HFC/HC BLEND FOR CAR CLIMATE CONTROL

WITH MINERAL OIL AS LUBRICANT

[1]RAVIKUMAR T.S.*   [2]MOHAN LAL D

1. *Department of Mechanical Engineering, Hindustan Institute of Technology & Science, Hindustan University, Chennai, India.*

2. *Department of Mechanical Engineering, College of Engineering, Anna University, Chennai 600 025, India.*

ABSTRACT:

Polyalkaline glycol is used as the lubricant with R134a in automobile air conditioning systems and it is well known that PAG oil is highly hygroscopic in nature. Presence of moisture in the system may detoriate the system and its performance. Hence PAG oil is replaced with conventional mineral oil. In the present study, a new refrigerant blend R134a/R290/R600a with mineral oil as lubricant has been tested in an experimental test rig with R12 as base line and compared with the published results. Pull down test from an average cabin temperature of 50°C is performed while the compressor speed is varied successively from 1500-2250-3000-800 rpm with a constant heat load of 2500 W, and held at that speed for duration of 30 minutes. Both R12 and the HFC-HC blend is tested in an experimental test rig and it has been found that this new mixture is almost in par with R12, However, a slight loss in system performance is encountered. But, considering the service issues associated with R134a and PAG oil, this new mixture can have better performance than R134a and PAG combination.

Keywords*: Alternate refrigerants, Automobile air conditioning, Oil miscibility, Retrofitting, R134a/R290/R600a mixture, Refrigerant Blend.*

* Corresponding Author

1. INTRODUCTION

An automobile air conditioning system consists of evaporator, compressor, condenser, receiver-drier and expansion valve. The purpose of the receiver is to store and supply the excess refrigerant during varying load conditions and also it ensures only liquid take off to expansion valve. The continuous operation of the vehicle will make the cabin temperature to drop gradually. When the temperature has reached the set point the compressor clutch disengages from the engine drive and once it reaches the cut-in point the compressor gets engaged and restores the A/C operation. The automobile climate control system must be designed carefully to satisfy the wide range of operating conditions at varying speeds.

Because of the high vapor pressure for R134a and HC blend the resulting vapor pressure for the blend is expected to be more than that of R12. Muir[1] reported that R134a would consume 10 to 15% more power than that of R12 systems. Jung and Radermacher [2] had studied the performance of pure and mixed refrigerants in a single evaporator domestic refrigerator and reported that the COP of R134a is 3% less than that of R12. Camporese *et al*. [3] have reported that the refrigerating capacity and COP of the retrofitted R134a system are 5 to 8% less than that of R12 system. R134a and R600a (80:20) by mass have been experimentally investigated in a heat pump and 5% improvement in system COP was observed. Jung *et al*. [5] reported that R290and R600a mixture with 0.55 to 0.66 mass percentage of R600a could yield a 3 to 4% rise in energy efficiency when compared to R12. Sunami [6] has reported that suitable additives could improve the miscibility of R134a with mineral oil. Tashtoush *et al.* [7] found that butane/propane/R134a in the proportion of 25/25/30 by mass, without changing the lubricating oil (mineral oil) or condenser, the COP of the mixture is 5.4% less than that of R12.

Sekar and Mohanlal [8,9] has performed extensive research with R134a/R290/R600a refrigerant mixture as a replacement for R12 in deep freezers, walk-in coolers and refrigerators and it has been observed that the system COP increases by 3 to 15% and energy consumption is reduced by 5 – 15%. Refrigerant properties are the key factors for the evaluation of refrigerants and refrigerant mixtures. Ravikumar and Mohanlal [10] has tested the proposed mixture in a passenger car on road in the true running conditions with Original Equipment Manufacturer (OEM) components and reported that the cabin temperature for M09 is slightly higher than that of R12. REFPROP V7.1 [11] is used to calculate the thermodynamic properties of refrigerants and refrigerant mixtures, issued by National Institute of Standards and Technology. Continuous research is going on, in order to find ozone friendly, energy efficient and environmental benign refrigerant for air conditioning applications. Kiatsiriroat *et al.* [12] have analyzed the performance of R22/R124/R152a refrigerant mixture in an automobile air conditioning system with water streams at different temperature. COP is high when the composition of R22 is in the range of 20-30%. Joudi [13] has done experimental and computational performance analysis for both R12 and HC blend (HC blend is a mixture of propane 'R290' and isobutane 'R600a') in the proportion of 20/40/60/80 by mass, in which, the COP of the blend is equal to or more than that of R12. Struss *et al.* [14] had conducted experiments with vehicular wind tunnel to assess the effect of charge quantity and system performance with both parallel flow and serpentine condensers using R12 and R134a. The test results show higher head pressure with R134a, similar cabin temperature for R12 and R134a and fine-tuning of TXV can improve interior performance. Hirata *et al.* [15] compared the performance of R12 and R134a on various functional components, in which, the results show high discharge pressure and equal cooling capacity but under idling and high thermal load conditions R134a performance is inferior to R12. Dekieva *et al.* [16] analyzed the vehicles, after two years of run, for durability issues and system chemistries. OEM hoses provide sufficient containment for R134a in retrofit situations, and adequate compressor durability may be expected with the lubricants used. Herbert and Mohan Lal [17] have analyzed the effect of refrigerant charge quantity in a window air conditioner with mineral oil as lubricant. Addition of HC up to 20% takes care of mineral oil miscibility issues.

PAG oil is highly hygroscopic and during service, exposing the PAG oil to atmosphere for a considerable period of time may lead to excessive moisture ingress, which may detoriate the system and reduces the system performance. Hence, the conventional mineral oil that has no service issues can be used as the lubricant. As the mineral oil is not miscible with R134a, 9% of HC blend (45.8% R600a & 54.2%R290) by mass is added with 91% of R134a by mass. (This mixture henceforth called M09 meaning a mixture with 9% HC blend) The presence of 9% of hydrocarbon ensures proper oil return to the compressor and because of its better heat transfer characteristics; there is a rise in Refrigerating capacity. The percentage of HC is well within the flammable limits. In the above-cited literatures it can be seen that, R134a and mineral oil combination has not yet been tested in an automotive application. The same has been tested and compared with the results obtained from R12 and the already published results.

2. EXPERIMENTAL TEST RIG

An experimental test rig has been constructed to simulate the actual car climate control system. The schematic diagram of the experimental setup is shown in figure 1. The compressor is coupled to a belt driven electric motor. The motor is powered through a variable frequency drive (VFD). The speed of the motor is varied, by varying the pulse width of the VFD to simulate varying speed of the vehicle. The compressor and the motor speed are measured by a non-contact type optical tachometer with an accuracy of ±l rpm. Temperature sensors PT-100 RTD's with an accuracy of ± 0.25°C are fixed at 29 points as shown in the figure 1.

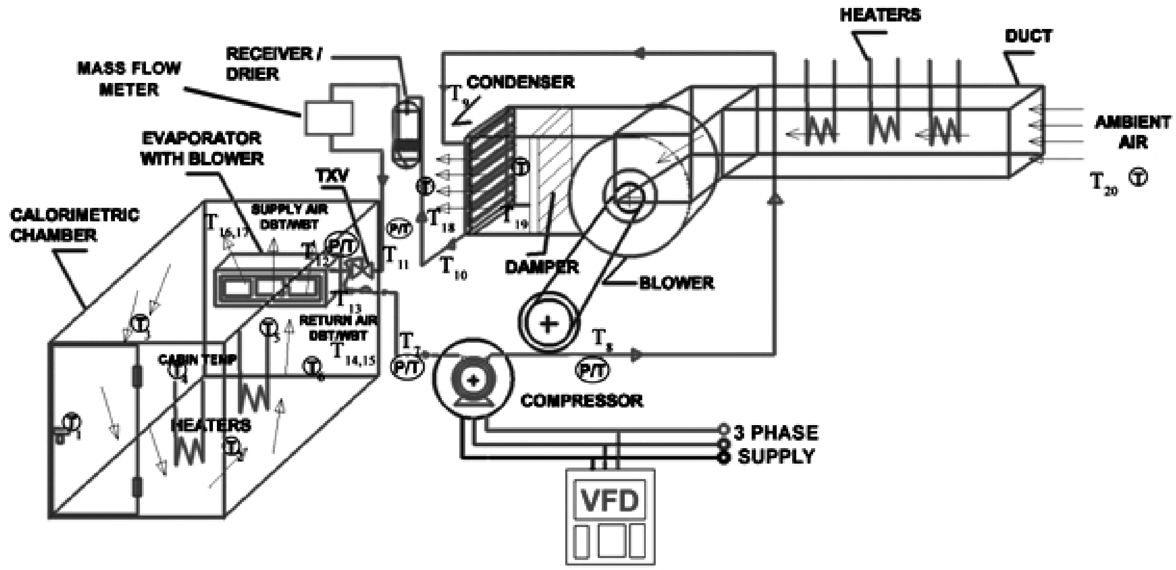
Six RTDs are connected to measure the internal cabin temperature. Seven RTDs are connected to measure the temperatures at the inlet and outlet of the four major components. Four RTDs at the evaporator air inlet and outlet to measure the dry bulb and wet bulb temperatures. Two RTDs to measure the condenser air inlet and outlet temperature. Nine RTDs to measure the temperature glide in the evaporator and one RTD to measure the ambient temperature.

An environmental chamber is constructed to represent the passenger compartment. A 1.5 tonne evaporator with 3 blower speeds, providing an average maximum velocity of 3 m/s and a maximum volume flow rate of 500 m^3/h is located inside the chamber. The chamber is provided with 6kW heater to simulate varying heating loads. The RTDs are located at various heights inside the cabin to ensure uniform air temperature with an average deviation of ±0.5°C. All the RTDs are connected to a 60-channel data logger, with a scanning interval of one second. The data logger is connected to a personal computer to store and analyze data.

The condenser is provided with a blower powered by a 3 HP motor running at 1500 rpm providing an average air velocity of 5 m/s and an air volume flow rate of 2500 m^3/h. The condenser duct is provided with a 9 kW heater. The heater load is varied by a variac to simulate varying ambient conditions. A water atomizer is provided to maintain constant RH value of the cabin.

All the power measurements are measured directly by using wattmeter. Four pressure gauges with an accuracy of ±0.01 bar are connected to measure the pressure across evaporator and condenser. A corriolis type mass flow meter with an accuracy of ±0.25% is used to measure the mass flow rate of the refrigerant at the outlet of the condenser after the receiver drier unit, which ensures, only liquid enters the mass flow meter.

All the components are inter-connected by flexible rubber hoses. The setup is charged with R12 and M09 refrigerants and the various operating parameters that influence the system performance are analyzed.



P – Pressure measurement, **T –** Temperature measurement

Figure 1: Experimental setup for Vehicle climate control

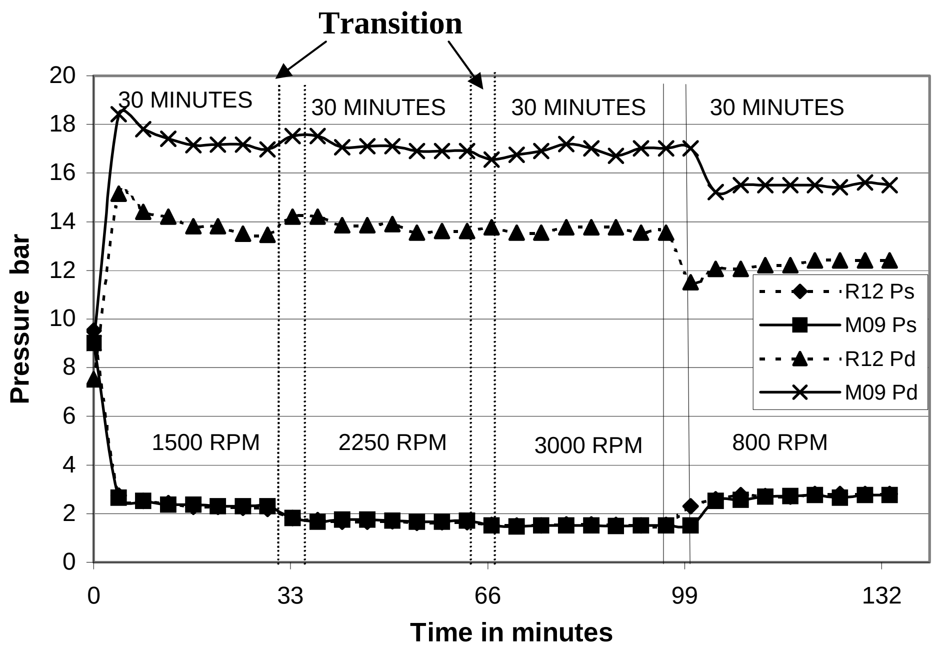


Figure 2: Variation in suction and discharge pressure at various rotational speeds

3. TEST PROCEDURE

Initially the cabin is heated to a temperature of 50°C by using electric heaters and maintained for one hour to simulate soaking in hot sun. The test conditions consist of four speeds (1500-2250-3000-800), the duration of each speed is 30 minutes. The test conditions are based on published literature [13] [14] [16]. The evaporator blower is set to the maximum volume flow rate of 500 m^3/h, The condenser velocity is set to 5 m/s, ambient temperature is set to 38°C by varying the heater resistance in the condenser duct. The cabin heaters are tuned to provide a constant heat load of 2500W. An air-circulating fan is provided in the cabin to ensure uniform air distribution. Once the required conditions are met the compressor is switched on for pull down operation.

4. RESULTS AND DISCUSSIONS

Figure 2 to 9 shows the effect of operating parameters on the vehicle climate control system. The variation in suction and discharge pressure for various speeds with respect to time is shown in Figure 2. As the speed is increased, rate of refrigerant that is removed from the evaporator is more, which reduces the evaporator pressure. The refrigerant that is transferred from evaporator gets accumulated in the condenser, increasing the condenser duty; this shoots the condenser pressure. The suction pressure for M09 is 5% more and the rise in discharge pressure is 20 – 25% more when compared to R12. The rise in discharge pressure will increase the work of compression.

The variation in compressor suction and discharge temperature with respect to time is shown in Figure 3. The suction temperature for both the refrigerants is more or less the same, whereas the discharge temperature for M09 is less than that of R12 by 2 to 10ºC. This is attributed due to the lower index of compression for the mixture. Since the heat capacity of M09 is higher than that of R12, the suction superheat rise and temperature rise due to work of compression, is less for M09 than that of R12.

The variation in average cabin temperature for various speeds with respect to time is depicted in Figure 4. The average cabin temperature is 1.5 to 2°C more than that of R12. The variation in mass flow rate with respect to time for the four speeds is shown in the Figure 5. When the compressor is switched ON the mass flow rate rises as high as 65g/s initially. But, it reduces to 20g/s in a very short period of 3-5 minutes. This is because the cabin temperature has come down from 50°C to 25°C. As the system approaches balancing conditions the mass flow rate reduces gradually and becomes almost constant during steady state. At the end of 30 minutes as the speed increases the mass flow rate also increases due to increase in compression rate. The mass flow rate at idling condition reduces drastically due to the reduction in pumping power. This shoots up the cabin temperature, return air temperature and supply air temperature. The mass flow rate for M09 is 8 to 15 % less than that of R12. The compressor shell temperature for M09 and R12 with respect to time at various speeds is shown in Figure 6. The compressor shell temperature for M09 is considerably less (10%) than that of R12 because of lower compressor discharge temperature, which will enhance the life of compressor and the lubricant.

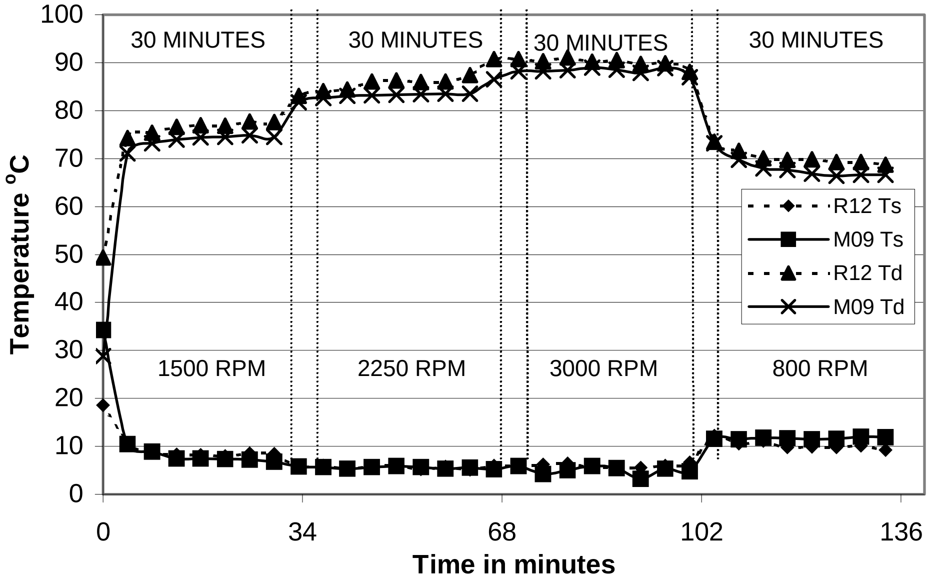


Figure 3: Variation in suction and discharge temperature at various rotational speeds

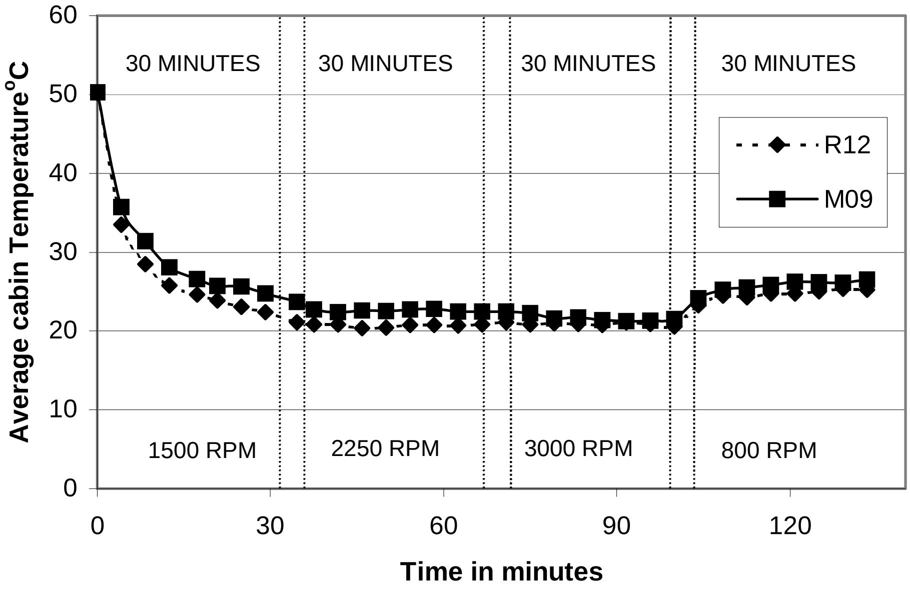


Figure 4: Variation in average cabin temperature at various rotational speeds

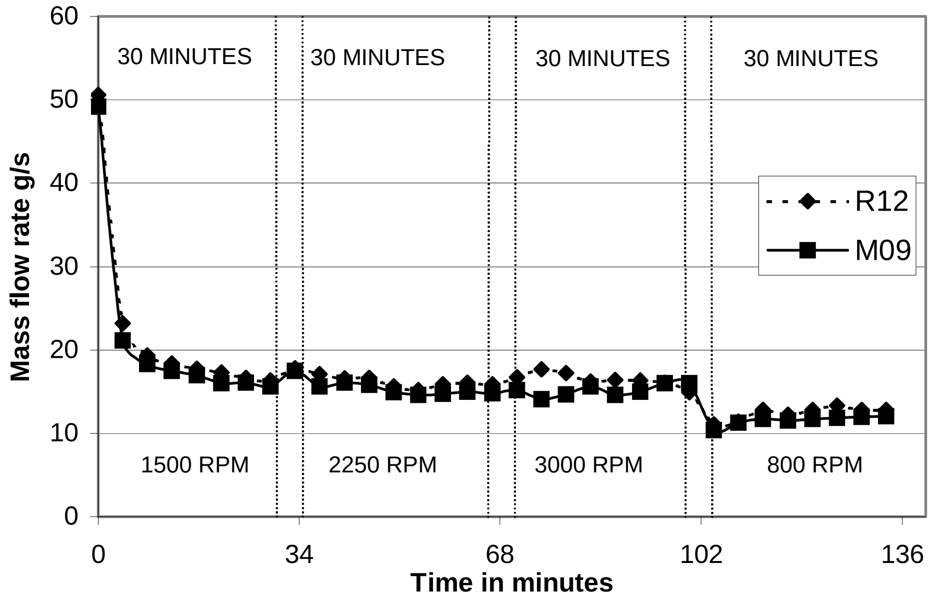


Figure 5: Variation in mass flow rate at various rotational speeds

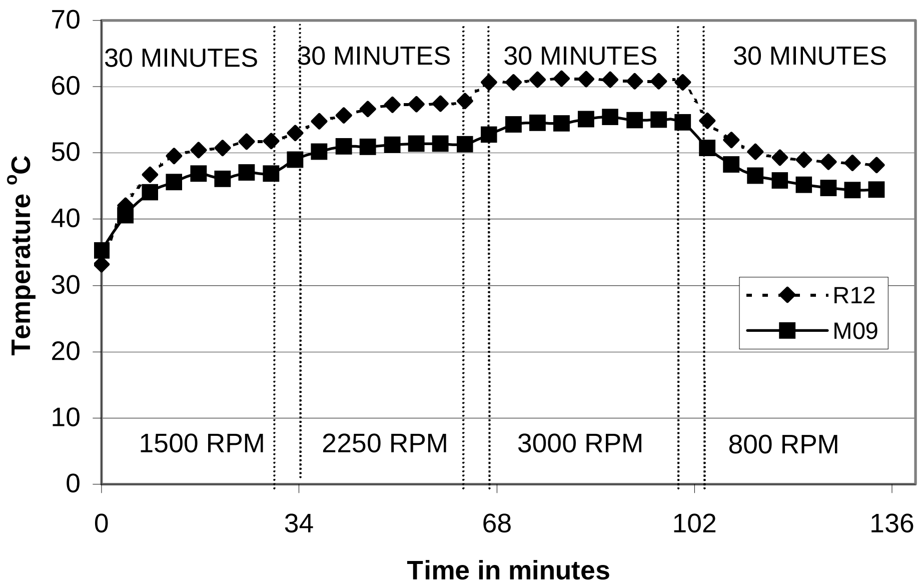


Figure 6: Variation in compressor shell temperature at various rotational speeds

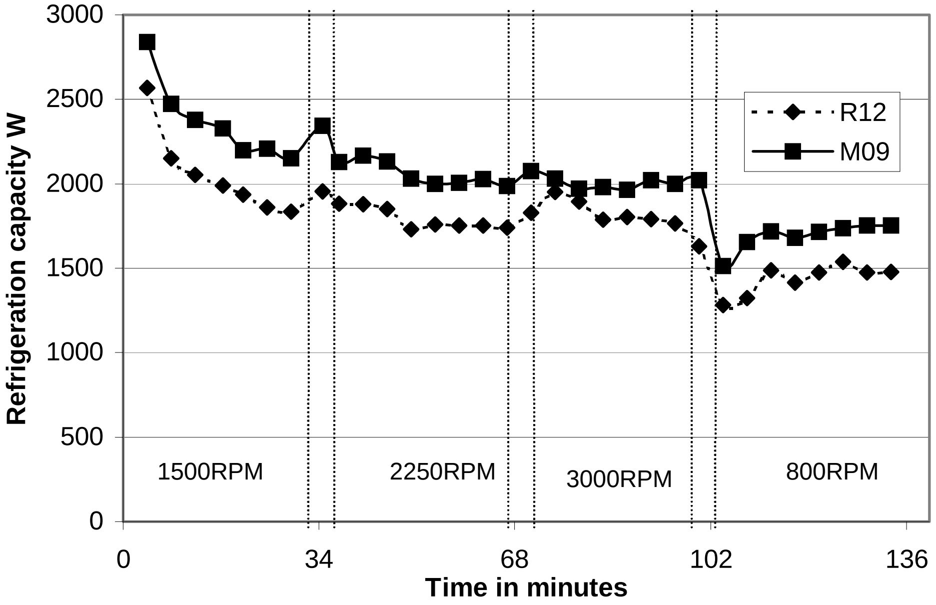


Figure 7: Variation in refrigeration capacity at various rotational speeds

Figure 8: Variation in compressor power at various rotational speeds

Figure 9: Variation in COP at various rotational speeds

The variation in refrigeration capacity with respect to time for the four speeds is shown in Figure 7. The refrigeration capacity for M09 is 15% more than that of R12 because of its high heat carrying capacity. Though the speed increases from 1500-3000 rpm the refrigeration effect is uniform since the load is uniformly maintained inside the test cabin. As the speed is reduced to idling speed only a lesser refrigerant is transported which reduces the refrigeration effect. The reduction in heat pumping makes the heat to get accumulated in the cabin, which results in a rise in cabin temperature.

The variation of power consumption at various speeds with respect to time is shown in Figure 8. The power is directly proportional to speed. As the speed increases and decreases the power requirement increases and decreases respectively. Pull down is carried out from 50°C at a constant heat input of 2500W. The observation during the first 30 minutes show the power requirement is more during the first 5 minutes due to high-accumulated heat content in the cabin as a result of soaking. As pull down continues gradually the power requirements decreases with decrease in cabin temperature. At the end of 30 minutes as the speed increases from 1500 to 2250 rpm there is a rapid rise in power requirements due to increase in compressor work. As the system gets balanced the power consumption reduces gradually. The same pattern is repeated during the rise in speed from 2250 rpm to 3000 rpm. As the speed is reduced to idling speed there is steep fall in the graph, which show a drastic reduction in compressor power. The power consumption for M09 is 22% higher than that of R12.

The variation in COP at various rotational speeds is shown in the Figure 9. The compressor work increases drastically with increase in speed due to rise in pressure, temperature and increased work of compression. The rise in compressor work for M09 reduces the average COP by 5 to 15%. The uncertainty in COP is found to be 4% (COP = 2.19 ± 0.09).

The test results are compared with the published results Struss *et al.* [14] in which the average cabin temperature for R12 at a speed of 89 km/h is 20°C and for R134a it was around 22°C. The rise in cabin temperature is 2 to 3°C. The evaporator suction and discharge pressure for R12 and R134a show similar trends. The results obtained in Hirata [15], in which R12 and

R134a are compared, agree well with the present work and ensures the superior performance of M09 when compared with R134a.

5. CONCLUSION

Refrigerants R12 and M09 are tested under similar conditions for 4 speeds. The average cabin temperature for M09 is 2°C more than that of R12, which show superior cooling capability of R12. There is no appreciable rise in evaporator pressure. However, the rise in discharge pressure for the M09 mixture is more than that of R12, which increases the work of compression. The reduction in discharge temperature for M09 improves the compressor life. The discharge pressure for the mixture can be reduced by improving the condenser performance or by increasing the airflow rate. Because of the increased heat carrying capacity of the M09 the refrigeration capacity is enhanced. Also, the drastic rise in compressor work reduces the system COP by 5 to 15%.

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Affiliations:

Dr. T. S. Ravikumar

Professor ,

Dept. of Mechanical Engineering

Hindustan Institute of Technology and Science

Hindustan University. Chennai, India.

Email: ssstsrk@yahoo.com

Mobile: +91 9840031657

Dr. D. Mohan Lal

Professor and Head,

Refrigeration and air conditioning division,

Dept. of Mechanical Engineering

College of Engineering, Guindy,

Anna University.