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Design & analysis of air conditioning system for off-road vehicle

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ABSTRACT

Automotive air-conditioning system has played an important role in human comfort and to some extent safety during vehicle driving in varied atmospheric conditions. Objective of this research is to design and develop an optimized air conditioning system for off road vehicle using R134a as a refrigerant. The purpose of this research is to meet requirements such as use the basic air conditioning cycle, effective use of airflow phenomena, to develop an air conditioning system at optimum quality, lowest cost and greatest customer satisfaction.
To design and develop air conditioning system for 5 passengers in vehicle with more packaging constraints, less space for air distribution system, condenser packaging constraints, pipe routine and each passenger should get better velocity of air by air distribution system. In this research 3D modeling of air conditioning system is carried out in CATIA V5, Ansys Fluent software package is used or 3D air flow analysis. Final test results are directly mapped on vehicle at stationary and cool down tests.

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I. INTRODUCTION

One can define the automotive air conditioning system which allows the safe operation of a vehicle in all kinds of weather by making a human body thermally comfortable and maintaining all glass areas free of ice on outside and fog or mist on inside. The purpose of this project is to meet the following objective.

- Use the basic air conditioning cycle
- Air flow phenomena
- Design. Development and manufacturing to produce a world class air conditioning system at optimum quality, lowest cost and greatest customer satisfaction.

A vehicle system performance is very subjective and consequently its design is customer comfort driven within given cost and timing constraints. The AC (Air Conditioning) panel outlet air flow direction, volume, velocity and temperature must be adjustable over a wide range of climatic and driving conditions. The system must be quiet, the controls easy to understand and operate. It must cool down vehicle quickly. The engineer while considering the customer also packages the compressor and drive, a condenser and plumbing of AC system in hot restricted space. The compressor must use minimum engine horsepower and be efficient over a wide range of RPM (Revolutions per Minute). The engineer has to negotiate styling request for aesthetic AC outlets and control appearance, NVH (Noise, Vibration and Harshness) request for minimum noise and vibration, engine cooling fan and shroud design, the aerodynamic and styling considerations of grille, engine engineering and emissions need for optimum engine idle RPM, etc. The engineer is also reminded to maintain timing schedules and to keep design, development and materials and labor costs at a competitive level while being frequently requested to change the AC (Air Conditioning) system during the early design stages of the vehicle to clear up real estate for other components. It has been observed that just a 5% cost investment in product design has a 70% cost influence on manufacturing productivity. Labor and material influence only 25% of the total cost. The largest influence on the development is at the lowest cost phase. The importance of good Air Conditioning product design and manufacturing assembly methods early in the vehicle development stage is of the highest priority. The effective implementation of new technology in automotive Air Conditioning systems requires a comprehensive understanding of Air Conditioning design and development principles. A successful system is one that adapts to the best configuration based on past application, state of the art design, new technology, experience and optimal design strategy.

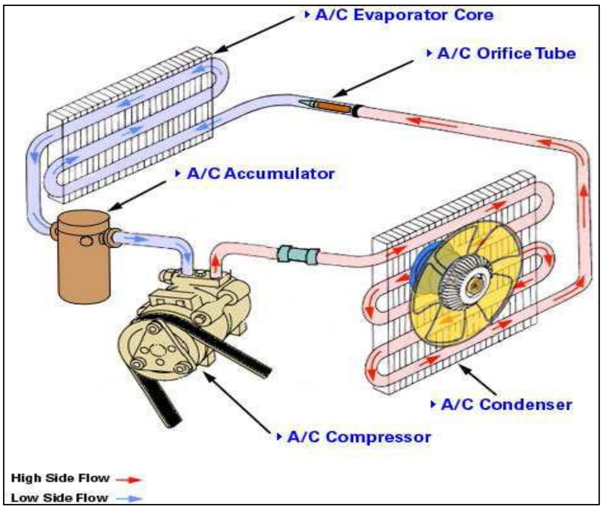

Fig. 1 Components & layout of air conditioning system

As mentioned above, the objective of this project is to design and develop an optimized Air Conditioning unit system for off-road application vehicle. In this project design and develop AC system for 5 passengers in vehicle with more packaging constraints, less space for air distribution system, condenser packaging constraints , long pipe routine , each passenger should get better velocity of air by air distribution system. Thus, this problem takes me as my project work. Fig. 1 shows layout of air conditioning system.

II. Cooling Load Calculations

Assumptions based on experience made are as follows,

- Number of passengers in cabin = 1 driver + 4 passengers
- Number of air changes per hour = 4
- Area and volume as required have been calculated.
- Temperature difference across firewall and floor panel is 10°C higher on account of engine and road heating.
- Vehicle heads west and solar load is calculated at 20°N latitude, 21st June at 4 p.m.in Pune.
- Outside film co-efficient = 34.1 W/m2°C
- Inside film co-efficient = 9.36 W/m2°C
- Thermal air conductance = 6.24 W/m2°C
- Total heat from driver / passenger = 0.22 kW / 0.1 kW
- Blower motor = 0.1 kW

A. Solar Load

To have maximum solar radiation to the exposed vehicle, it is aligned in East – West direction facing towards west side. Following data on 20°N latitude, 21st June at 4 p.m. in Pune has been recorded as,

TABLE I
HEAT LOAD THROUGH GLASS

Total sun load = 1.2 kW

The heat load through the various parts of the vehicle in the

Sr. No.	**Orientation Area**	**(m2)**	**(W/m2)**	**(kW)**
1	Wind shield (West)	2.09	505	1.055
2	Side glass (North)	0.70	38	0.0266
3	Side Glass(South)	0.70	105	0.0735
4	Rear Glass (East)	0.68	44	0.03

cabin is calculated and tabulated as shown in above table. For example: To calculate the heat transfer through Wind shield, we have,

U = 1 / (1/f0 + L/k + 1/fi)

=1/ (1/34.1+0.0035/0.744+1/9.36) =7.1 W/m2 °C

ΔT = Temperature difference between outdoor and indoor.

= 40 – 22 = 18 °C

Thus,

Q = m x Cp x ΔT

Q = 7.1 x 2.09 x 18

Q = 0.267 kW

Total heat load from all parts of the vehicle = 1.13 kW

B. Casual Gains

The heat added into the compartment through infiltration of air, passengers and light is calculated under casual gains. For these calculations, refer psychometric chart for enthalpy values.

Sensible Heat Load:

Q infiltration = Sensible heat through infiltrated air

= Cabin volume x Air changes / hour x Air density x DH

= 3.25 x 4 x 1.2 x (44 – 20)

= 0.104 kW

Q passenger = Total sensible heat from passengers

= Number of passengers x sensible heat from passenger

= 5 x 0.09

= 0.45 kW

Q light = Sensible heat from light

= 0.001 kW

Sensible load = 0.104 + 0.45 + 0.001

= 0.555 kW

Adding 20 % = 0.111 kW

Total sensible heat load = 0.66 kW

C. Latent Heat Load

Q infiltration = Latent heat through infiltrated air

= Cabin volume x Air changes / hour x Air density x Δh

= 3.25 x 4 x 1.2 x (54 – 44)

= 0.043 kW

Q passenger = Total latent heat from passenger

= Number of passenger x latent heat from passenger

= 5 x 0.06

= 0.3 kW

Latent load = 0.043 + 0.3

= 0.343 kW

Adding 20% = 0.0686 kW

Total latent heat load = 0.41 kW

Thus,

Net Casual heat gain = Sensible heat load + latent heat load
= 0.66 + 0.41
= 1.07 kW
Heat added by blower = 0.1 kW
Total Cooling Load = 1.13 + 1.2 + 1.07 + 0.1 = 3.678 kW
=1.0 Ton

It means we need to size the system, in such a way that it should balance the required cooling demand under a given time as specified by customer. In the next chapter we will discuss about the sizing of each component and heat balance of a system

III.HVAC DESIGN

Q (condenser) = Q (evaporator) + W (compressor work)
Specify Conditions
Refrigerant = R134a
T ambient = 40 °C DBT, 25 °C WBT
T evaporator out, air = 6 °C
T sub-cooling = 10 °C
T superheating = 8 °C
P discharge = 18 bar
P suction = 3 bar
Air quantity = 8.0 kg/min @ 350Pa
T condenser inlet, air = 40 °C
T cabin target, air = 22 °C
A. Evaporator Duty Determination
The evaporator duty consists of dry air sensible heat and the sensible and latent heat of condensation of moisture in the air. The sensible and latent heat is absorbed by the refrigerant flowing through the evaporator. The dry air sensible heat is determined from,
Q = m Cp DT
As our consideration in the 100% recirculation mode, the average temperature assumed at the evaporator inlet is 30°C. The air temperature differential across the evaporator, is then,
$\Delta T = 30 - 6 = 24$ °C
The value of 'm', the amount of air flowing across the evaporator, is determined from the body leakage curve or can be approximated if the curves are not available. In our case, the best value approximated by experiments is, 8.0 kg/min as a starting value for mid-size passenger car against the static pressure of 350 Pa. The dry air sensible heat absorbed by the refrigerant then, from equation
Q, sensible = 8.0 x 1.0 x 24
Q, sensible = 3.2 kW
Since evaporator also condenses the water vapor in the air, the latent heat must be added to required evaporator duty. The latent heat of condensation of moisture in the air and amount of condensate is determined using psychrometric chart *(See Fig 2).* The latent heat absorbed by the refrigerant in the evaporator is;

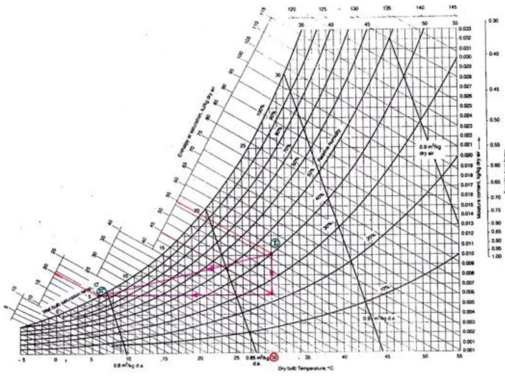

Fig. 2 Psychrometric Chart

Q, latent = m x (ΔH) = 8 x (54 – 44)
Q, latent = 1.33 kW
The total heat absorbed by the refrigerant is then,
Q, Total = Q, sensible + Q, latent
= (3.2 + 1.33) kW
Q, Total = 4.53 kW
B. Refrigerant Flow Determination
The refrigerant flow is determined from,
*Q = M (*DH*)*
Where,
Q (kW), Evaporator duty
M (kg/hr), Refrigerant flow
DH (kJ/kg), Change in enthalpy of the refrigerant entering and leaving the evaporator.
The values of the corresponding thermodynamic properties will get from p-h diagram. The refrigerant flow from equation becomes,
m = Q/Δh = 4.53 / (405-278)
m = 128.4 kg/hr
C. Determination of Compressor Requirement
Four basic parameters required to determine the air conditioning compressor are;
• Volumetric efficiency
• Brake horsepower
• Refrigerating capacity
• Compressor Pulley Ratio
Too high a pulley ratio will result in excessive horsepower drawn at high RPM and may require a compressor interrupt to protect for compressor durability, too low a pulley ratio will not meet refrigerant flow demand at low engine idles. *Volumetric efficiency*, it is used to describe how efficiently the refrigerant gas is drawn into the compressor. In our case, from the supplier's performance curve,
Volumetric efficiency = Intake volume / displacement = 60% @ 2800 compressor RPM
Thus from the p-h chart,
Specific volume, suction (V) = 0.071 m³/kg
Entropy, suction (S) = 1.75 kJ/kg .K
Enthalpy, suction (H) = 405 kJ/kg
Now,
Compressor volume displacement, cc/min = mass flow rate x specific volume
= (128.4 x 0.071 x 106) / 60
= 1, 51, 940 cc/min
Compressor displacement per revolution needed would be,
= 1, 51, 940 / 2800
= 54.26 cc/rev
Thus the actual displacement needed = 54.26 / 0.6

= 90.44 cc ≈ 110 cc

The compressor selected is therefore 110 cc which is nearest size in next available range.

Now we will calculate refrigeration capacity work and compression ratio. Further more, refrigeration effect and Co-efficient Of Performance (COP).

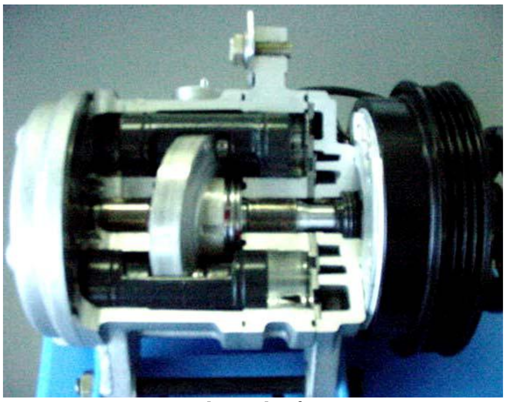

Fig. 3 Constructional Details of 110 cc Compressor

D. Compressor Bracket Design

Factors to be considered while designing compressor mounting bracket are as follows:

1. Compressor weight
2. Compressor center of gravity
3. Compressor moment of inertia
4. Compressor belt duty cycle
5. Type of belt tensioner

Steps of compressor mounting bracket design:

1. Resultant is calculated with the help of crankshaft pulley torque (T1), compressor pulley torque (T2), centre of gravity of compressor, jerk load
2. From this resultant bracket thickness is calculated. Also thickness is changes according to bracket material.
3. CAE, FEA, NVH analysis is carried out on bracket, after completion of bracket design.
4. Compressor mounting bracket is tested by two methods.

i) Torture track cycles

ii) Vibration test rig

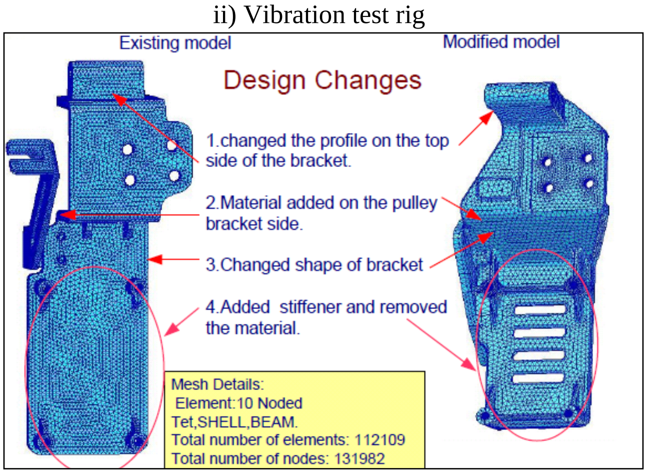

Fig. 4 Existing & Modifies model of bracket

Fig. 4 shows the design changes in existing bracket model. There are 4 changes are made in the bracket are as follows:

1. Changed the profile on the top side of the bracket.
2. Material added on the pulley bracket side.
3. Changed the shape of the bracket.
4. Added stiffener & removed the material.

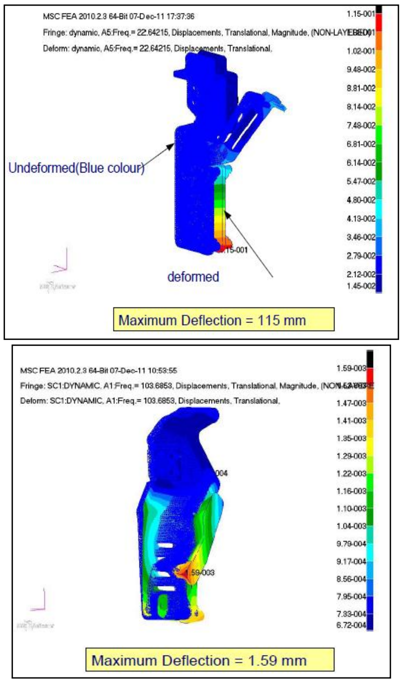

Fig. 5 CAE of Existing & modified model for maximum deformation

Fig. 5 shows the CAE of Existing & modified model for maximum deformation. In existing model maximum deflection occurs is 115 mm. In modified model maximum deflection occurs is 1.59 mm. New modified model is having better result.

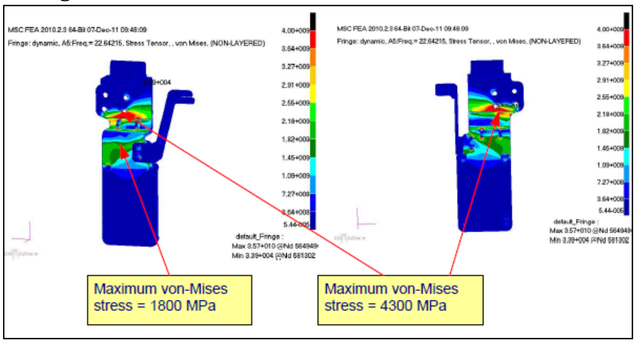

Fig. 6 CAE of Existing model for maximum stress

Fig. 6 shows the CAE of Existing model for maximum Von-Mises stress on front side is 1800 MPa & at rear side is 4300 MPa

Fig. 7 shows the CAE of modified model for maximum Von-Mises stress on front side is 320 MPa & at rear side is 380 MPa

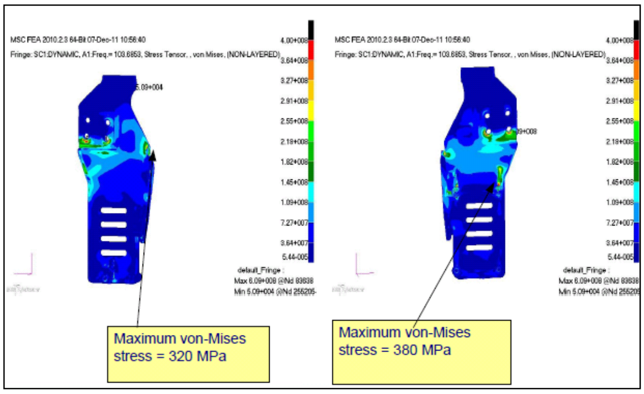

Fig. 7 CAE of modified model for maximum stress

E. Determination of Condenser Duty

The required condenser duty is determined from the basic system balance equation,

Q (condenser) = Q (evaporator) + Q (compressor)

= 4.53 + 1.5 = 6.03 kW

The bench mark speed is 60 km/hr, at which the ram air speed gets drop down and the total air flow available at the condenser inlet is 0.645 kg/s.

Thus, the air side heat transfer rate is calculated as,

Q, air side = m x Cp x (T air, outlet – T air, inlet) = 0.645 x 1 x (52 – 40)

Q = 7.74 kW

The calculated heat transfer under the given condition satisfies the basic system balance equation. Now, effectiveness of the condenser is calculated as,

Effectiveness (ε) = (T air, outlet – T air, inlet) / (T sat@18 bar – T air, inlet)

= (52 – 40) / (64 – 40)

Effectiveness (ε) = 0.5

The temperature of the high temperature, high pressure gas at the compressor discharge, will be, 78°C. The saturation temperature at discharge will be, 64 °C. The degrees of superheat in the discharge gas is then,

Degrees of superheat = 78 °C – 64 °C = 14°C

F. TXV Capacity Determination

The system design capacity i.e. evaporator load is calculated as 4.53 kW (1.28 ton).

Furthermore, to determine exact valve capacity, we need to consider the total pressure drop across the valve considering all the friction losses in plumbing. Also, we need to consider the liquid refrigerant correction factor which depends upon the temperature of refrigerant entering TXV. Thus, from the standard table we have,

TXV Rating (Evaporator Duty) = 1.21 ≈ 1.28

Liquid refrigeration correction factor = 1.42

Pressure drop correction factor = 1.15

TXV Capacity = TXV Rating x Liquid refrigeration correction factor x Pressure drop correction factor

= {1.21 x 1.42 x 1.15} Tons of Refrigeration

TXV Capacity = 1.97 ≈ 2.0 Ton

G. Blower & ducting

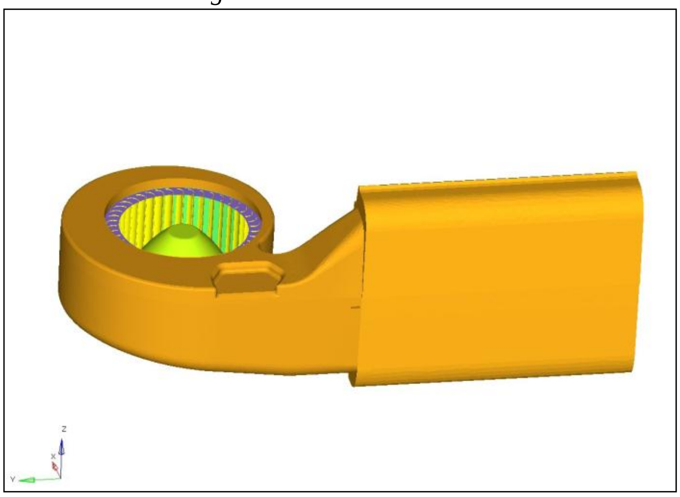

Fig. 8 Blower unit

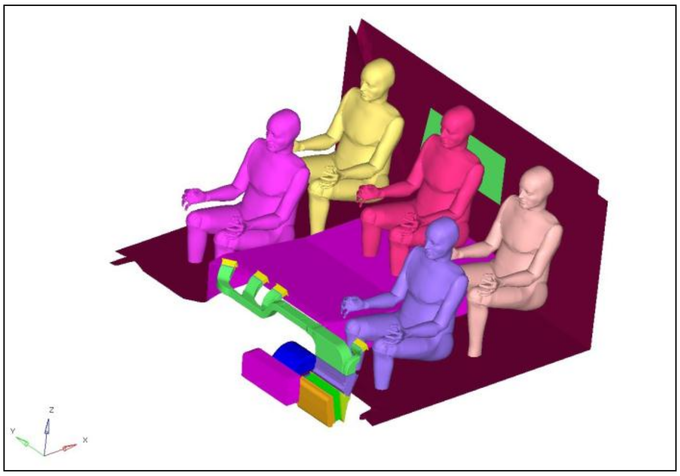

Fig. 9 seating positions

Fig. 8 shows the designed blower unit & Fig. 9 shows the cabin seating positions & ducting of the vehicle.

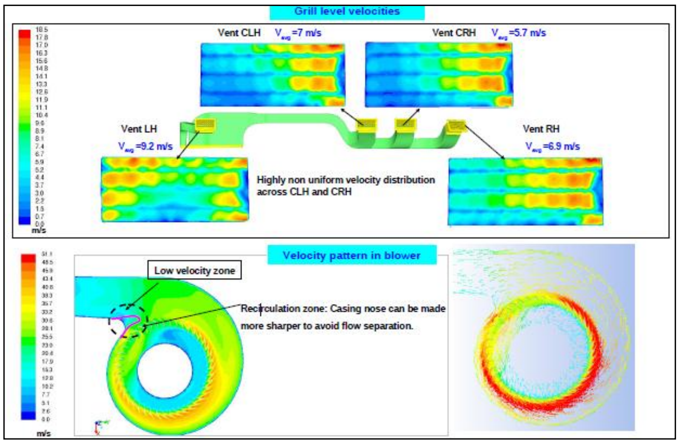

Fig. 10 CFD analysis of blower & ducting

Fig. 10 shows the CFD analysis of blower, ducting & grills. We can see the velocity pattern in the blower, in ducting & in grills.

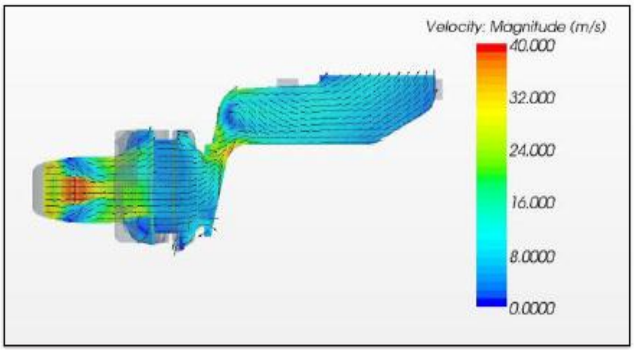

Fig. 11 CFD analysis of AC unit

Fig. 11 shows the CFD analysis of AC unit. We get the velocity patterns at various locations in the unit.

The pressure drop is also measured in the unit housing at recirculation mode. Fan plane after blower to evaporator in is 149 Pa, evaporator in to evaporator out is 229 Pa & evaporator out to duct out is 316 Pa. Total pressure drop in the distribution housing is 694 Pa

After system design & packaging various tests were completed on the vehicle. Test conducted on vehicle are as follows:

IV. TEST & RESULT

1. EBHS
2. Airflow measurement
3. Velocity scanning
4. Charge optimization test
5. ODCD test
6. Cool down test at moderate condition
7. Cool down test at severe condition
8. Anti-icing test

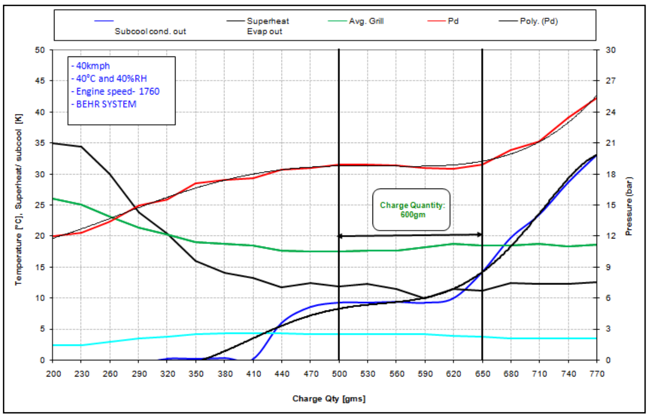

Fig.12 Result of charge optimization test in graphical form

Fig. 12 shows the result of charge optimization test in graphical form. From this test charge quantity of the system is decided. The test is conducted at 40°C temperature & at 40% RH. From the charge optimization test the charge quantity came is 600 grams.

After that with 600 grams charge quantity the further cool down tests are carried out & completed successfully with meeting required target temperatures. See Fig. 13 shows the result of moderate cool down test & Fig. 14 shows the results of severe cool down test.

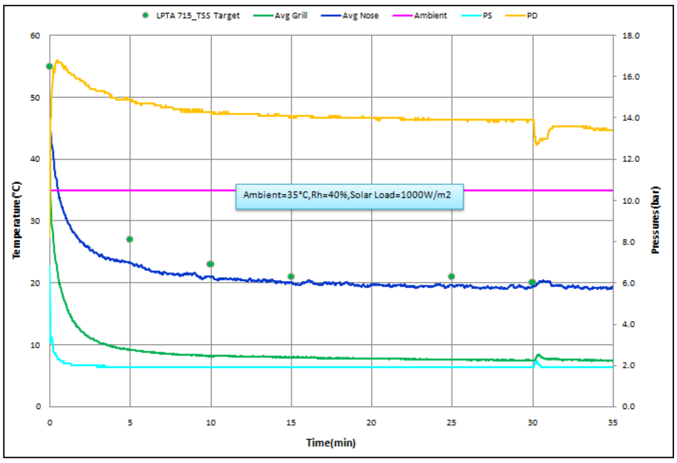

Fig. 13 Result of moderate cool down test in graphical form

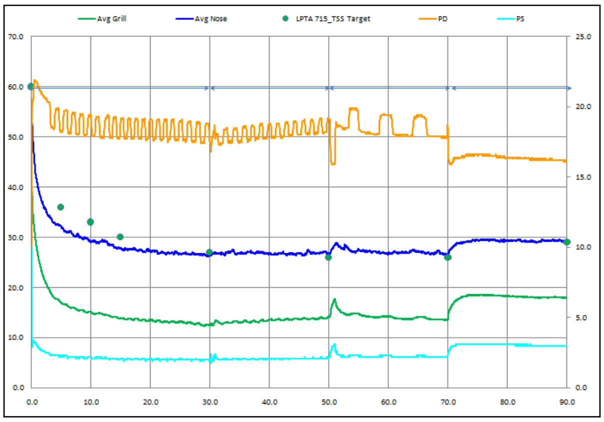

Fig. 14 Result of severe cool down test in graphical form

V. CONCLUSION

Air conditioning performance meeting targets in moderate & in severe test conditions with margin.

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REFERENCES

[1] R.K.Shah, BSME-ASME International conference on Thermal Engineering, Dhaka, Bangladesh, 2006.
[2] Arsha Viswambharam, Sheetal Kumar Patidar, Khyati Saxena, International Journal of Scientific & Research Publications,volume 4,Issue 4,April 2014.
[3] Toshiba Kondo, Akiva Katayama, Hideki Scetake, Masatoshi Morishita,Mitsubishi Heavy Industries Technical review,Volume 4,June 2011.
[4] Md. Shahid Imam, Dr.M.Sameer Basha, Dr.Md.Azizuddin, Dr.K.Vijay Kumar Reddy,International Journal Innovative Research in Science,Engineering & Technology, volume 2, Dec.2013.
[5] G.Victos, J.Gryzagoridis, S.Wang, Department of Mechanical Engineering,Capetown,2012