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Study of Performance Enhancement of Half Effect Vapor Absorption System Using Loop Heat Pipes

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Abstract: The Half Effect Vapor Absorption Refrigeration System (VARS) has performance lower than the Single Effect VARS due to several reasons. This research aims to enhance the performance of the half effect system by incorporation of a Loop Heat Pipe (LHP) between the high absorber, high generator and condenser (which is actually replaced by the LHP) to avail the intra-cycle heat exchange. The simulations show that COP I and COP II increase by 64 % and 27 % respectively. Also the LHP condenser temperature $T_{Cond}$is dependent on the generator temperature T_G. At higher temperatures of TG the increase in COP $_{II}$ is more than that in COP $_I$. Average heat leak from the LHP $Q_{Leak}$is around 14.38 kW and average heat utilized due to the LHP $Q_{Cond}$is found to be 79.52 kW.

I. INTRODUCTION

A half-effect vapour absorption system (Fig 3) consists of 2 generators, 2 absorbers, a condenser, an evaporator, 2 pumps, 2 heat exchangers and 3 throttling valves. The half-effect cycle is basically a combination of two single-effect absorption cycles each working at different pressure levels. This system has been developed for relatively low-temperature heat source application. Also the COP of the half-effect system is relatively lower because it rejects more heat than a single-effect cycle. Heat from high temperature external source, transfers to the generator and the absorbers reject heat to the surroundings.

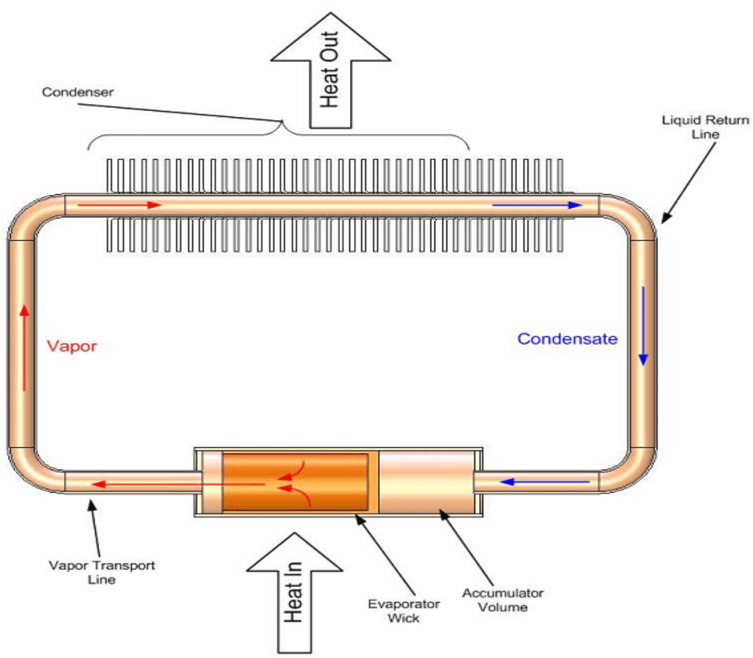

Fig 1: Cyclic process of a Loop heat pipe [36]

Loop heat pipes (LHPs) are two-phase devices used to transfer heat and maintain temperature. It was developed 1980s for spacecrafts uses. Heat in evaporator vaporizes the working fluid at the wick outer surface, then the vapor flows down the system of grooves and through the vapor line towards the condenser. The fluid condenses in the condenser part. A two-phase chamber (compensation chamber) is present at the end of the evaporator which works at a slightly lower temperature (than the evaporator and the condenser). The lower value saturation pressure in the reservoir draws the condensates and liquid return line. Then the fluid flows into a central pipe where it is fed to the wick.

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LHPs are self-primed as the volumes of the reservoir, condenser and vapor and liquid lines are controlled so that fluid is always existing to the wick. The reservoir volume and fluid are set to always have fluid in the reservoir even if the condenser and vapor and liquid lines are full of fluid.

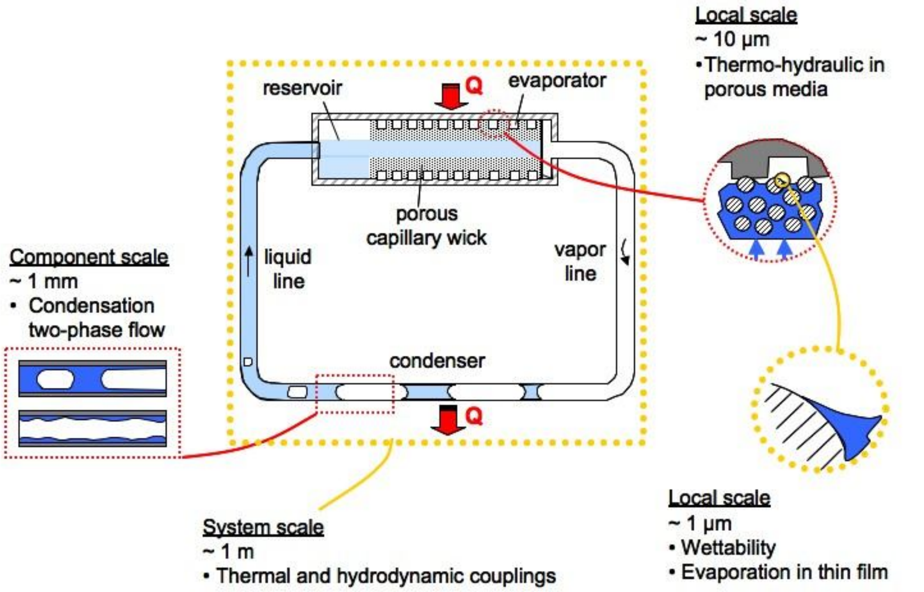

Fig 2: Porous Wick in the LHP.[35]

The LHPs have an inverted wick, and the vapors are located adjacent to the heated surface. These wicks are prepared by power metallurgy. The outer surface of the wick is in contact with the heated surface. Circumferential and axial grooves are required to generate flow channels for the vapors flow which can be machined in the wick, or to the evaporator body.

II. LITERATURE REVIEW

Fabian Korn et al. [2012]performed several vital experiments on heat pipes to establish it to be one of the most effective procedures to transport thermal energy from one point to another, mostly used for cooling[6].Sameer Khandekaret al.[2010]performed experiments on the global thermal performance modeling of Pulsating Heat Pipes (PHPs) requires local, spati-o-temporally coupled, flow and heat transfer information during the characteristic, self-sustained thermally driven oscillating Taylor bubble flow, under different operating conditions[7].JozefHužvár, Patrik Nemecet al. [2007]used heat pipe, observed its basic principles and operating limits. High temperature heat pipes were evaluated for use in energy conversion applications such as fuel cells, gas turbine re-combustors, and Stirling cycle heat sources, with the resurgence of space nuclear power, additional applications include reactor heat removal elements and radiator elements[8].R.Z. Wanget al.[2008]added heat pipes in adsorption water chiller or ice maker initials. His work showed that the adsorption refrigerators are very efficient [10].PrachaYeunyongkul et al.[2009]aimedat experimentally investigating the application of a closed loop oscillating heat pipe (CLOHP) as the condenser for a vapor compression refrigeration system[14].R. Rajashree et al.[1990]went through a numerical analysis of an unsteady, viscous, laminar, incompressible, two dimensional heat and mass transfer, in the vapour gas region of gas loaded circular heat pipe [20]. Da-Wen Sun (1996) performed a detailed thermodynamic analysis of the properties of these binary fluids and expressed in polynomial equations. The performances of three cycles were compared. M.M. Talbi et al. (2000) carried out an exergy analysis on a single-effect absorption refrigeration cycle with lithium-bromide±water as the working Fuid pair. E. Kurem et al.(2001) analyzed the Absorption Heat Pump (AHP) and Absorption Heat Transformers (AHT) using ammonia-water and water-lithium bromide solutions. A fundamental AHP and AHT systems was described and explained the operating sequence. R.D. Misra et al. (2002) applied the therm-o-economic theory is to the economic optimization of a single effect water/LiBrvapour absorption refrigeration system for air-conditioning application. S.A. Adewusi et al (2004). studied the performance of single-stage and two-stage ammonia–water absorption refrigeration systems (ARSs).They calculated entropy generation of each component and the total entropy generation of all the system components as well as COP of the ARSs . S. Arivazhagan et al. (2006) investigated experimentally on the performance of a two-stage half effect vapour absorption cooling system .The prototype is designed for 1 kW cooling capacity using HFC based working fluids (R134a as refrigerant and DMAC as absorbent). RabahGomri et al. (2008) performed exergy analysis of double effect lithium bromide/water absorption refrigeration system. The system consisted of a second effect generator between the generator and condenser of the single effect absorption refrigeration system, including two solution heat exchangers between the absorber and the two generators. S.C. Kaushik et al. (2009) presented the energy and exergy analysis of single effect and series flow double effect water–lithium bromide absorption systems. They developed a computational model for the parametric investigation of the systems. Berhane H. Gebreslassie et al. (2010) performed an exergy analysis, which only considered the unavoidable exergy destruction, conducted for single, double, triple and half effect Water–Lithium bromide absorption cycles. Gulshan Sachdeva et al.(2014) performed an exergy analysis of VAR system using LiBr-H2O as working fluid with the modified Gouy-Stodola approach. Karl Ochsner (2008)et al. (2008) developed a new CO2-heat pipe with high-grade steel corrugated pipe system, which – contrary to other pipe systems permits raw length up to 100 m. They also described the establishment of the heat pump system in general.There are different sorts of the VARS namely half, single, double, triple effects etc. Thorough study of the research papers it can be observed that rigorous thermodynamic and thermo-economic studies have already been performed on almost all of the possible systems. The ground work for further improvements in the systems has also been created. Some other components can be brought in the system and some further enhancements in the performance can be brought in.In the research work LHP can be used as the new component for intra-cycle heat exchange and reduce the requirement of heat input for the cycle to operate. Also replacements of other bulky components can be done by the LHP, which would reduce the cost and size of the system

III. SYSTEMS DESCRIPTION

Figure 3 shows a schematic of Half Effect VARS which is working between 100ºC and 5°C. As the simulation shows the half effect cycle has lower COP $_I$& COP $_{II}$ than the single effect cycle, one can apply some modifications to improve the performance of the half effect cycle. The refrigeration capacity of the cycle is set to be 90 kW and the working fluid can be NH_3-H_2O solution. The system consists of 2 generators, absorbers and heat exchangers.

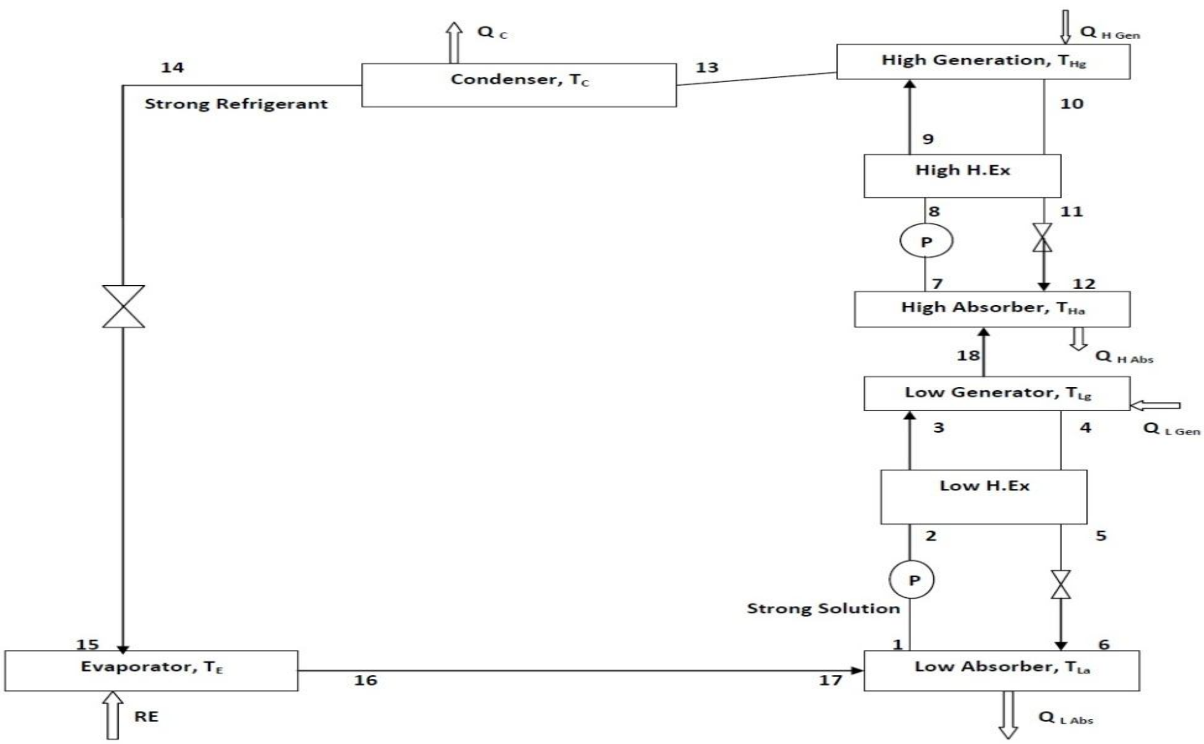

Fig 3: A Half Effect Vapour Absorption System

Figure 4 on the other hand shows the modifications proposed in the above system to have improved performance. There can be a LHP installed between the High Absorber and High Generator removing the heat exchanger and the condenser, thus reducing the size and cost of the system. The LHP acts as the condenser and the heat exchanger altogether. The benefit of high temperature heat transfer will result in decrease in energy and increase in COP $_{II}$. Also it reduces the requirements in the high generator. There can also be a LHP used between the High absorber and Low Generator similar to the arrangement that will reduce the heat requirements in low generator. The table 1 has the list of the terms used in the research work.

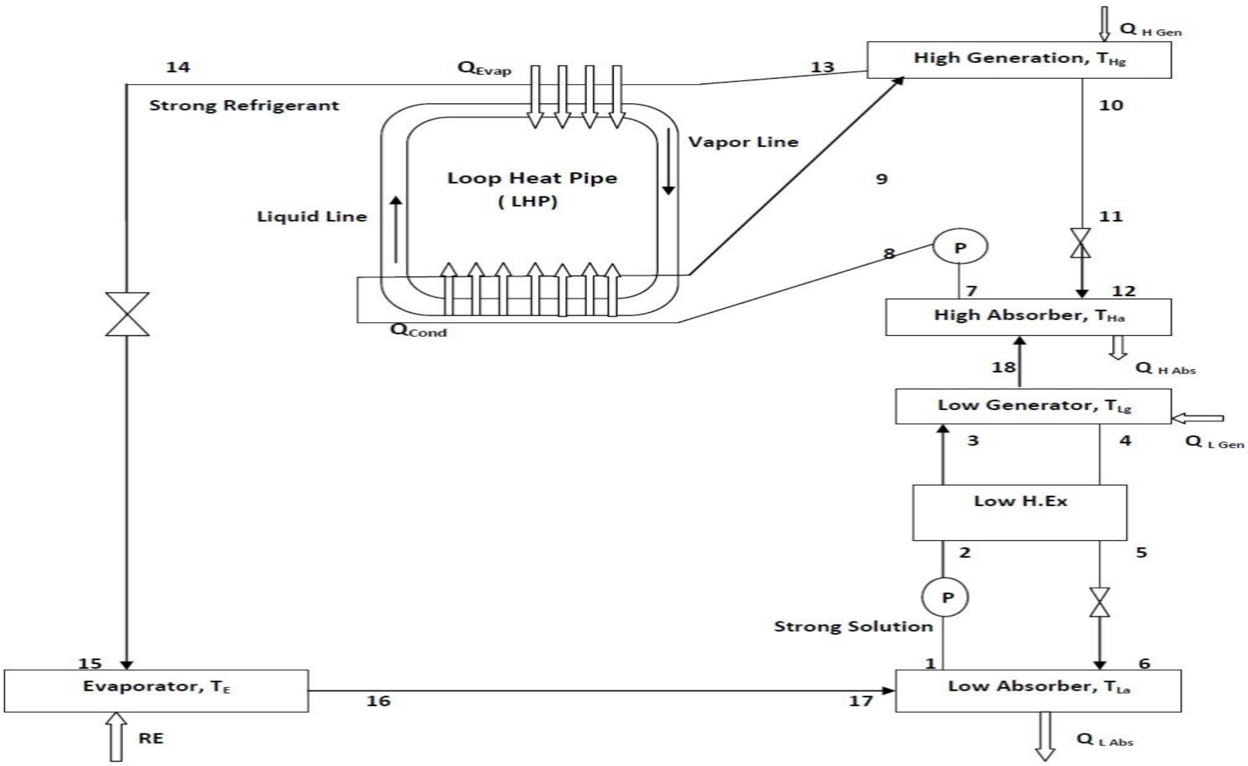

Fig.4: Modified Half Effect Vapour Absorption System

Table 1: Terms Used in Simulation

Terms	Abbreviations
Refrigeration Effect in kW	RE (kW)
Heat rejected in absorber in kW	Q_a(kW)
Heat supplied in generator in kW	Q_g (kW)
Heat rejected in condenser of LHP in kW	Q_{cond}(kW)
Heat absorbed in evaporator of LHP in kW	Q_{eva}(kW)
Absorber Temperature in°C	T_{La},T_{Ha}(°C)
Generator Temperature in °C	T_{Hg}, T_{LG} (°C)
LHP Condenser Temperature in °C	T_c (°C)
Evaporator Temperature in °C	T_E,T_e(°C)
Heat Rejected in Condenser in kW	Q_C (kW)
First Law Coefficient of Performance	COP_{I}
Second Law Coefficient of Performance	COP_{II}
Heat Leaked from the LHP in kW	Q_{Leak} (kW)
Percentage Improvement in First Law Coefficient of Performance	$\%COP_{I\ imp}$
Percentage Improvement in Second Law Coefficient of Performance	$\%COP_{II\ imp}$
Improvement in First Law Coefficient of	$COP_{I\ imp}$

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Performance	
Improvement in Second Law Coefficient of Performance	$COP_{II\ imp}$

IV. RESULTS AND DICSUSSIONS

The simulations show the following results. Fig 5 & Fig 6 show the improvements in COP_I and COP_{II} with varying heat utilized in the condenser of LHP (Q_{Cond}) with reference to the performance of the system without modifications. It can be inferred from the Fig 5 that the improvements in COP_I is directly proportional to the heat being utilized. The average improved COP_I ,that the simulations show, is around 0.69.

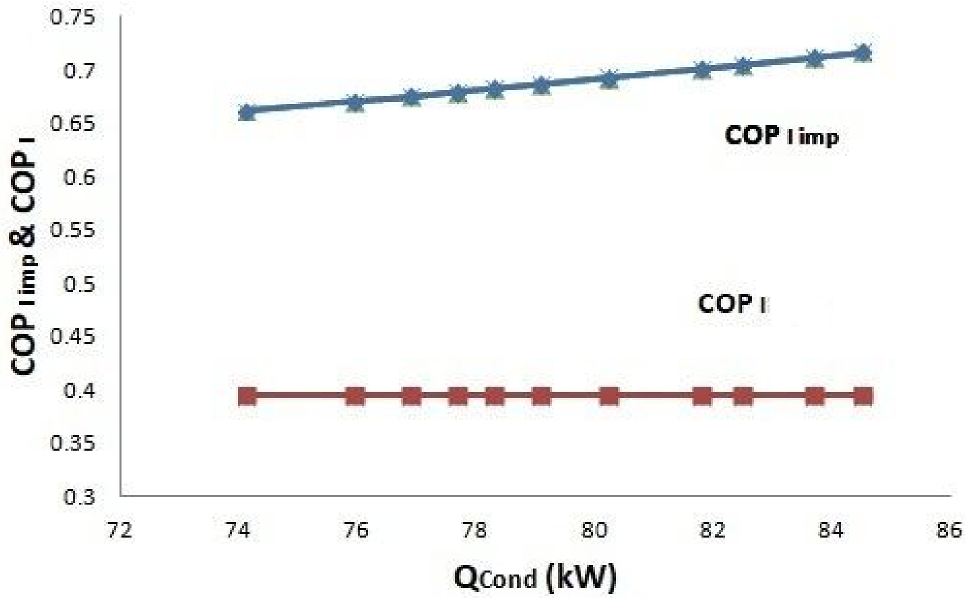

Fig 5: Comparison between COP_I and $COP_{I\ imp}$ plotted with Q_{Cond}

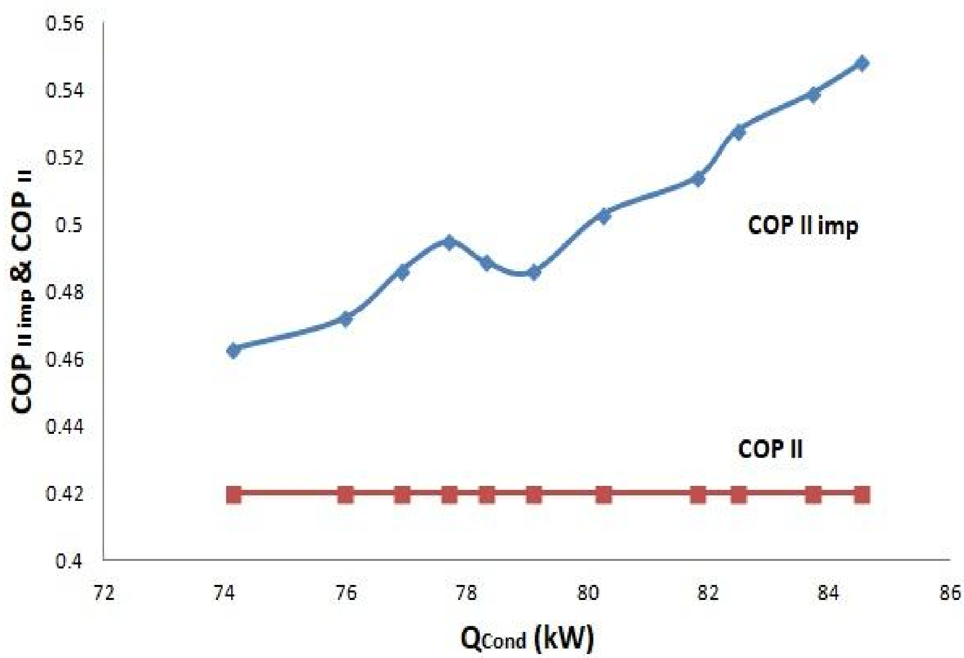

Fig 6: Comparison between COP_{II} and $COP_{II\ imp}$ plotted with Q_{Cond}

The Fig 6 shows that with reuse of heat in the LHP decreases the loss of exergy increasing the COP $_{II}$. The COP $_{II}$ decreases a little but then it rises steadily similar to the COP $_{I}$. Also the heat transfer at high temperatures helps in decreasing the exergy loss. The average enhanced COP $_{II}$ has been calculated to be 0.502 .

Moreover the Fig 7 and Fig 8 show the relative comparison between the COP $_{I}$&COP $_{II}$ .Fig 7 shows the comparison between the COP $_{I}$ and COP $_{II}$ with the varying Q_{Cond}. Both rise steadily for the entire range of the Q_{Cond}.

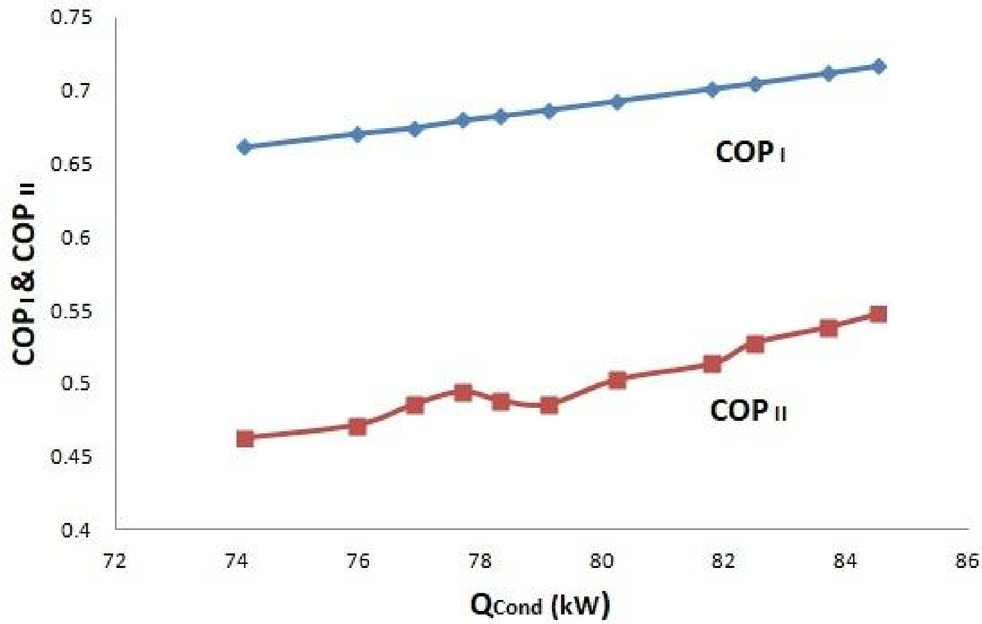

Fig 7: COP $_{I}$ and COP $_{II}$plotted with Q_{Cond}

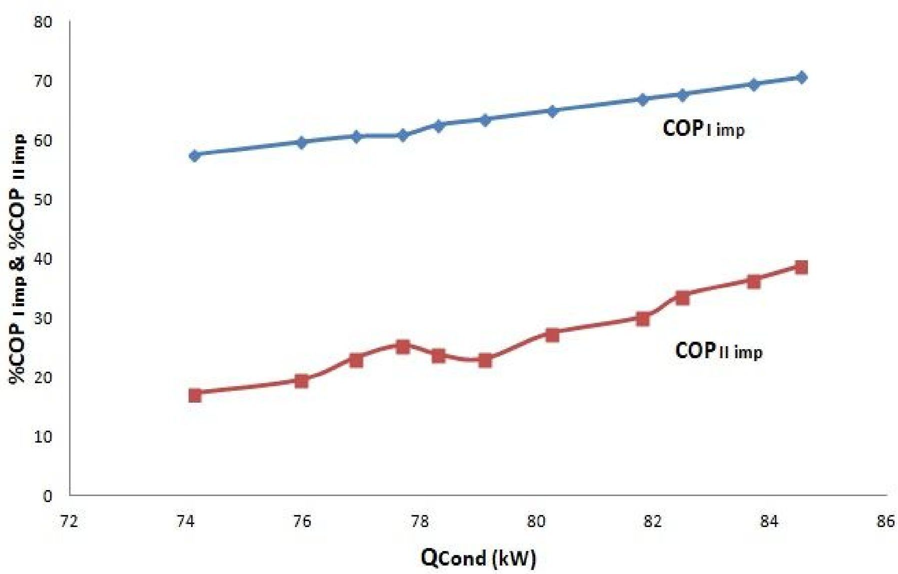

Fig8: Variation of %COP $_{I\ imp}$ and % COP $_{II\ imp}$ plotted with Q_{Cond}

The fig 8 shows the percentage improvement in the COP $_{I}$ and COP $_{II}$ with Q_{Cond}.The average rise in the COP $_{I}$ is 64 %, where as that in COP $_{II}$ is 27 %. In both the figures, COP $_{I}$ and COP $_{II}$ the rise is parallel.

The Fig 9,10,11,12 show the variation of COP $_{I}$ and COP $_{II}$ with T_C and T_G . It can be seen that the relation between the two temperatures is linear. Higher the T_G, higher will be the T_C. With high temperatures of heat exchange, the COP$_{II}$rise is more than the rise in COP $_{I}$. Whereas the rise in COP $_{I}$ becomes flat.

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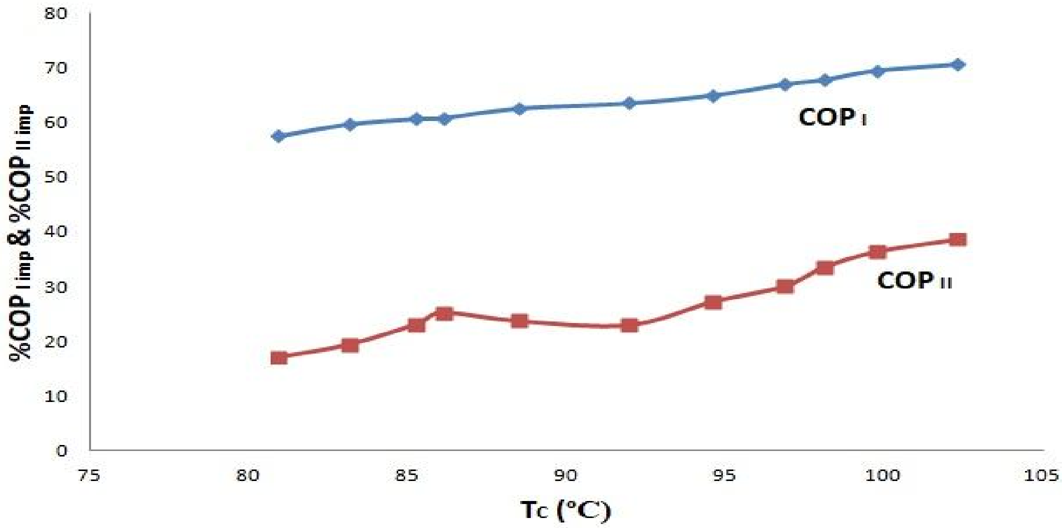

Fig9: Comparison of %age improvements in COP $_I$& COP $_{II}$ varying the T_C

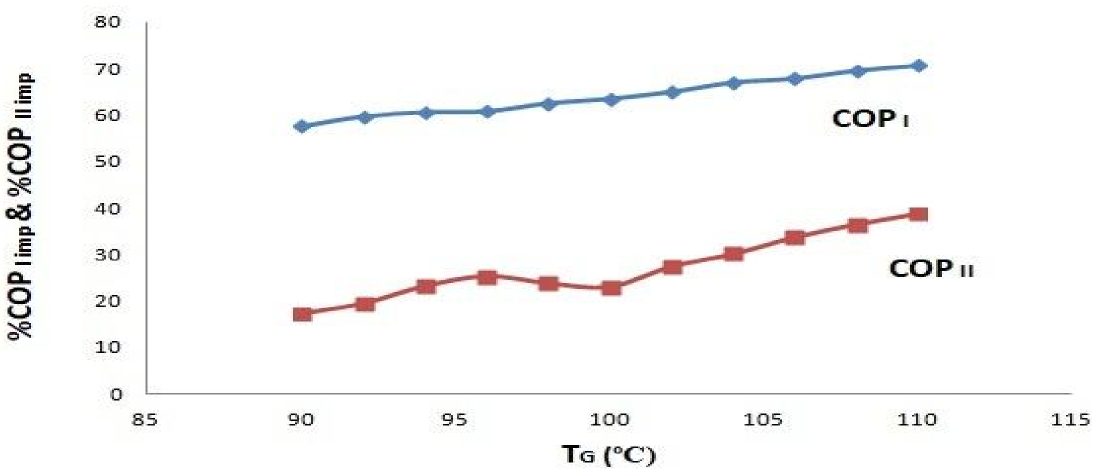

Fig 10: Comparison of %age improvements in COP $_I$& COP $_{II}$ varying the T_G

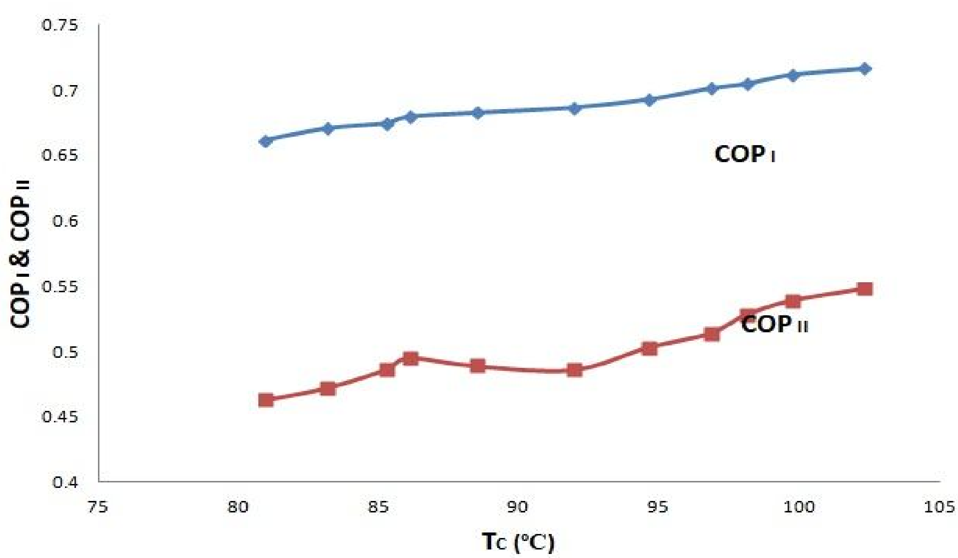

Fig 11: Comparison of COP $_I$& COP $_{II}$ varying the T_C

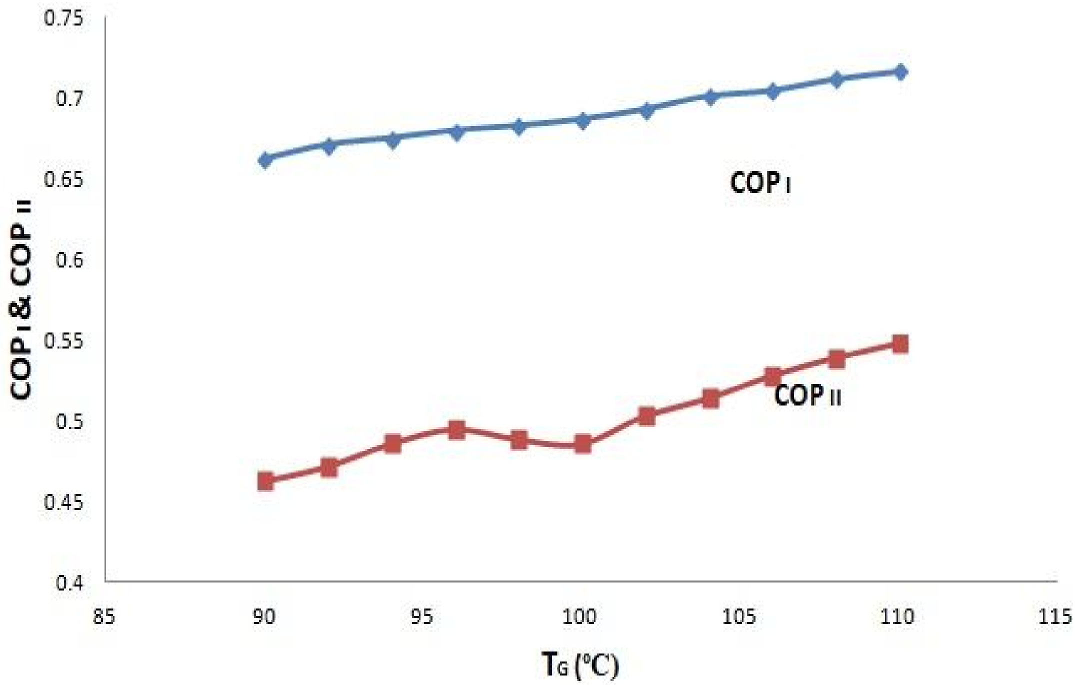

Fig 12: Comparison of COP $_I$& COP $_{II}$ varying the T_C

The Fig 13 & Fig 14 show the values of Q_{Cond}&$Q_{Leak}$with respect T_C and T_G. With increase in the temperatures the heat leaked from the LHP is found to reduce, whereas the heat reusable increases.

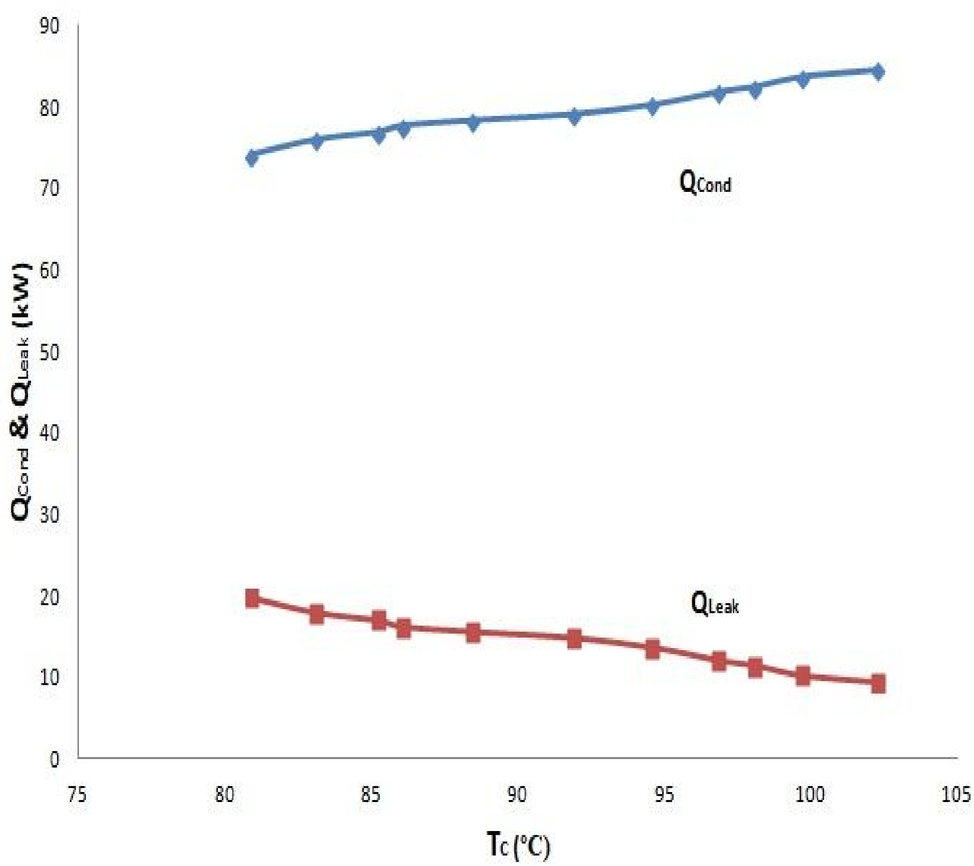

Fig 13: Comparison of Q_{Leak}&Q_{Cond}varying the T_C

The T_C is a dependent parameter on T_G directly. As the T_G increases the Tc also increases the heat to be supplied and the maximum temperature up to which it can be transferred also increases. The average $Q_{Cond}$is observed to be 79.52 kW.

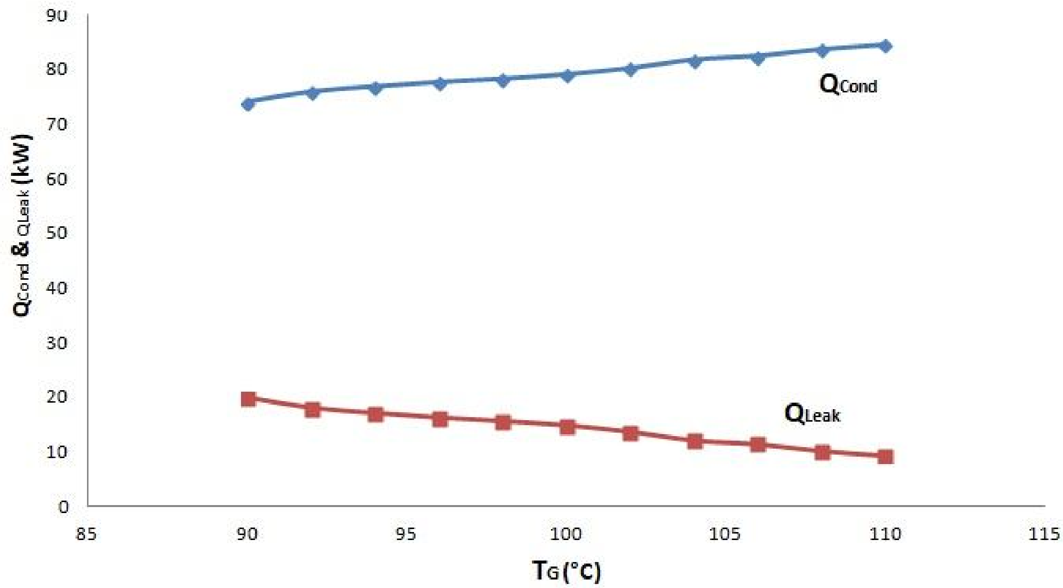

Fig 14: Comparison of Q_{Leak}&Q_{Cond}varying the T_G

$Q_{Leak}$is found to decrease as the operating temperature increases. The operating temperature increases the heat transfer coefficient hence more heat is transferred in the evaporator of the LHP, reducing the Q_{Leak}. The average Q_{Leak} is calculated to be 14.38 kW.

V. CONCLUSIONS

Through the simulations followings can be the conclusions for the Half Effect VARS:

A. COP $_I$& COP $_{II}$ increase with the Q_{Cond} and the average values for the modified value are 0.69 & 0.502 respectively.
B. The average rise in the COP $_I$ is 64 %, where as that in COP $_{II}$ is 27 %.
C. Higher the T_G, higher will be the T_C. With high temperatures of heat exchange, the COP $_{II}$ rise is more than the rise in COP $_I$. Whereas the rise in COP $_I$ becomes flat.
D. With increase in the temperatures the heat leaked from the LHP is found to reduce. Average $Q_{Leak}$is around 14.38 kW.
E. The operating temperature increases the heat transfer coefficient hence more heat is transferred in the evaporator of the LHP having the average Q_{Cond}as 79.52 kW.
F. The exergetic losses can be reduced, along with the reduction in size of the system, making the system flexible.

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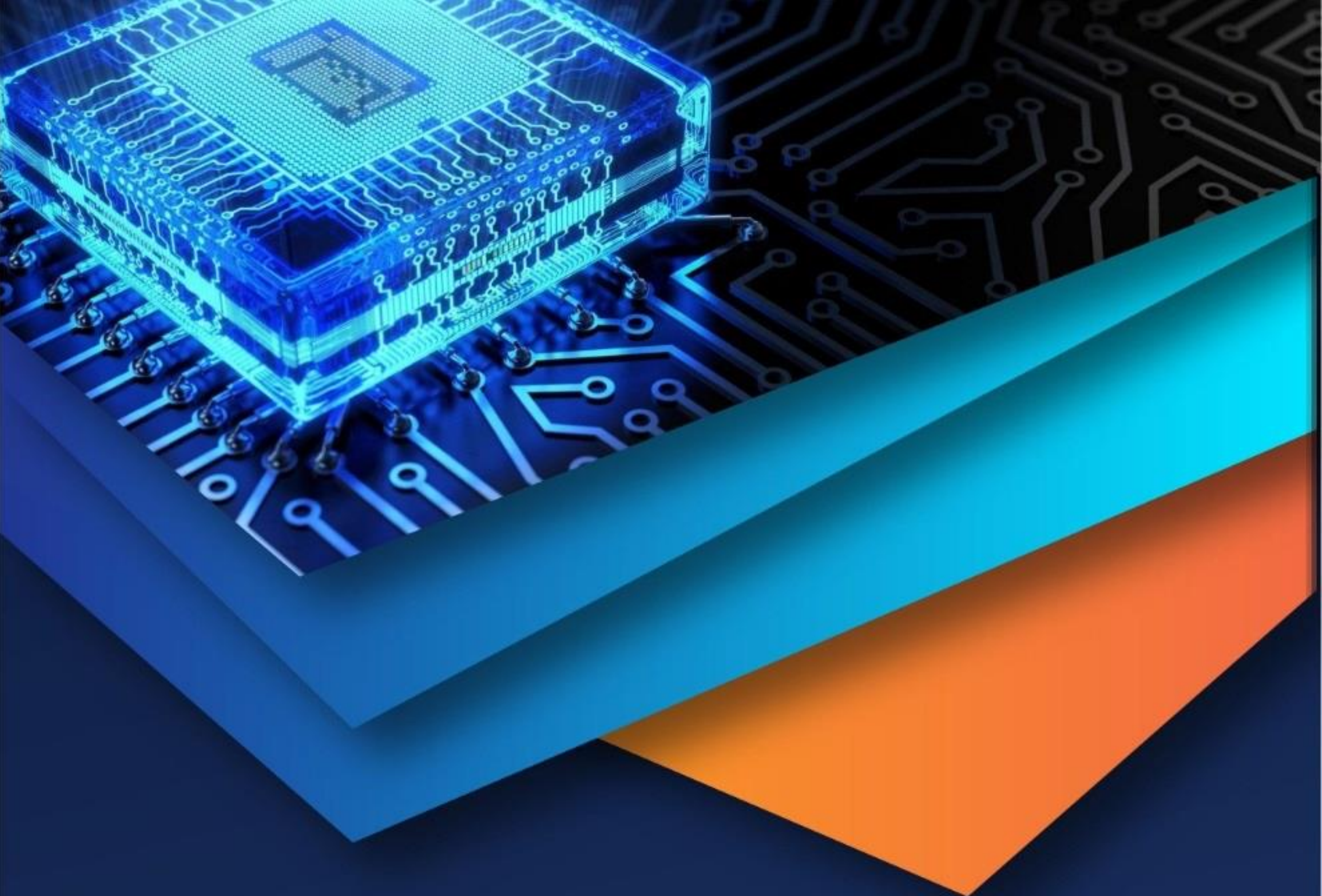

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