions for R365mfc, R1224yd (Z) & R718 has been recorded as 10.35kW, 9.903kW & 31.545kW respectively, increasing with the temperature.

iii. Furthermore, the Maximum COP for variable Peak GPC Temperature Condition for R365mfc, R1224yd (Z) & R718 has been observed to be as 0.2856, 0.241 & 0.285 respectively, showing a downwards trend with increasing temperature.

iv. Whereas, for Peak Boiler Pressure Maximum Refrigeration Effect has been calculated for R365mfc, R1224yd (Z) & R718 at 12.93, 11.41 & 17.65 increasing with the pressure.

v. The COP under the variable pressure condition for R365mfc, R1224yd (Z) & R718 is 0.275, 0.241 & 0.289.

vi. For Water as working fluids of the LHP, the least Mass Flow Rate in ERC & LHP for R1224yd (Z) & R718 has been 0.8651 kg/s & 0.07329 kg/s and 0.07071 kg/s & 0.08299 kg/s respectively.

vii. Moreover, for Acetone as working fluids of the LHP, Heat Input Available has been maximum at ERC & LHP for R1224yd (Z) & R718 is 142 kW & 160.1kW and 155.9 kW & 197.1 kW respectively, for variable pressure conditions.

viii. For Variable Peak GPC temperature condition for Water as working fluids of the LHP, the least Mass Flow Rate in ERC & LHP for R1224yd (Z) & R718 has been 0.747 kg/s & 0.072 kg/s and 0.0595 kg/s & 0.08299 kg/s respectively.

ix. Moreover, for Acetone as working fluids of the LHP, Heat Input Available has been maximum at ERC & LHP for R1224yd (Z) & R718 is 154.3 kW & 182.6kW and 181.5 kW & 363.1 kW respectively, for variable pressure conditions.

It can be observed that for the combination of 1224yd (Z) (ERC)-Water (LHP) as compact, eco-friendly & industrially viable combined system can be presented.

RECOMMENDATIONS FOR FUTURE RESEARCH WORKS

It can be recommended that further experimental work be initiated on this novel system to obtain a low-cost, eco-friendly Industrial refrigeration system that can be used for the storage of different sensitive items as well as for comfort usages. Studies on customization of the LHP must also be taken up for heavy industrial applications.

REFERENCES


[34] http://www.ertc.od.ua/en/about_ert_en.html