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Exergy Performance of the Environmentally Friendly Refrigerant Mixture: R365mfc and R152a

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Abstract- This work presents the exergy performance of new environmentally friendly refrigerants 1,1-difluoroethane (R152a) , 1,1,1,3,3-Pentafluorbutane (R365mfc) and particularly their mixture. A vapor compression refrigeration cycle was simulated to see the performance of a single stage cycle at different conditions. A comparative study including a wide range of effective parameters was done. The exergy destruction technique as a new factor for the design of the vapor compression refrigeration cycles was implemented. R365mfc has achieved higher Coefficient of Performance (COP). Furthermore an exergy analysis was employed for every component and for the whole cycle. Among all, the mixtures of R152a and R365mfc have provided higher exergy destruction, power consumption, cooling and heating capacity as compared with the pure fluids.
Keywords- R152a, R365mfc, Mixture, Exergy Destruction, COP, Refrigeration Cycle.

I. INTRODUCTION

The last two decades have witnessed a significant increase of the harmful materials of Chlorofluorocarbons (CFCs) and Hydro-chlorofluorocarbons (HCFCs) based refrigerants in the atmosphere. These materials have been widely used as solvents, foam blowing agents, aerosols and especially as refrigerants due to their preeminent properties such as stability, non-toxicity, non-flammability and availability. However, these substances have a harmful effect on the earth's protective ozone layer. Since the declaration of the Montreal protocol in 1987 the use of these substances is consequently regulated [1]. Moreover the CFCs had been detected as substances contributing to global warming. Nowadays the world leading countries have legalized to stop the consumption of CFCs by replacing them with HFCs [3-5].

This paper supports more accurate energy efficient data [18] for those materials, including the pure fluids and mixtures of R152a and R365mfc, through the use of the exergy analysis techniques [19]. The R152a is a HFC type refrigerant [21]. This fluid has zero ODP and a GWP value of 120. The boiling point temperature at a pressure of p=1.013 bar is ϑ^s=–24.0°C, the molecular weight M=66.1 kg/kmol. R152a is a medium pressure refrigerant for the medium temperature refrigeration range [2]. The physical, thermodynamic and refrigeration characteristics are similar to those of the refrigerants R12 and R134a. Although R152a is a good refrigerant substitute for R12, it is not used in its pure form because of its flammability (flammability limits 3.7–21.8% by volume in air). For this reason, R152a is labeled as" highly flammable". The R365mfc is a HFC type refrigerant. This fluid has zero ODP and a GWP value of 782. The boiling point temperature at a pressure p=1.013 bar is ϑ^s=+41.4°C, the molecular weight is at M=148.07 kg/kmol. R365mfc is a new fluid which is mainly used for the production of rigid polyurethane foams used in insulation purposes where a liquid foaming agent with a low thermal conductivity and a high vapor pressure at low temperatures are needed. Regarding to thermodynamic properties and safety requirements, R365mfc is employed as a main component in binary blends with 7 or 13 mass% with 1,1,1,2,3,3,3-Heptafluoropropane (R227ea) in the production of liquid foaming agents [2]. Besides the fact that this fluid has no negative impact on the ozone layer, further environmental benefits can be achieved using R365mfc, like in the production of polyurethane foams. To the best of our knowledge, only a very limited amount of energy efficient data of R365mfc is available in the literature [20].

II. SIMULATION CRITERIA

A. Compressor

In a refrigeration cycle, this component has two main functions within the refrigeration cycle. It's main function is to increase the pressure of the refrigerant vapor through the process of compression, to reach the condensation temperature necessary to reject the heat to the ambient, (see Fig.4). The process '1' to '2`'' symbolizes an ideal isentropic compression, in which there is no heat exchange with the surroundings and no entropy production. However, in reality there are always some heat losses from the compressor and also some dissipation of energy due to mechanical friction and potential leakage in the compressor. An

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adiabatic compressor involves power input and energy entering and leaving by the fluid stream. The first law of thermodynamics for a steady state reads for this case, neglecting potential and kinetic energy of the fluid streams [13]:

$$\dot{W}_{in} + \dot{m}h_1 = \dot{m}h_2 \tag{1}$$

or:

$$\dot{W}_{in} = \dot{m}(h_2 - h_1) \tag{2}$$

where $\dot{m}$ is the mass flow rate of the refrigerant in (kg/sec). Considering an adiabatic compressor with a steady-flow compression process, an entropy balance may be written as [14]:

$$\dot{S}_{in} - \dot{S}_{out} + \dot{S}_{gen} = \Delta\dot{S}_{sys} = 0 \tag{3}$$

$$\dot{S}_{gen} = \dot{S}_{out} - \dot{S}_{in} \tag{4}$$

$$\dot{S}_{gen} = \dot{m}(s_2 - s_1) \tag{5}$$

For an isentropic ideal compression $\dot{S}_{gen} = 0$, thus $s_2 = s_1$.

The exergy destruction during the compression process yields [17]:

$$\dot{E}x_{in} - \dot{E}x_{out} - \dot{E}x_{dest} = 0 \tag{6}$$

By arrangements equation (6) yields:

$$\dot{E}x_{dest} = \dot{E}x_{in} - \dot{E}x_{out} = \dot{W}_{in} + \dot{E}x_1 - \dot{E}x_2 = \dot{W}_{in} - \Delta\dot{E}x_{12} \tag{7}$$

$$\dot{E}x_{dest} = \dot{W}_{in} - \dot{m}[h_2 - h_1 - T_0(s_2 - s_1)] \tag{8}$$

$$\dot{E}x_{dest} = \dot{m}(h_2 - h_1) - \dot{m}[h_2 - h_1 - T_0(s_2 - s)] \tag{9}$$

$$\dot{E}x_{dest,Comp} = \dot{m}T_0(s_2 - s_1) = T_0\dot{S}_{gen} \tag{10}$$

where T_0 is the ambient temperature. The difference between isentropic and real adiabatic compression taking into account the dissipation of energy can be expressed by the isentropic compressor efficiency as follows:

$$\eta_{Comp,isen} = \frac{\dot{W}_{isen}}{\dot{W}_{act}} = \frac{\dot{m}(h_{2'} - h_1)}{\dot{m}(h_2 - h_1)} = \frac{(h_{2'} - h_1)}{(h_2 - h_1)} \tag{11}$$

where $h_{2'}$ is the outlet enthalpy of ideal isentropic compression. The exergy efficiency of the compressor may be expressed as the ratio of the reversible work to the actual work as follow:

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$$\eta_{Comp,ex} = \frac{\dot{W}_{rev}}{\dot{W}_{in}} = 1 - \frac{\dot{E}x_{dest,Comp}}{\dot{W}_{in}} \quad (12)$$

There are several types of compressors [12], divided into two types:

1- Positive displacement compressors
   - Screw compressors.
   - Rotary compressors.
   - Rolling piston compressors.
   - Scroll compressors.
2- Dynamic compressors
   - Turbo compressors.

In positive displacement compressors, a certain volume of vapor is captured in a space that is continuously reduced by the compressing device (piston, scroll, screw or similar) inside the compressor. The reduction in volume increases the pressure of the vapor during the operation of the compressor. The principle of dynamic compressors, also known as turbo compressors, is different. Here, the vapor is compressed by accelerating it with an impeller. The pressure is further increased in the diffuser, where the kinetic energy is transformed into enthalpy. Turbo compressors are interesting for very large capacities, where the inlet flows may be approximately 2000 m^3/h or more. Compressors can also be installed in either single or multistage configuration, and can be connected to each other in series or in parallel. The selection of the compressors depends on different factors such as refrigeration capacity, volumetric flow rate, compression ratio, thermal and physical properties of the refrigerants [6][15].

B. Condenser

The main objective of the condenser is to transfer the hot discharge vapor from the compressor at superheating conditions to a slightly sub-cooled liquid flow by transferring heat from the refrigerant to the heat sink. The basic operation of condensers is divided into three parts [15][7]:

2-2a: De-superheating. 2a-2b: Condensation. 2b-3: Sub-cooling.

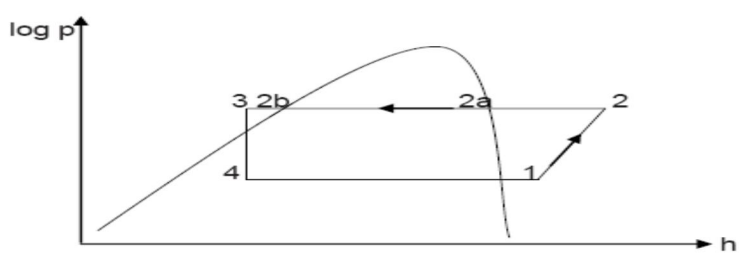

Fig.1. Condensing process.

All above operations can be carried out inside the condenser, or the de-superheating or sub-cooling operations could also be carried out in a separate heat exchanger. The heat rejection can be seen in a log p-h diagram as shown in Fig.1. The first part of the condenser de-superheats the gas to the saturation temperature. It is a one- phase heat transfer where the temperature of the refrigerant vapor decreases gradually, depending on the heat capacity of the refrigerants used. When the refrigerant reaches its saturation temperature at (state '2a'), the latent heat is rejected. The total heat rejection is represented in ('2a-2b').Finally, the fully condensed refrigerant (state '2b') is sub-cooled a few degrees to ensure that only liquid enters the expansion valve as indicated in (state '3')in which a one phase heat transfer operation occurs. The temperature of the refrigerant decreases during the de-superheating and the sub-cooling processes, but it remains constant during the condensing process, if it is a pure refrigerant. The energy rejected from the refrigerant heats the heat sink, whose temperature thus increases. The temperature of the heat sink can be increased to approach or even exceed the condensing temperature. However, there is a minimum temperature difference that must be considered for stable operation, well known as the pinch temperature. This minimum temperature occurs at the beginning of the

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condensation process (in a counter-current condenser) . The conservation of mass principle requires for a steady state:

$$\dot{m}_1 = \dot{m}_2 = \dot{m} \tag{13}$$

With negligible kinetic and potential energies the energy balance around the condenser gives:

$$\dot{Q}_H + \dot{m}h_3 = \dot{m}h_2 \tag{14}$$

The rate of the heat rejected can be written as:

$$\dot{Q}_H = \dot{m}(h_2 - h_3) \tag{15}$$

The overall heat transfer coefficient U_{Cond} in the condenser can be calculated as:

$$\dot{Q}_H = (UA)_{Cond}\Delta T = \dot{m}(h_2 - h_3) \tag{16}$$

ΔT presents the temperature gradient inside the condenser (condensing and ambient temperature). The exergy destruction can be determined by writing an exergy balance on the condenser as follow:

$$\dot{E}x_{dest} = \dot{E}x_{in} - \dot{E}x_{out} = \dot{E}x_2 - \dot{E}x_3 - \dot{E}x_{\dot{Q}_H} \tag{17}$$

$$\dot{E}x_{dest,Cond} = \dot{m}[h_2 - h_3 - T_0(s_2 - s_3)] - \dot{Q}_H\left(1 - \frac{T_0}{T_H}\right) \tag{18}$$

The exergy destruction in the de-superheating process can be given as:

$$\dot{E}x_{dest,De\,sup} = \dot{m}[h_2 - h_{2a} - T_0(s_2 - s_{2a})] - \dot{Q}_{De\,sup}\left(1 - \frac{T_0}{T_H}\right) \tag{19}$$

The exergy destruction in the condensing process can be given as:

$$\dot{E}x_{dest,Cond} = \dot{m}[h_{2a} - h_{2b} - T_0(s_{2a} - s_{2b})] - \dot{Q}_{Cond}\left(1 - \frac{T_0}{T_H}\right) \tag{20}$$

The exergy destruction in the sub-cooling process can be given as:

$$\dot{E}x_{dest,Sub} = \dot{m}[h_{2b} - h_3 - T_0(s_{2b} - s_3)] - \dot{Q}_{Sub}\left(1 - \frac{T_0}{T_H}\right) \tag{21}$$

Where T_0 is the ambient temperature. Usually $T_0 = T_H$.

C. *Throttling Device*

In practice, throttling devices, also called either expansion valves or throttling valves, are used to reduce the refrigerant condensing pressure from high conditions to low conditions at the evaporator. By the throttling process the flow of the refrigerant fluid will be regulated to pass into the evaporator. These devices are also designed to proportion the rate at which the refrigerant enters the

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cooling coil to the rate of evaporation of the liquid refrigerant in the coil; the amount depends on the amount of heat being removed from the refrigerated space. The most common throttling devices are [15] [12]:

• Thermostatic expansion valves
• Constant-pressure expansion valves
• Float valves
• Capillary tubes.

The steady-flow energy balance with negligible kinetic and potential energies is given by:

$$\dot{m}h_3 = \dot{m}h_4 \tag{22}$$

$$h_3 = h_4 \tag{23}$$

A throttling valve is essentially an isenthalpic device. The entropy balance on the throttling valve is given by:

$$\dot{S}_{in} - \dot{S}_{out} + \dot{S}_{gen} = \Delta\dot{S}_{sys} = 0 \tag{24}$$

$$\dot{S}_{gen} = \dot{m}s_4 - \dot{m}s_3 = \dot{m}(s_4 - s_3) \tag{25}$$

$$\dot{E}x_{dest,Expa} = \dot{m}T_0(s_4 - s_3) = T_0\dot{S}_{gen} \tag{26}$$

D. Evaporator

In the evaporator, the refrigerant boils by absorbing energy from the heat source which reduces its temperature. This heat source may be a gas or liquid, depending on the system [12]. The evaporation process occurs as shown in Fig.2. The sub-cooled liquid refrigerant at high pressure (state '3') is expanded through the expansion valve, the temperature decreases (state '4'). The mixture of liquid and vapor from the expansion valve enters the evaporator and starts to boil, since the heat is transferred from the heat source , the evaporating refrigerant absorbs energy from the heat source, whose temperature is reduced. When 100% of the refrigerant has become saturated vapor (state'4a'), the temperature of the vapor will start to increase , the vapor becomes then superheated. The refrigerant flow leaving the evaporator will be 100% superheated vapor as in (state'4') as shown in Fig.2 [15]. The total energy absorbed by the refrigerant in this case consists of the latent energy of evaporation plus the sensible energy of superheating.

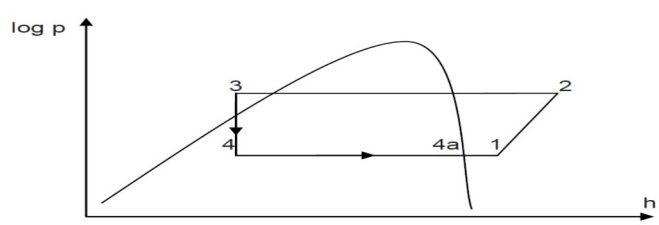

Fig.2. Evaporating process.

The evaporation temperature of the pure refrigerant remains constant and corresponds to the saturation pressure level. With negligible kinetic and potential energies the energy balance around the evaporator gives:

$$\dot{Q}_L + \dot{m}h_4 = \dot{m}h_1 \tag{27}$$

The rate of the heat absorbed can be written as:

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$$\dot{Q}_L = \dot{m}(h_1 - h_4) \tag{28}$$

The overall heat transfer coefficient U_{Evap} in the evaporator can be calculated as:

$$\dot{Q}_L = (UA)_{Evap}\Delta T = \dot{m}(h_1 - h_4) \tag{29}$$

The coefficient of performance (COP) of the refrigeration system becomes:

$$COP = \frac{\dot{Q}_L}{\dot{W}_{in}} \tag{30}$$

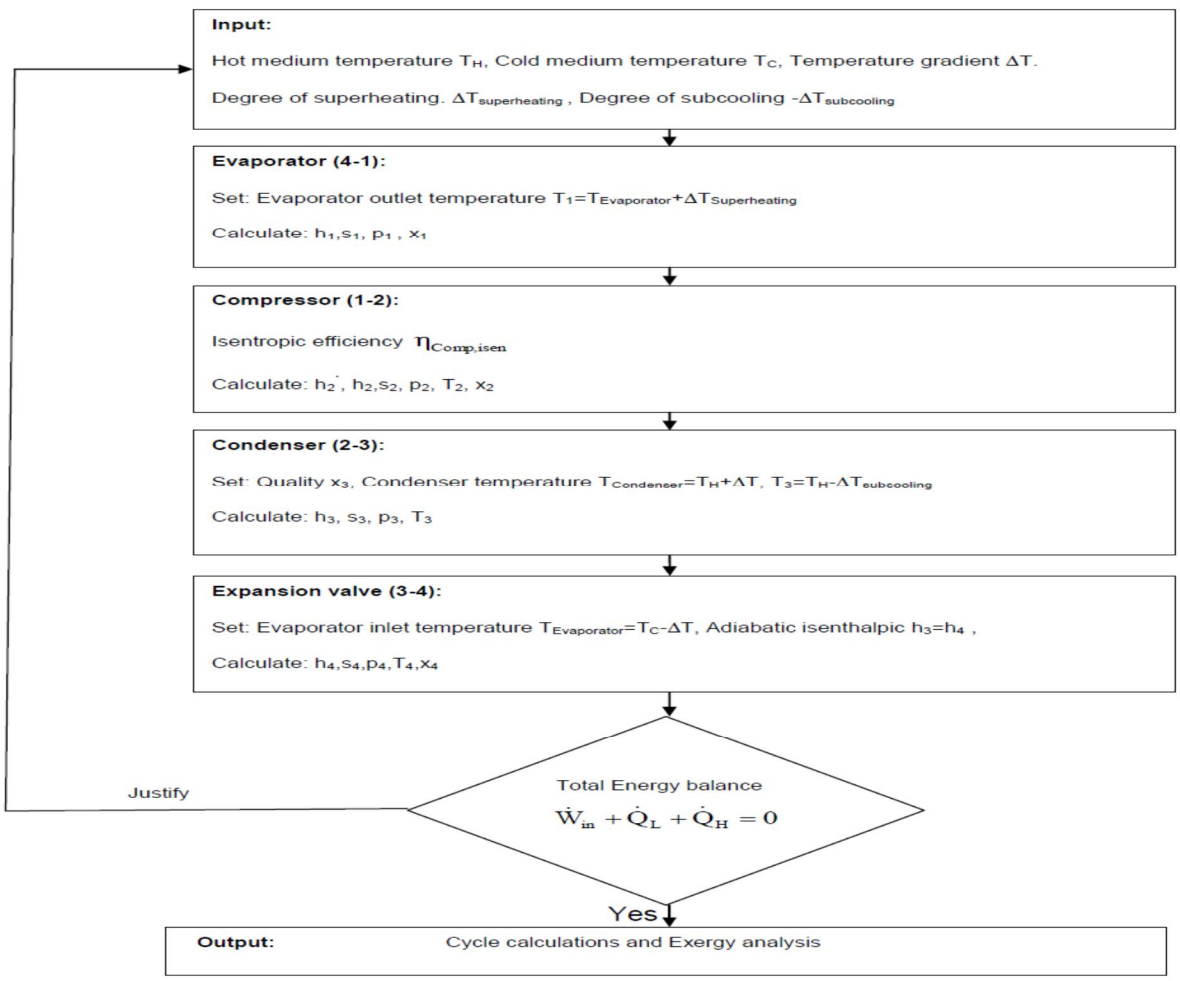

Fig.3. Structure of the program used for calculating the refrigeration cycle properties.

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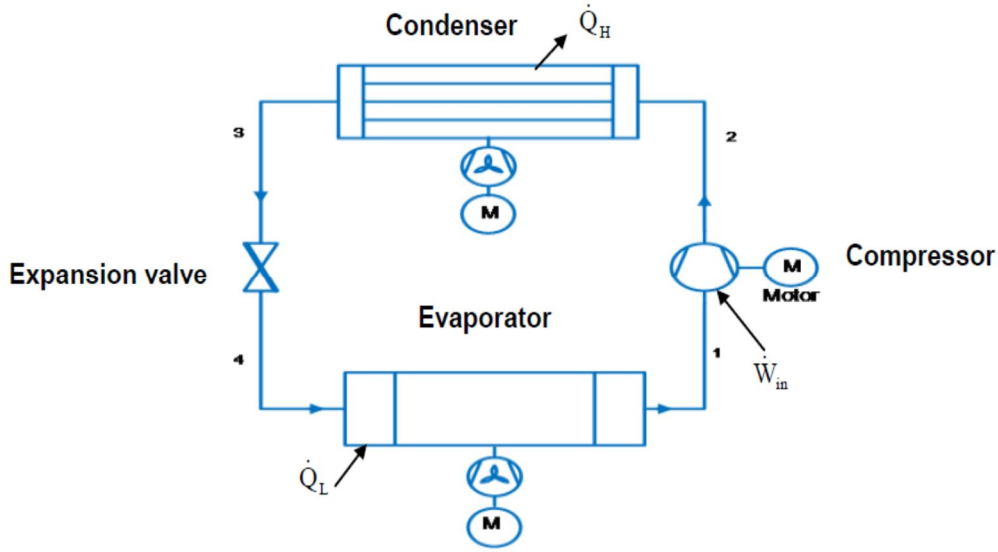

Fig.4. Components and energy diagram for a single stage vapor compression refrigeration cycle.

The exergy destruction can be determined by writing an exergy balance on the condenser as follow [16]:

$$\dot{E}x_{dest} = \dot{E}x_{in} - \dot{E}x_{out} = \dot{E}x_4 - \dot{E}x_1 - \dot{E}x_{\dot{Q}_L} \tag{31}$$

$$\dot{E}x_{dest,Evap} = \dot{m}[h_4 - h_1 - T_0(s_4 - s_1)] - \left[-\dot{Q}_L\left(1 - \frac{T_0}{T_L}\right)\right] \tag{32}$$

The exergy efficiency of the evaporator may be expressed as the ratio of the exergy of the heat transferred to the low temperature medium to the exergy decrease of the refrigerant due to receiving heat from the cold reservoir:

$$\eta_{Ex,Evap} = \frac{\dot{E}x_{\dot{Q}_L}}{\dot{E}x_4 - \dot{E}x_1} = \frac{-\dot{Q}_L\left(1 - \frac{T_0}{T_L}\right)}{\dot{m}[h_4 - h_1 - T_0(s_4 - s_1)]} = 1 - \frac{\dot{E}x_{dest,Evap}}{\dot{E}x_4 - \dot{E}x_1} \tag{33}$$

The total exergy destruction of the cycle can be written as:

$$\dot{E}x_{dest,total} = \dot{E}x_{dest,Comp} + \dot{E}x_{dest,Cond} + \dot{E}x_{dest,Expa} + \dot{E}x_{dest,Evap} \tag{34}$$

All the equations above were implemented into a Matlab program. The properties were calculated through the use of the REFPROP [9] program. This program, developed by the National Institute of Standards and Technology (NIST) [8], calculates the thermodynamic and transport properties of industrially important fluids and their mixtures [10] [11]. REFPROP is an acronym for REFerence fluid PROPerties, this program is accompanied with the most accurate pure fluid and mixture models currently available. The program uses three models for the thermodynamic properties of pure fluids: equations of state explicit in Helmholtz energy, the modified Benedict-Webb-Rubin equation of state, or an extended corresponding states (ECS) model [11]. Mixture calculations employ a model that applies mixing rules to the Helmholtz energy of the mixture components; it uses a departure function to account for the departure from ideal mixing [9]. The algorithm was employed using a MATLAB code communicates with REFPROP program version 9.1. The subroutines used in this program were developed at the Institute for Thermodynamic,

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University of Hannover, Federal Republic of Germany.

III. RESULTS AND DISCUSSIONS

Fig.5 illustrates the effect of the cold medium temperature on the coefficient of performance COP for system with different refrigerants. It is clearly seen that there is a big difference between the COPs for both pure fluid and the mixture of R152a and R365mfc. It is seen that some refrigerants are more efficient than others. A comparison results indicates that a system consisting of (30%R152a, 70%R365mfc) and (50%R227ea, 50%R365mfc) are about equally efficient. Also system consisting of (50%R134a, 50%R365mfc), (80%R152a, 20%R365mfc) and (50%R152a, 50%R365mfc) are about equally efficient. Among all the refrigerants with reach a high COP R365mfc is followed by R152a. It is to be noted the mixture of R365mfc and R152a does not exceed the performance efficiency of its pure fluids.

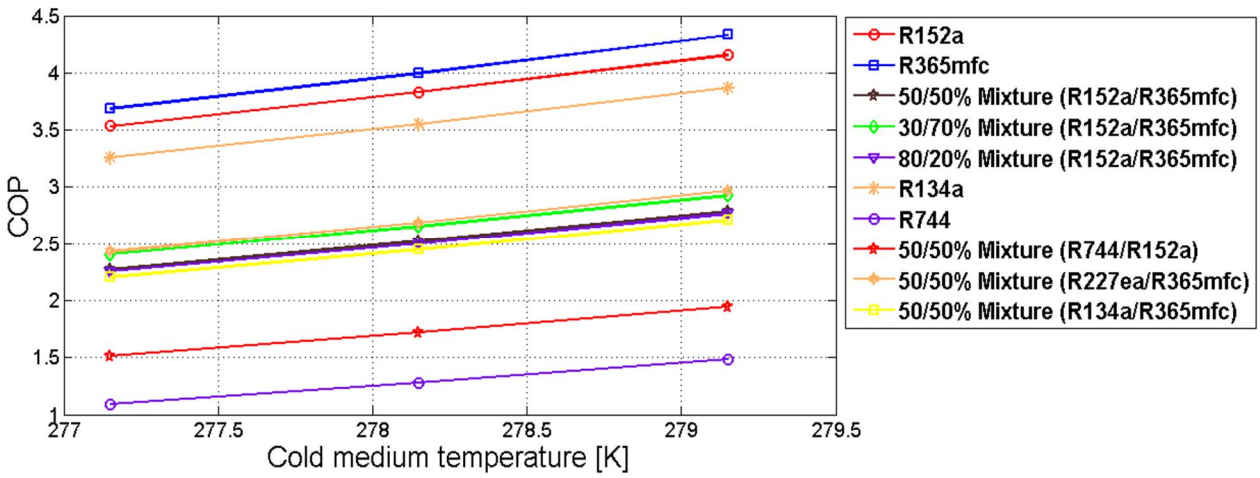

Fig.5. Variation of COP with cold medium temperature (Ambient temperature at T=298.15K).

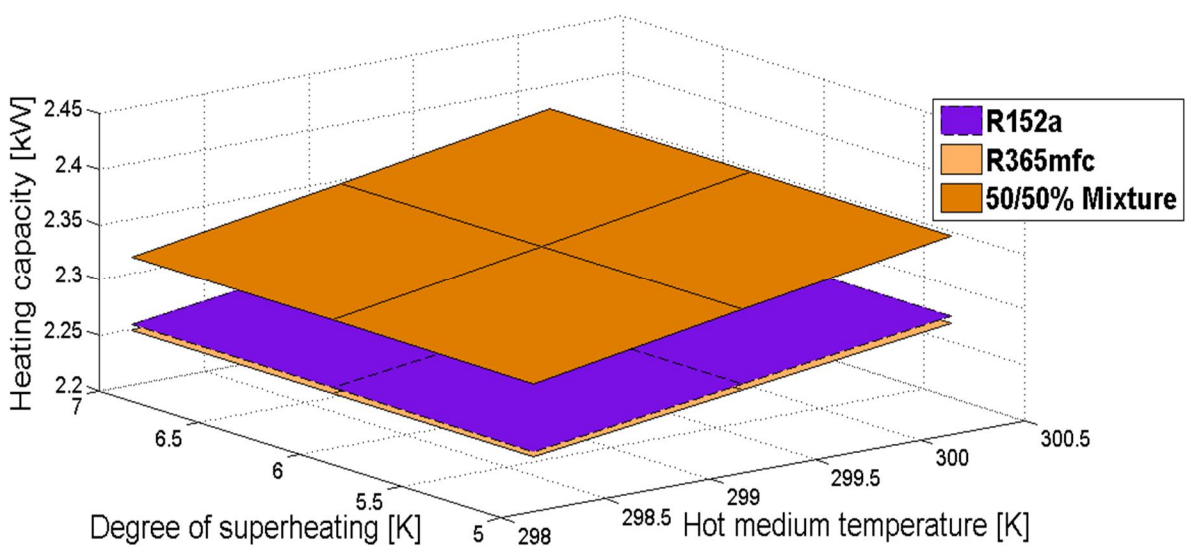

Fig.6. Hot medium temperature and superheating effect on heating capacity.

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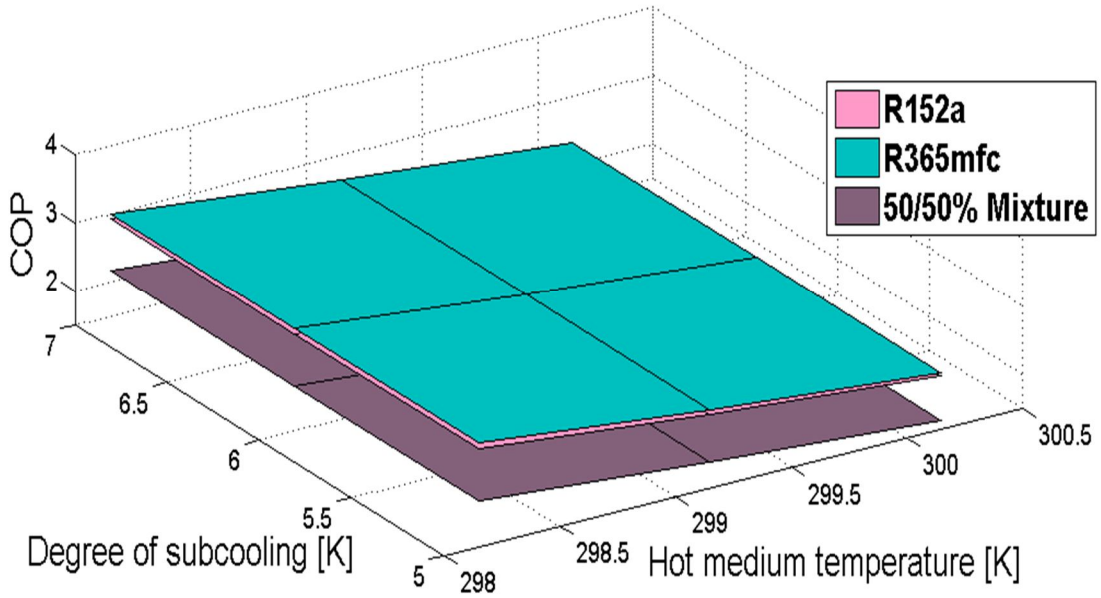

Fig.7. Hot medium temperature and sub-cooling effect on coefficient of performance.

As shown in Fig.6 the mixture form of the refrigerants was provided a higher heating capacity compared to pure fluids which indicated a less capacity. Likewise, the process of cooling the refrigerant fluid below its condensing temperature at a given pressure is known as a sub-cooling. Sub-cooling provides 100% refrigerant liquid to enter the expansion device, preventing vapor bubbles from impeding the flow of refrigerant through the expansion valve and maintaining stable flow of liquid. If the sub-cooling is caused by a heat-transfer method external to the refrigeration cycle, the refrigerant effect of the system is increased, because the sub-cooled liquid has less enthalpy than the saturated liquid.

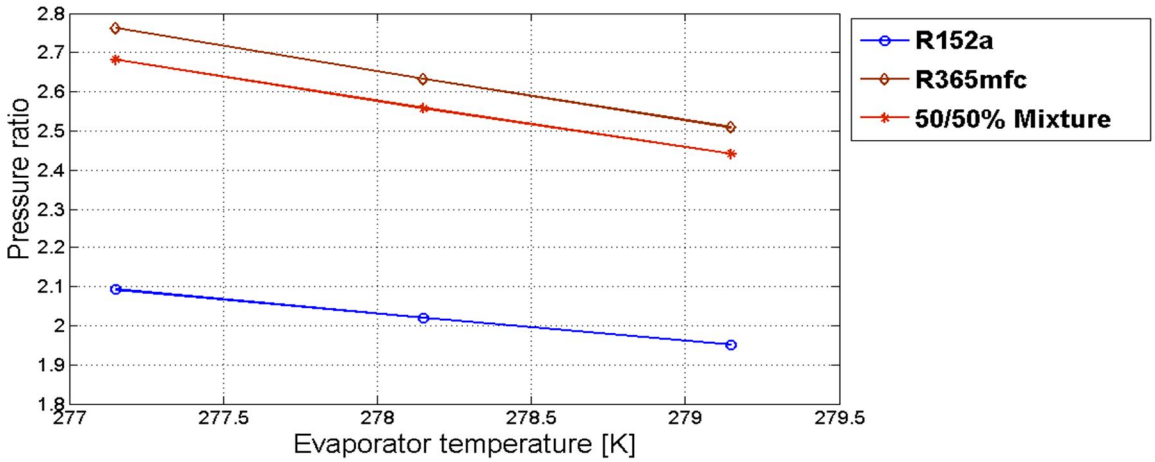

Fig.8. Variation of evaporator temperature with pressure ratio.

Note that the performance of a simple vapor-compression refrigeration system can be significantly improved by further cooling the liquid refrigerant leaving the condenser coil. This sub-cooling of the liquid refrigerant can be accomplished by adding a mechanical sub-cooling loop in a conventional vapor-compression cycle.

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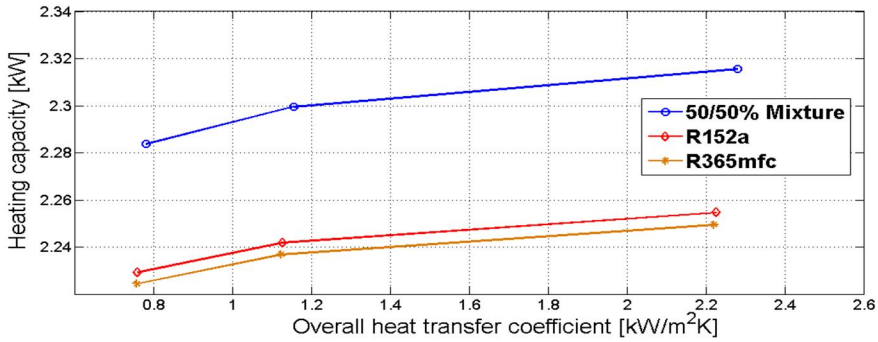

Fig.9. Variation of overall heat transfer coefficient with heating capacity.

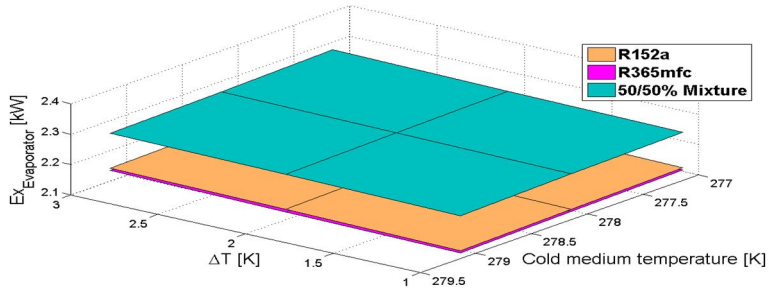

Fig.10. Total exergy destruction in evaporator.

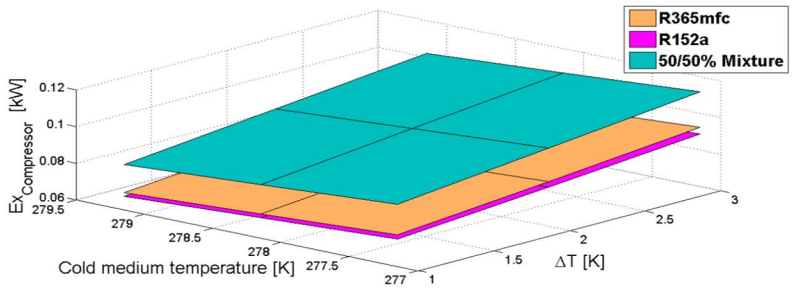

Fig.11. Total exergy destruction in compressor.

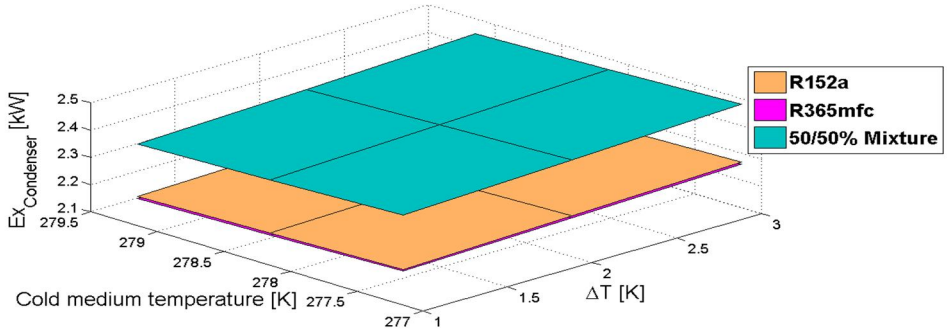

Fig.12. Total exergy destruction in condenser.

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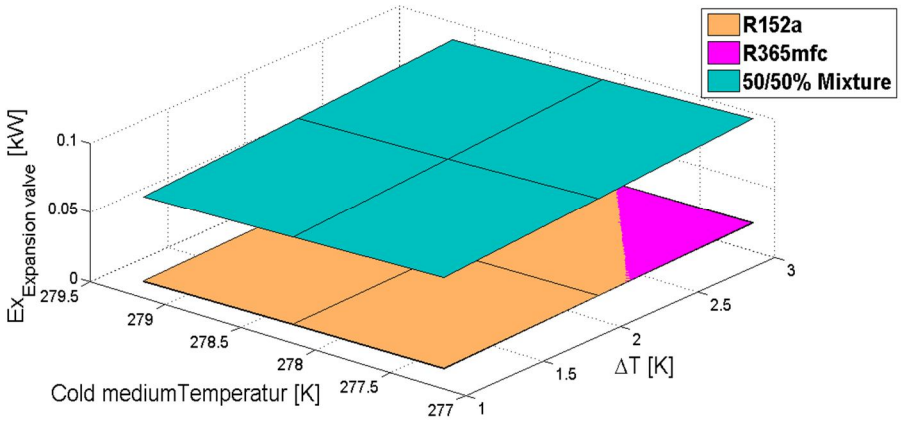

Fig.13. Total exergy destruction in expansion valve.

As verified in Fig.7 the effect of the sub-cooling show an increase on the COP for all fluids used, with R365mfc confirmed a higher COP rather than R152a and the mixture form which presented a lower COP. Fig.8 illustrates the variations of pressure ratio with varying evaporator temperature. The results show that the pressure ratio decreases with increase in the evaporator temperature. The trends exhibit similarity for all working fluids, a higher value observed by R365mfc followed by the mixture form and R152a respectively.

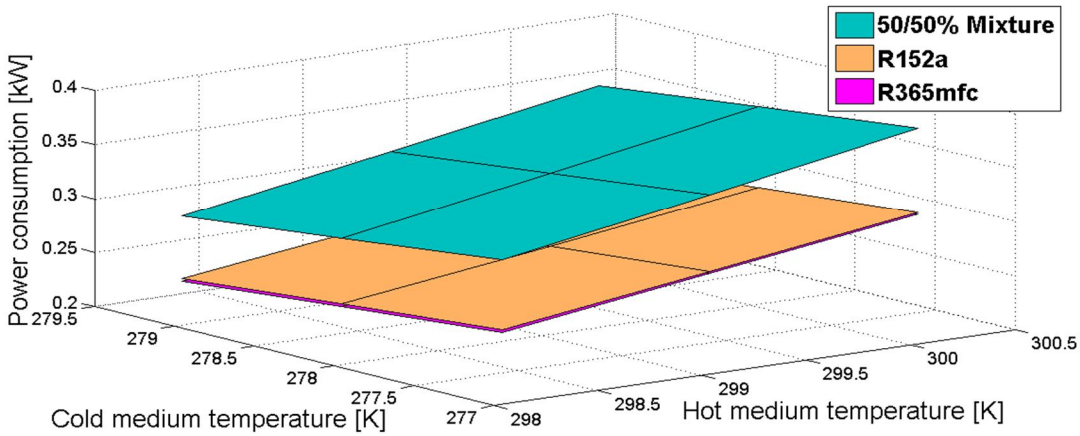

Fig.14. Variation of power consumption with cold and hot medium temperature.

The pressure ratio for R365mfc is higher, that is to say heavier compressor work is required for R365mfc while the same compressor could be employed for the others. Comparing the overall heat transfer coefficients in the condenser as indicated in Fig.9, the mixture presents worst heat transfer coefficient in compared with pure fluids. The heat transfer from both heats generating equipment in the refrigeration cycle is always accompanied by exergy transfer. The exergy destruction of each component in the cycle was estimated. As shown in Fig.10 to Fig.13; the use of the mixture form of the refrigerants implied a higher exergy destruction value among all the components of the cycle. The simulation results verified that the greatest irreversibility occurred in the condenser and the minimum irreversibility is occurred in the expansion valve for most working fluids. Possible irreversibilities in the evaporator and the condenser occur normally due to the temperature differences between the two heat exchanger fluids, pressure losses, flow imbalances and heat transfer with the environment. The irreversibility in the compressor is normally occurs from mechanical–electrical losses due to imperfect electrical, mechanical and isentropic efficiencies and emphasize the need for careful selection of this equipment, since components of inferior performance can considerably reduce overall system performance.

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The minimum irreversibility is associated with the expansion valve possibly due to the pressure drop of the refrigerant passing through it. Fig.14. verified the variation of the power consumption with hot and cold medium temperatures. The illustration implied that the mixture form of the refrigerant requires more heat consumption to drive the cycle rather than R152a and R365mfc respectively. It is also observed for all working fluids, the higher is the power consumption the lower is the COP.

Table I. Properties of the fluids [20][21].

Component	M_w (kg/kmol)	T_c (K)	P_c (bar)	ρ_c (kg/m^3)	ω
R365mfc	148.07	460.0	3266.0	473.84	0.377
R152a	66.051	386.41	4516.8	368.0	0.27521

According to the thermodynamics properties of the working fluids, It is observed that both pure fluids have higher critical temperature, with R365mfc achieved a higher value than R152a. The critical temperature should be above the operating system temperature, if the operating temperature is above the critical temperature, it is impossible to condense the vapor by compressing it to high pressures. The refrigerant R365mfc achieved a higher molecular weight than R152a. The molecular weight of the refrigerant affects the compressor size since the specific volume of the vapor is directly related to it , the high molecular weight gives high specific volumes for the vapor ,which is preferred for centrifugal compressors ,whereas for reciprocating compressors a low molecular weight for the refrigerant is advantageous.

IV. CONCLUSIONS

The simulation results of this work have proved that, the greatest irreversibility occurred in the condenser and the minimum irreversibility is occurred in the expansion valve for most working fluids. The consequences of these results were discussed in details. The results data also, implied that the mixture form of the refrigerant requires more heat consumption to drive the cycle, as well as greater exergy destruction, rather than the pure fluids R152a and R365mfc.

V. ACKNOWLEDGMENTS

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Abbreviation

CFCs	chlorofluorocarbons
HCFCs	hydrochlorofluorocarbons
HFCs	hydrofluorocarbons
GWP	global warming potential
ODP	ozone depletion potential
NIST	national institute of standard and technology
COP	coefficient of performance
$\dot{m}$	mass flow rate
$\dot{h}$	enthalpy
$\dot{S}_{gen}$	entropy generation
$\dot{W}$	compressor work
η	exergetic efficiency

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REFERENCES

[1] United Nations Environment Program: Montreal protocol on substances that deplete the ozone layer, 2000.
[2] H. Buchwald, F. Flohr , J. Hellmann, H. König , C. Meurer: Solkane Pocket Manual , Refrigeration and air conditioning technology, Solvay Fluor und Derivate,2007.
[3] B. G. Lee, H. Kim, H. Kim, S. D. Lee, and I. Seo, J. Ind. Eng.Chem.,3,160 (1997).
[4] H. Kim, H. S. Kim, B. G. Lee, Y. S. Kwon, and K. Y. Park, J. Ind. Eng. Chem., 3, 203 (1997).
[5] K. H Lee, K.W. Seo, Y. I. Mok, K. Y. Park, and B. S. Ahn, J. Ind. Eng. Chem., 6, 305 (2000).
[6] Dincer. Ibrahim, Mehmet Kanoglue: Refrigeration systems and applications, second edition, 2010.
[7] Refrigerants application guide, McQuay air conditioning, AG31-007, 2002.
[8] NIST Reference Fluid Thermodynamic and Transport Properties-REFPROP, Version 9, Boulder, Colorado.
[9] NIST Reference National Institute of Standards and Technology, Physical and Chemical Properties Division,, Boulder, Colorado.
[10] Outcalt, S.L. and McLinden, M.O.: A modified Benedict-Webb-Rubin equation of state for the thermodynamic properties of R152a (1,1-difluoroethane), J. Phys. Chem. Ref. Data, 25(2):605-636, 1996.
[11] McLinden, M.O. and Lemmon, E.W.: Thermodynamic Properties of R-227ea, R-365mfc, R-115, and R-131, J. Chem. Eng. Data, 2013.
[12] Stoecker, W.F.: Industrial refrigeration handbook, first edition, 1998.
[13] Reinhard Radmacher ,Yunho Hwang: Vapor compression heat pump with refrigerant mixtures, pp. 7-8, 2005.
[14] Yunus Cengel: Thermodynamics an engineering approach, second edition, 2007.
[15] SWEP Refrigeration handbook, www.swep.net.
[16] A. Bejan, Entropy Generation Through Heat and Fluid Flow, Willey, New York, 1982.
[17] R.A. Gaggioli, Available energy and exergy, Internat. J. Appl. Thermodyn. 1 (1–4) (1998) 1–8.
[18] M.J. Moran, Availability Analysis: A Guide to Efficient Energy Use, Prentice-Hall, Englewood Cliffs, NJ, 1982.
[19] Dincer, I. (2002b) `The role of exergy in energy policy making', Energy Policy, Vol. 30,pp.137-149.
[20] A. P. Froeba, K. Krzeminski, and A. Leipertz, Int. J. Thermophys. **25**:987 (2004).
[21] Reiner Tillner-Roth: Die thermodynamische Eigenschaften von R152a, R134a und ihren Gemischen, Messungen und Fundamentalgleichungen, PhD thesis, 1993.

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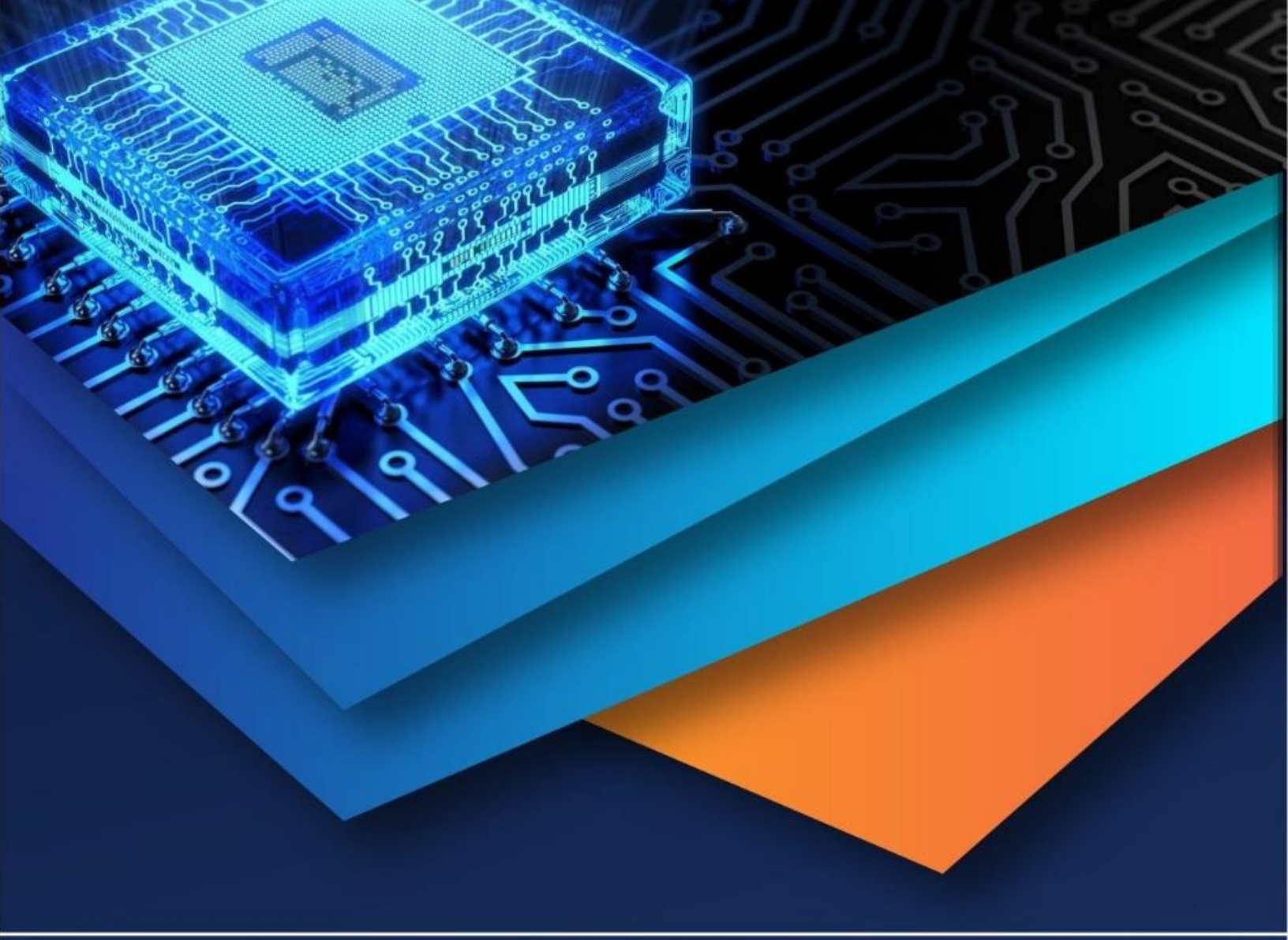

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