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Automobile A.C. By Utilizing Exhaust Gases

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I. INTRODUCTION

Energy efficiency has been a major topic of discussions on natural resources preservation and costs reduction. Based on estimates of energy resources reduction at medium and long terms, internal combustion engines are potential energy sources for absorption refrigeration systems, as about one third of the energy availability in the combustion process is wasted through the exhaust gas. Thus use of the exhaust gas in an absorption refrigeration system can increase the overall system efficiency.

II. LITERATURE REVIEW

Absorption refrigeration was discovered by Nairn in 1777, though the first commercial refrigerator was only built and patented in 1823 by Ferdinand Carré, who also got several patents between 1859 and 1862 from introduction of a machine operating on ammonia–water. The absorption refrigeration system went through ups and downs, being the antecessor of the vapour compression refrigeration system in the 19th century. By that time systems operating on ammonia–water found wide application in residential and industrial refrigerators. Systems operating on lithium bromide–water were commercialized in the 1940's and 1950's as water chillers for large buildings air conditioning.

Perez-Blanco- Substitution of petroleum-based combustion fuels in the 1970's affected the application of absorption refrigeration, but, at the same time, new opportunities arose, such as usage of solar energy to operate this system energy costs and other factors has contributed to frequent use of low temperature energy waste from chemical and commercial (supermarket) industries to operate absorption refrigeration systems.

III. PROPOSED MODEL OF AUTOMOBILE AIR CONDITIONING SYSTEM

A. Principle

The proposed model of automobile air conditioning system is based on vapour absorption system. The vapour absorption system uses low grade heat energy, instead of mechanical energy as in case of vapour compression system, in order to change the condition of the refrigerant required for the operation of the refrigeration cycle.

B. Aqua Ammonia System

The absorption refrigeration system with a pump uses ammonia as the refrigerant and an aqueous ammonia solution as the absorbent. Any means of heating can be used, but natural gas, steam or LP gas and exhaust gas from an automobile are the most popular. The systems operate under two pressures. The high side pressure is from 200 to 300 psi and low side, 40 to 60 psi. The high and low side are separated by check valves, liquid traps, a pump or other controlling devices. The operational system can be divided into four sections including generator, condenser, evaporator and absorber.

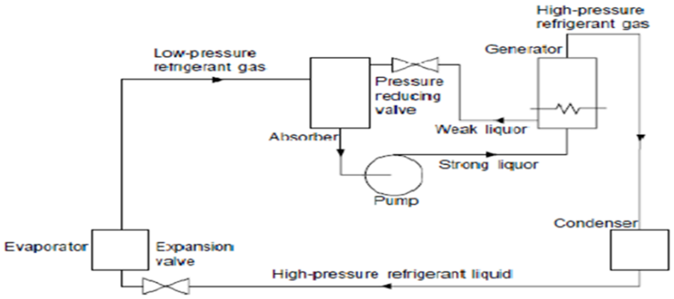
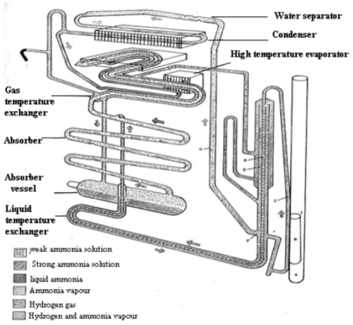


Fig. Aqua ammonia system

The generator as shown in figure is heated by a external source. Heat causes the liquid to boil and the ammonia in the liquid turns to vapour. The vapour will rise up through the tube to the air cooled condenser. In the condenser, heat from the vapour is removed by the cooler air passing across it. The vapour will condense into a liquid which then acts as a refrigerant.

C. Electrolux System

A typical unit is shown in figure it is charged with ammonia, water and hydrogen. The amount of this combined solution is at a pressure which will allow the ammonia to condense at room temperature. The necessary heat for the generator is obtained by using either a gas burner underneath the tube, or waste heat.



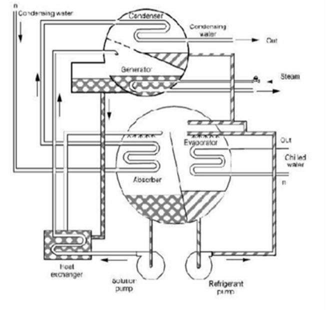


Fig. Electrolux System

Fig. Lithium Bromide System

D. Lithium Bromide System

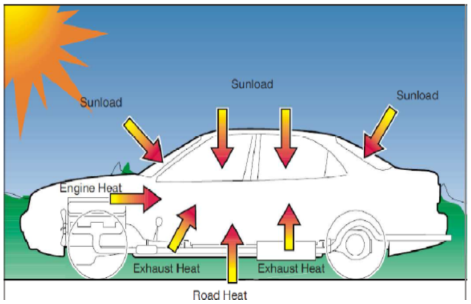
The lithium bromide absorption refrigeration system uses a solution of lithium bromide in water. In this system, the water is being used as a refrigerant whereas lithium bromide, which is a highly hydroscopic salt, as an absorbent. The lithium bromide has a strong affinity for water vapour because of its very low vapour pressure. Since lithium bromide solution is corrosive, therefore inhibitors should be added in order to protect the metal parts of the system against corrosion. Lithium chromate is often used as a corrosion inhibitor. This system is very popular for air conditioning in which low refrigeration temperatures are required.

IV. CASE STUDY

A. Cooling Load Calculation

One of the critical issues or tasks in designing any system, especially car air conditioning system, is to calculate the load that should be removed from the desired place. There are so many considerations that should be taken into account in calculating the load. Heat sources, which will impose loads on the cooling equipment, may be broken down as follows:

Radiation heat through the glass.
Conductive heat through the body.
Convective heat due to infiltration.
Convective heat due to fresh air intake.
Passengers’ heat.
Instruments heat.



For cooling load calculation we had considered the design condition for the conditioned space 25°C DBT and 60 % RH. The outdoor condition is 45°C DBT and 30% RH. For the purpose of the case study we had selected Ambassador Car. The various cooling loads are estimated as follow.

Heat load due to occupants (cl1)
cl1 =Np* Qs Per person
Where, Np=passenger capacity of car.
=5*115
=575 W
Heat load due to lights (Cl2)
Cl2 =NL* W + wattage of instrumental lamp
Where, NL= no. of lights in the car.
=2*10+10
= 30 W
Heat load due to appliances (Cl3)
Cl3= wattage of music system.
= 200
= 200 W
Heat load due to infiltration (Cl4)
Cl4 = 1.2CFMi*(to-ti)
CFM = ACH* Vc/60
Where,
t_o= outdoor air temperature ^{0}F, t_i = indoor air temperature ^{0}F
CFMi = air infiltration rate in (CFM), Vc = internal volume of the car (ft^3)
ACH = number of air change per hour due to infiltration
=1.2*(34.01/60)
= 0.68 ft^3/hr
=1.2*0.6802*(113-77)
=27.8 Btu/hr
=7.9058 W
Heat Load due to ventilation (Cl5)
Cl5=1.1*CFMv*(to-ti)
CFMv=Np*ACH
Where, CFMv=air ventilation flow rate, ACHv= number of air change per hour per person
CFMv =5*3.767
=18.83 CFM
=1.1*18.83*(113-77)
=745.866 Btu/hr
=218.397 W
Heat load from roof of car by conduction (Cl6)
Cl6=Ur*Ar*(tr-ti)
Where, Ur= overall heat transfer coefficient of roof (W/m2k)
= 4.63 W/m^2k
Ar = area of roof in (m2), tr = temperature of roof
Temperature of roof can be found by following relation
Itα= hout (tr-to) + (tr -ti) / (1/hin+1/k)]+ε*σ*(tr4-to4)

Where, It = total incident solar radiation on the roof, (W/m^2)
= 300 W/m^2hr
α = absorptive of roof, h_{out} = heat transfer coefficient of outside air
h_{in} = heat transfer coefficient of inside air, k = thermal conductivity of roof material
σ = Stefan Boltzmann constant, ε = roof emissivity
300*0.08=41.6*(tr-113) + [(tr-77) / (1/6+1/078)] + [0.12*5.67*10 - 8(tr^4-113^4)]
Solving by trial and error method,
tr =114.51^{o}F ≈ 46^{o}C
Cl6 = 4.63*2.025*(46-25)
=188 W

Heat conducted trough glass (Cl_7)
Cl7= Ug*Ag*(to-ti)
Where, Ug = overall coefficient of glass
Ag = area of glass, to = outdoor air temperature (^{o}C), t_i = indoor air temperature (^{o}C)
=5.06*2.1*(45-25)
=212.52 W
Heat load from side door panels and rear door (Cl8)
Cl8=U*A*(to-to)
Where, A=area of side door panels and rear door.
=4.63*5.3*(45-25)
=490 W
Heat load due to Insolation (Cl9)
Cl9= It (τ + Ua /hout)
It = Idh+ Id
Idh = beam radiation falling on the glass.
= (lex * cf) / exp (B / sin β)
= (2074 * 1.05) / exp (0.7/ sin50)
= 874 W/m^2
Id=diffuse radiation falling on the glass
= cIdh* (l+cosΣ)/2
=0.1*874*(1+cos40)/2
=77.17 W/m^2
Where, Iex = extra terrestrial solar radiation, Cf = clearness factor, B = extinction coefficient
τ= glass transmissivity, from table12.2, β = solar angle with the horizon,
Σ= angle between the surface and the horizontal, C=constant
It= 874+77.17
It = 951.17 W/m2
Cl9=951.17*(0.7+5.06*0.08)/41.6
= 666.69 W
Heat load from floor Cl10
Cl10=U*Af*(to-ti)
Where, A_f = effective area of floor
= 2.5*2.10(45-25)
=105 W
Total heat gain =addition of all heat gain
= Cl1 + Cl2 + Cl3 + Cl4 + Cl5 + Cl6 + Cl7 + Cl8 + Cl9 + Cl10
= 575+30+200+7.90+218.39+188+212.52+490.78+666.69+105
= 2693.79 W
Considering bypass factor of the cooling coil = 0.2
Capacity of cooling coil =2693.79+ (0.2*2693.79)
= 3232.54 W
Amount of tonnage required = (3232.54 /1000×3.517) = 0.919 ≈ 1 TR

B. Analysis Of An Aqua- Ammonia System
Let us consider following some temperature for case study
In an aqua ammonia vapour absorption system the following data is available: -
Temperature of weak solution in generator =156 ^{o}C
Temperature of strong solution admitted to generator = 40 ^{o}C
Temperature of condenser = Temperature of absorber = 40 ^{o}C
Temperature rise in evaporator =11 ^{o}C
Analysis for 1 tonne refrigeration capacity: -
At 20.3 bar and 156^{o}C
Poor solution concentration, ξa = 0.2

At 2.1 bar and 40^{o}C
Rich solution concentration, $\xi r = 0.34$
Also the concentration of vapour leaving the analyzer, in equilibrium with the entering rich solution is
$\xi 5 = \xi r^v = 0.93$

It is convenient to put the thermodynamic properties and flow rates at various sections in tabular form as follows.

State Point	Pressure P Bar	Temperature T o C	Concentration ξ kgNH3/ kg mixture	Enthalpy H kJ/kg	Flow factor M Kg/s
1	20.3	-	0.34	-	-
2	20.3	156	0.2	616	4.1
3a	20.3	67	0.2	205	4.1
4	2.1	40	0.34	63	5.1
4a	20.3	40	0.34	63	5.1
1a	20.3	-	0.34	-	5.1
5,7	20.3	-	0.913	1947	5.1
8	20.3	53	0.913	507	5.1
9	20.3	40	0.913	444	5.1
10	2.1	-16	0.913	444	5.1
11	2.1	5	0.913	1281	5.1
12	2.1	-	0.913	-	5.1

Table 4.1: thermodynamic properties and flow rates at various sections.

Mass flow of ammonia through evaporator *(m)*

$$m = 1 * \frac{3.5}{h11 - h10} = \frac{3.5}{1281 - 444}$$

m = 0.0041 Kg/sec.
Where,
h11 = Enthalpy of ammonia vapours at outlet of evaporator.
h10 = Enthalpy of ammonia vapours at inlet of evaporator.
Specific rich solution concentration rate (*f*)
$f = (\xi 5 - \xi a) / (\xi r - \xi a)$
$\xi 5$ = Concentration of vapour leaving analyser, ξa = Weak solution concentration
ξr = Rich solution concentration
= (0.913 - 0.2) / (0.34 - 0.20)
= 5.1 Kg/kg of vapour
Heat supplied per in the generator
$Qh = h5 - h2 + f\,(h2 - h1a)$
h5 = enthalpy of rich solution
h2 = Enthalpy of weak solution
=1947 – 616 + 5.1(616 - 393)
= 2468 kJ/kg (vapour)
Q (kJ/sec) = 0.0041*2468
= 10.1188 KJ/sec
Heat rejected in the absorber per sec.
$Qa = h12 - h3 + f\,(h3 - h4)$
= 1344 - 205 + 5.1(205 - 63)
=1865 KJ/kg
= 0.0041 *1865
=7.64 KJ/sec

Heat rejected in condenser
Qc = mr (h7 – h8)
= 0.0041(1947 – 507)
= 5.904 KJ/sec
Specific volume of solution at 40oC,at inlet to the pump
ύ4 = ξr ύNH3 + (1- ξ4) ύH2O
=0.34(1.726*10^{-3}) +0.66(1.008*10^{-3})
=1.251*10^{-3} m^3/kg
Pump work-
Wp = f ύ4 (pk – po)
= 5.1 1.251 *10 - 3(20.3 - 2.1)* 105*
= 11600 KJ/kg
= 11600 * 0.0041
Wp = 47.56 J/sec or W
Coefficient of performance-
COP = qo / qh + qp
Where,
qo = Refrigerating effect (kJ/kg), qh = Heat added in generator (kJ/kg),
qp = Pump work (kJ/kg).
COP = 837/ (2248 + 11.6)

COP = 0.34

Mass flow of ammonia through evaporator = 0.0041 kg/sec.
Heat supplied per in the generator = 10.1188 kJ/sec
Heat rejected in the absorber = 7.64 kJ/sec
Heat rejected in condenser = 5.904 kJ/sec
Pump work = 47.56 J/sec or W
Coefficient of performance of system = 0.34

C. Case Study Of An Four Cylinder Petrol Engine

Engine specification:-
Make – Premier Automobiles limited, India
Engine type- = 4 cylinder, in line, water cooled
Bore = 68 mm
Stroke =75mm
Capacity =1089cc
Brake power = 35.4 kW at 5000 rpm
Engine heat balance sheet at 1/4th throttle opening -
Air consumption = 0.0479 kg/sec
Fuel consumption = 0.001165 kg/sec
Calorific value of fuel = 45500 kJ/sec
Heat supplied = 0.001165 * 45500
=53.007 KJ/sec
Brake power of an engine (B.P.) = WN/2000 kW
= 5 * 2500/2000
= 6.25 kW
Heat equivalent to B.P. = 6.25 kJ/sec
Heat loss to exhaust gas *(Qg) = _mg * cpg * (tg – ta)*
= (0.0479 + 0.001165) * 1.1 * (384 – 30)

= 19.1053 kJ/sec

Heat loss in unaccounted form and
Heat rejected to cooling water = *Qs – B.P. – Qg*

=53.007 - 6.25 - 19.1053

=27.6517 kJ/sec

So heat available = Heat in exhaust
= 19.1053 KJ/sec (36.04%)

Let us assume that the effectiveness of heat exchangers be 0.7
Net heat available for VARS = 13.373 KJ/sec

Table 4.2 Measured data and exhaust useful heat

Throttle position opening	Engine speed (rpm)	Air pressure mm of H2O	Time for consumption of 50c.c fuel (sec)	Exhaust temperature oC	Mass of fuel Kg/ s (10-3)	Mass of air Kg/ s (10-3)	Useful exhaust heat Kg/ s
Idling	780	10	80	340	0.466	37.9	13.079
1/4th	2500	16	32	384	1.16	47.9	19.105
	2250	16.7	34	375	1.09	49.0	19.012
	2000	14.5	34	372	1.09	45.6	17.533
	1750	12.2	33	355	1.13	41.9	15.37
1/2	2500	20	30	465	1.24	53.6	26.22
	2250	21.2	28	448	1.33	55.24	25.97
	2000	18.2	27	409	1.381	51.19	21.84
	1750	17.1	29	415	1.28	49.6	21.51

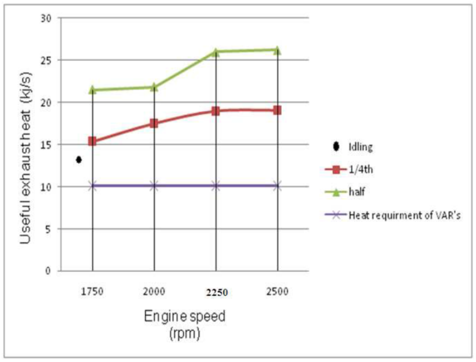


Figure 4.1 Engine speed against useful exhaust heat

The above graph shows the amount of useful exhaust heat at different speed and throttle valve openings is much greater than the heat requirement of vapour absorption refrigeration system. The amount of useful exhaust heat available even at idling condition is just sufficient to run vapour absorption refrigeration system.

V. COMPARISON BETWEEN VAPOUR COMPRESSIONS SYSTEM AND VAPOUR ABSORPTION SYSTEM FOR AUTOMOBILE AIR CONDITIONING-(1TR CAPACITY)

Distinguishing factor	vapour compression system	Vapour absorption system
Type of energy input	High grade energy(mechanical)	Low grade energy (Waste heat)
COP of system	2.56-3.9	0.32-0.52
Power of compressor/pump	1.5 KW	0.05 KW
Performance at part load	Poor	Poor Good
Chances of leakage	More	Less
Charging of Refrigerant\	Simple	Difficult
Size of system for given capacity	Less	More

Table: 5.1 comparisons between vapour compressions system and vapour absorption system.

VI. RESULTS AND DISCUSSION

The measured data as given in Table 4.2 is used to calculate mass flow rate of air, mass flow rate of fuel and maximum useful heat available in exhaust gas at 25% & 50% throttle opening position, considering speed of engine as running parameter. The Table 4.2 shows that heat available in exhaust when engine is running in idling condition is about 13.079 KJ/sec. Also useful heat available in exhaust gas varies 15.37 KJ/sec at 1750 rpm to 19.105 KJ/sec at 2500 rpm when throttle opening is 1/4th and with half throttle opening the amount of exhaust heat varies from 21.51 KJ/sec at 1750 rpm to 26.22 KJ/sec at 2500 rpm. This shows that useful heat is about 1.5 times when throttle position becomes half. In this way at full opening of throttle valve significant heat can be obtained from the exhaust gases. The absorption refrigeration system of one ton capacity consumes about 50 W of power to drive the circulation pump, which is much less than power is consumed by compressor of conventional air conditioning system of same capacity. For installing one ton air conditioning unit operating between 45^0C condenser temperature and 5^0C evaporator temperature the heat required for generator unit is about 10.1188 KJ/sec. The useful heat available in the exhaust gas during idling and 1/4th throttle opening of an engine is more than the heat required in the generator and able to run air conditioning unit.

VII. CONCLUSIONS

The engine exhaust gas was confirmed as a potential power source for absorption refrigeration systems.

Analysis shows that useful heat available in the exhaust gas is sufficient to generate ammonia vapour from ammonia solution for one ton air conditioning unit.

The domestic absorption refrigerator has low coefficient of performance and does not provide the cooling capacity needed for automotive application. However, a dedicated absorption refrigeration system may be able to take advantage of the exhaust gas power availability and provide the cooling capacity required for automotive air conditioning.

Introduction of the absorption refrigeration system in the engine exhaust system does not cause significant pressure drop in the exhaust flow. As the absorption refrigeration system consumes very less power to drive the circulation pump, the net output power for same capacity of an engine increased and specific fuel consumption was decreased.

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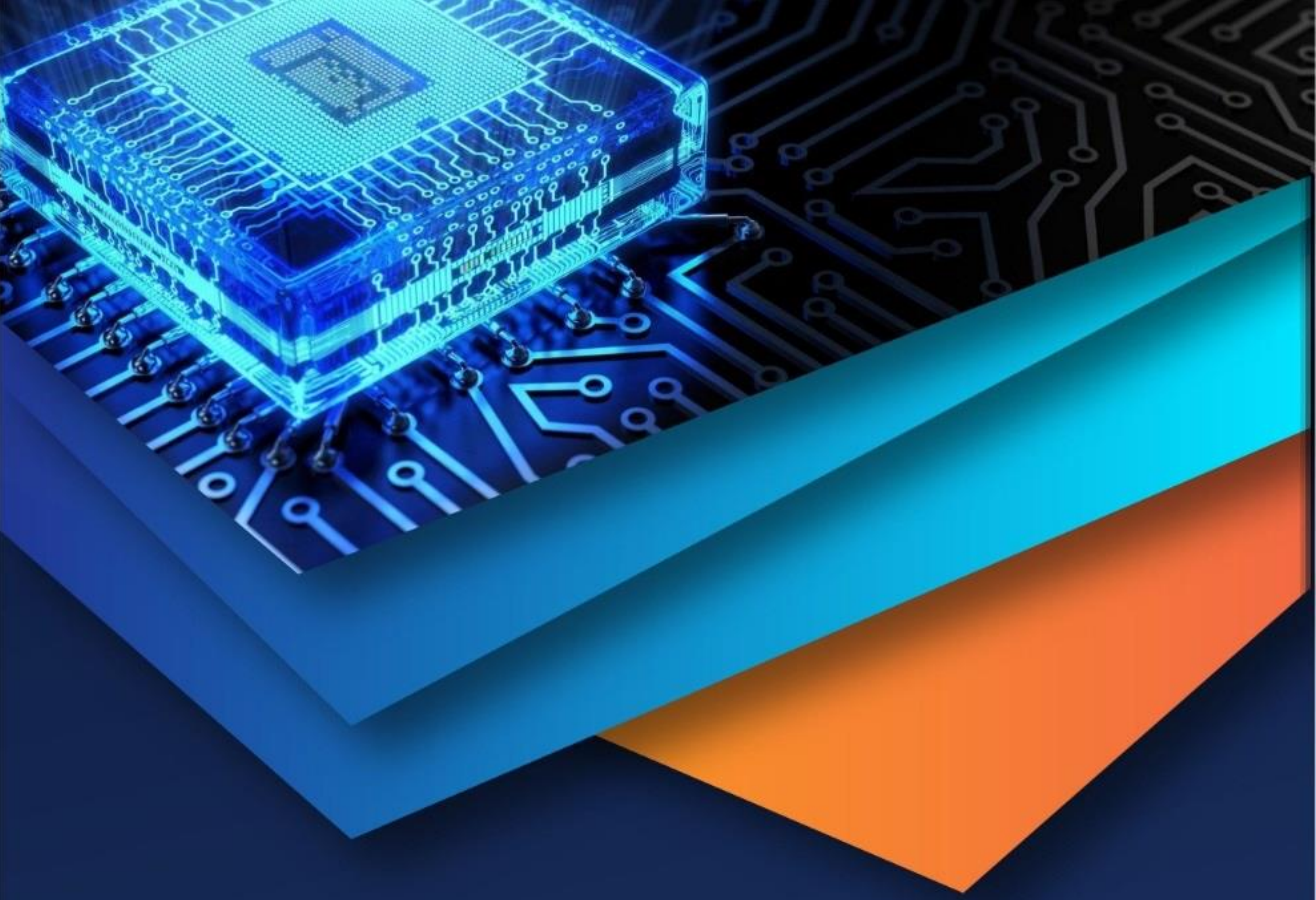

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