Experimental investigation on split type window Air conditioner using HFC and HC mixture as ecofriendly refrigerant alternate to HCFC-22

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Abstract. This paper reports on an experimental study evaluating a window air conditioner's performance when R152a/R290/R600/R600a refrigerant mixture (by various mass percentages) is used as a possible alternative to R-22. Alternative refrigerants are local, cheap, and environmentally friendly. Unmodified R-22 window AC was charged with R152a/R290/R600/R600a. During the experiment, parameters such as coefficient of performance, refrigeration effect, pressure ratio, compressor discharge temperature, refrigerant mass flow rate, and compressor power are analyzed. The present work shows that R152a-15%/R290-15%/R600-35%/R600a-35% is the best-performing alternative refrigerant. Proposed blends and R22's performance were compared.

1 Introduction

In the current scenario, as chlorine contributes to ozone depletion, refrigeration and air-conditioning industries are switching to chlorine-free refrigerants. Developed countries must phase out HCFC 22 by 2030 and developing countries by 2040.To solve the ill-effects of traditional refrigerants like ozone layer depletion, researchers around the world are identifying alternative refrigerant mixtures for window Air conditioner. Research journals list many R-22 alternatives. The suggested alternatives are HFC and HC refrigerants, each with benefits and drawbacks. To meet the Montreal and Kyoto protocols, it is essential to find environmentally-friendly refrigerants. Hydrocarbon (HC) refrigerants are eco-friendly, but their flammability is dangerous. so, safety standards in many countries prohibit charging air conditioners above a certain level. Hydrocarbons require constant safety attention, such as preventing system leakage. Hydrofluorocarbons are incompatible with mineral oil, so they're used with polyolester (POE) oil. This polyolester oil is highly hygroscopic, causing moisture-related problems.

2 Literature review

The search for suitable alternatives to R22 and other toxic refrigerants with eco-friendly properties and comparable performance began in the early days. In the 1920s, 'Carrier and Water Fills' researched improved chiller refrigerants to replace R22. Donald et al. (1994) studied R-407C to replace R-22 in heat pumps and air conditioners. Kim et al. (1994) tested heat pumps with azeotropic R-134a/HC-290 and R-134a/R-600a mixtures. Macline & Leonardi (1997) measured refrigerant performance and suggested that zeotropic blends could reduce the high energy consumption of HC and HFC air-conditioning appliances. Purkayastha and Bansal (1998) studied HC290+LPG to replace R-22. Yang et al. (1999) studied R-32 / R-290, R-125 / R-290, R-32 / R-125 / R-152a, and R-32 / R-125 / R-290 to find a replacement for R-22. Motoshi (1999) studied lubricants for use with HFCs and compressor lubrication. Samuel and Daniel tested R-404A, R-407C, R-408A, R-410A, and R-507 to replace R-22 (2000). Jabaraj et al. investigated the performance of R407C/R290/R600a refrigerant mixtures without changing the mineral oil (2006). Oruc et al. conducted experiments with newly proposed HFCs such as R417A, R422A, R422D, and R424A to investigate their performance for use as drop-in replacements for the existing R22 (2016).

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From the literature, R22 is India's main refrigerant. R22 is nontoxic, nonflammable, and has a low ozone depletion potential, but it's being phased out. The Kyoto Protocol suggests reducing its use. Most alternative refrigerants lack R22's saturation properties, energy efficiency, and safety. Hydrocarbon (HC) blends are more energy-efficient but flammable, limiting their use in existing systems. HC and HFC refrigerants can be mixed to make them less flammable. HC refrigerants can be mixed with HFCs. HFC/HC mixtures miscible with Polyester oil. HFC/HC mixtures have a lower GWP than HFC alone.

3 Refrigerant mixture selection

Hydrocarbons, also known as HCs, share many of the same characteristics as CFCs and HCFCs. Butane (R600), isobutane (R600a), and propane (R290), as well as mixtures of these three, are the hydrocarbons that have the best properties as a refrigerant. These substances are suitable candidates for replacing HCFCs because they satisfy the thermophysical and other requirements. Hydrofluorocarbon (HFC) and hydro carbons (HC) refrigerant mixture is shown in a lot of published literature to be a better replacement for HCFC22 in refrigeration and air-conditioning systems. In addition, it was hypothesized that the incorporation of HCs(290/600/600a) into HFC152a would render the compound compatible with POE oil.

According to the research that has been conducted on the various properties of refrigerants, the properties of vapour pressure, liquid density, vapour density, latent heat, liquid viscosity, and vapour viscosity are the most important properties to consider when trying to predict the properties of alternative refrigerants and mixtures. For an operating temperature of 0°C to 50°C, REFPROP 9.0 (NIST2002) software predicted the aforementioned properties of the refrigerant mixtures including HCFC22. The proposed HC and HFC mixtures each have their own unique properties. The following percentages of composition need to be researched in order to replace R22:as will be shown below.

- M1 -R152a-55%/R290-25%/R600-10%/R600a-10%
- M2 - R152a-25%/R290-15%/R600-30%/R600a-30%
- M3 - R152a-15%/R290-15%/R600-35%/R600a-35%
- M4 - R152a-15%/R290-55%/R600-15%/R600a-15%
- M5 - R152a-55%/R290-15%/R600-15%/R600a-15%

M5 is one of the chosen combinations of refrigerants. Low temperature glide was found, and M3 has a low GWP (15.1) and a high COP and acceptable temperature glide.

Table1 shows the COP and temperature glide for mixtures with different amounts of HCFC 22 at a temperature of -15oC for the evaporator and 35oC for the condenser. A considerable number of refrigerant mixtures that exhibit large temperature glides during phase change. The use of such refrigerants will influence the temperature distribution in the evaporator and condenser, thereby influencing the behaviour of a refrigeration system. The acceptable limits for temperature glides of refrigerant mixtures for both Te and Tc should be less than 12oC under specified operating conditions. Since they are non-azeotropic (refrigerant) mixtures, their temperature changes during a phase change at constant pressure.

GWP of blend can be calculated (from IPCC 2013),

GWP of blend (M1) = Proportion by % mass of component A x GWP of A +
Proportion by % mass of component B x GWP of B +
Proportion by % mass of component C x GWP of C +
Proportion by % mass of component D x GWP of D

Table 1. Compositions of refrigerant mixtures, along with their corresponding coefficients of performance and temperature glide

Refrigerant Mixtures	R152a (%wt.)	R290 (%wt.)	R600 (%wt.)	R600a (%wt.)	Temperature glide (°C)	
					Teva	Tcond
M1	55	25	10	10	6.5	7.2
M2	25	15	30	30	10.3	13.9
M3	15	15	35	35	6.9	10.3
M4	15	55	15	15	5.6	4.9
M5	55	15	15	15	2.6	2.4

Table 2. Thermo physical properties of R22 and refrigerant mixtures with various proportions of R152a/R290/R600/R600a

Refrigerant	**Molecular Weight (g/mol)**	**Boiling Point (°C)**	**Freezing Point (°C)**	**Critical Temperature (°C)**	**Critical Pressure (°C)**	**Latent Heat (kJ/kg**	**Saturated Liquid Density (kg/m3)**	**Saturated Vapor Density (kg/m3)**	**Temperature Glide (°C)**	**Saturated Liquid Specific Heat (kJ/kg/K)**	**Specific volume m3/kg**
R22	86.498	40.81	151.41	96.145	49.900	182.74	1190.7	44.232	0.0	1.4191	0.00190
M1	58.720	29.98	124.65	117.14	51.156	230.36	604.01	44.31	6.8	2.4893	0.00341
M2	55.813	33.14	143.12	132.15	44.578	283.78	535.42	23.42	10.8	2.6521	0.00414
M3	56.911	36.11	166.23	142.19	41.334	302.89	532.85	18.63	6.3	2.6533	0.00422
M4	46.232	32.83	159.67	103.56	44.924	256.23	454.36	44.80	5.4	3.1474	0.00431
M5	62.441	34.08	147.78	115.86	50.367	276.61	774.21	18.83	2.5	1.6631	0.00306

Figure1 shows the variation of vapour pressure with saturationtemperature. The properties are obtained from REFPROP8.0. With an increase in HC mass percentage, suction and discharge pressures rise. M3's vapour pressure is 39.72% lower than R22's at 50oC. R22's pressure ratio at 7oC evaporating temperature was 3.12%, while R152a's was 3.94. Due to M3's higher suction pressure, adding HC to R152a reduced the pressure ratio. With the addition of HC to R152a, the pressure ratio decreases, resulting in less compression work and a better system.

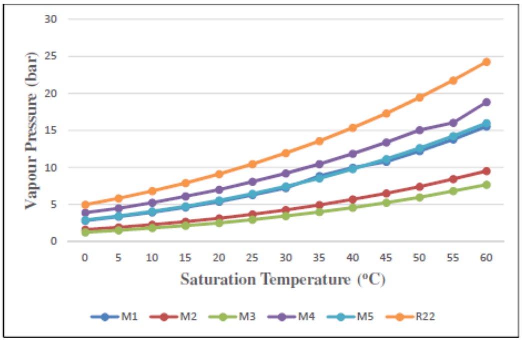

Fig. 1. Variation of Vapour Pressure with Saturation Temperature

Figure 2 shows HCFC 22 and M3 liquid density versus saturation temperature. Liquid density is 47.9% to 48.98% lower than HCFC22 and decreases with HC290 blend in refrigerant mixture. HC refrigerant mixtures will have less charge.

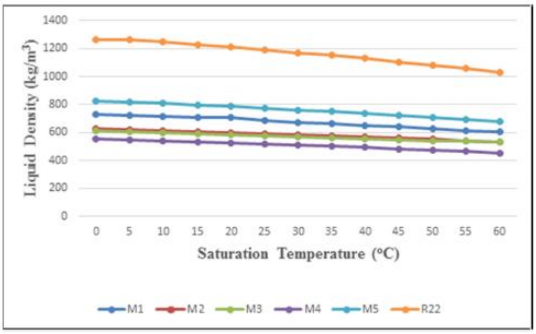

Fig. 2. Variation of Liquid Density with Saturation Temperature

Liquid density increases compression work. Since the system is vapour compression refrigeration, vapour density must be studied. Figure 3 shows refrigerant vapour density vs. saturation temperature. The mixture's vapour density is 69.27% to 71.63% lower than R22.

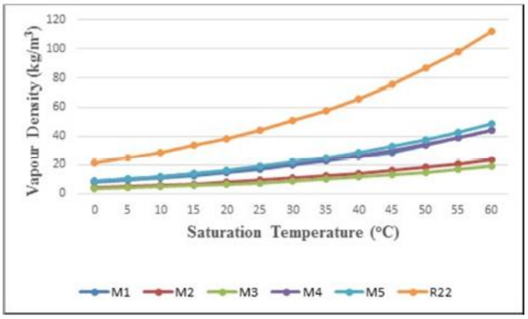

Fig. 3. Variation of Vapor Density with Saturation Temperature

Figure 4 shows the HCFC22 and HC latent heat variation. Latent heat of vaporisation determines refrigeration capacity. HC mixtures have higher latent heat than HCFC22. Due to an increase in HFC152a mass, alternative mixtures have 24.34% to 40.30% higher latent heat of vaporisation. Thus, alternative mixtures with less mass may have the same or better cooling effect than R22.

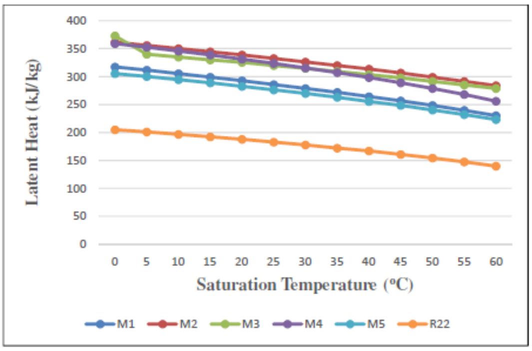

Fig. 4. Variation of Latent heat with Saturation Temperature

Figures 5 and 6 show the liquid and vapour viscosity of the refrigerants. Reduced viscosity improves refrigerant heat transfer. Viscosity affects flow in expansion devices. The hydrofluorocarbon and hydrocarbon mixtures are less viscous than R22. Increased HC in the refrigerant mixture reduces viscosity. In liquid and vapour phases, the selected mixtures are 3.02% to 12.14% less viscous than R22. HC/HFC mixtures have better transport and thermodynamic properties.

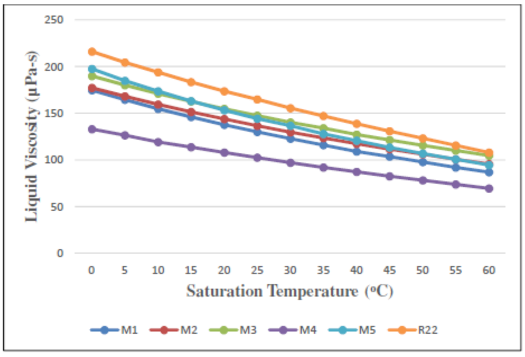

Fig. 5. Variation of Liquid Viscosity with Saturation Temperature

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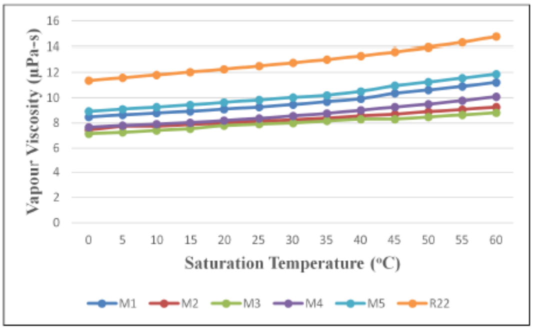

Fig. 6. Variation of Vapor Viscosity with Saturation Temperature

4 Experimental test facility

An experimental test facility has been created to study R22 and other vapour compression refrigeration systems.

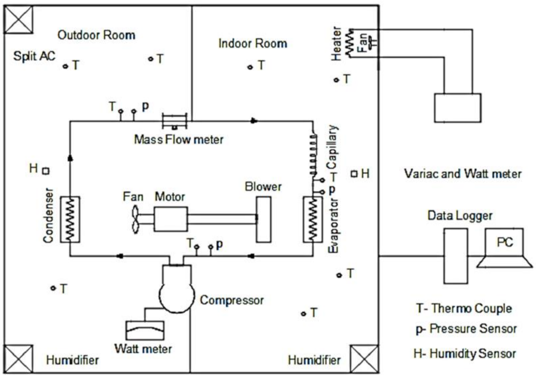

Fig. 7. Experimental test facility

Figure 7 shows a window air conditioner and environmental chamber. The indoor and outdoor chambers meet BIS: 1391-1992a. A 2500 W heater is mounted in the indoor room to provide heat load. The power supply can be measured using a watt metre with 0.5% accuracy. Fans circulate air within chambers to ensure uniform temperature. Poly Urethane Foam (PUF) insulated panels are used as chamber material to limit heat infiltration to 5% of capacity. In outdoor chambers, a 2 TR split air conditioner removes condenser heat to maintain air temperature. Indoor and outdoor chambers have humidifiers to maintain relative humidity. The mass flow metre panel, measures Coriolis refrigerant mass flow rate. A 0.5% accurate digital Watt metre measures compressor power. Pressure transducers with 0.25% accuracy measure refrigerant pressures, and J-type thermocouples with 0.1% accuracy measure temperatures. All humidity sensors are linked to an AGILENT computerized data acquisition system. Experimentation and analysis were carried out in accordance with the BIS:1391-1992 standard. A test for heat infiltration was performed before the experimentation was carried out, and the results showed that the leakage was within the acceptable range. Experiments were performed for six different sets of condensing temperatures, with the evaporating temperature being held constant at 7 degrees Celsius (45o, 47o, 49o, 51o, 53o and 55oC). In all test conditions, heat infiltration was added to the heater load to get actual refrigeration capacity. First, R22 refrigerant was used. Keeping the evaporator temperature at 7oC, the condenser temperatures are 45o, 47o, 49o, 51o, 53o, and 55o C.First, R22 system performance was determined. The AC system was retrofitted with R152a/R290/R600/R600a blends and mineral oil as lubricant. All mixtures were tested (M1-M5).

5 Results and discussion

First sequence of tests maintains condenser temperature (Tc) at 50oC while varying evaporator temperatures (Te) from 2oC to 12oC. In the second series of tests, condenser temperatures are 45oC, 47oC, 49oC, 51oC, 53oC, and 55oC, but evaporator temperature is 7oC. After the R22 tests are complete, the air conditioner is retrofitted with M1 to M5 refrigerant mixtures and the same experimental procedure is repeated. The performance parameters such as Coefficient of performance, Refrigeration effect, Pressure ratio, Compressor discharge temperature, Refrigerant mass flow rate, Compressor power are analyzed during the experimentation.

5.1 Coefficient of Performance

Figure 5.16 depicts how the temperature of the evaporator impacts the coefficient of performance (COP) for different mixtures of R152a/R290/R600/R600a and R22. The higher the temperature of the evaporator, the greater the increase in the coefficient of performance. This increase in COP is dependent on two different parameters, namely the refrigeration effect, which grows as the evaporator temperature rises, and the compressor power, which falls as the evaporator temperature rises.

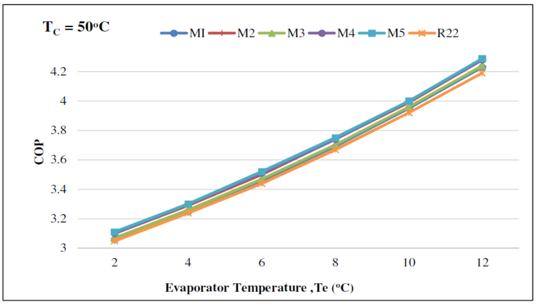

Fig. 8. Effect of Evaporator temperature on Co-efficient ofPerformance

As can be seen in Figure 8, M3 has a higher COP than the other proportions due to its superior cooling capacity. Based on the data presented above, it can be deduced that at 7°C evaporator temperature and 50°C condenser temperature, M3, M2, M5, and M1 all have higher performance coefficients than R22, while M4 has a lower performance coefficient than R22 by a difference of 2.93%. For a range of condenser temperatures, the COP increases. This may be because of their greater latent heat of vaporisation, which improves their refrigeration capacity. A higher COP is the result of an increase in refrigeration capacity that exceeds the increase in compressor work.The influence of R152a/R290/R600/R600a/R22 mixture mass proportion on the Coefficient of Performance with condenser temperature is shown in Figure 9. As the condenser temperature rises, the cooling capacity falls because the refrigerating effect is diminished and the pressure ratio across the compressor rises, increasing the work required by the compressor. As a result, the Coefficient of Performance is lowered due to the effects of these two parameters taken together (COP).

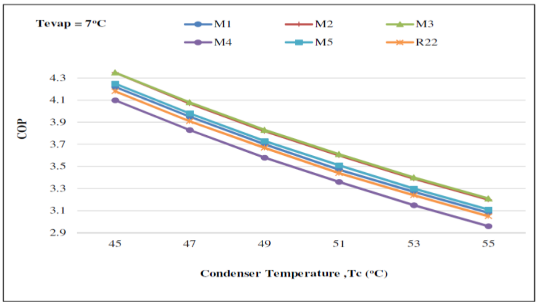

Fig. 9. Effect of Condenser temperature on Co-efficient ofPerformance

5.2 Refrigeration Effect

Figure10 shows the refrigeration effect with evaporator temperature for R152a/R290/R600/R600a and R22. Increasing evaporator temperature and latent heat of vaporisation increase refrigeration capacity. Due to their high latent heat of vaporisation, M3 has the highest cooling capacity. M3, M2, M4, M5, and M1 have higher refrigeration effects than R22 at 7oC evaporator and 50oC condenser temperatures.

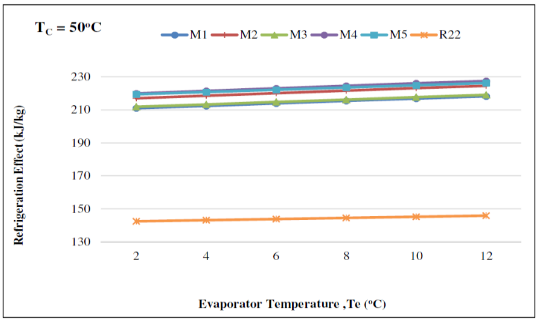

Fig. 10. Effect of Evaporator temperature on Refrigeration effect

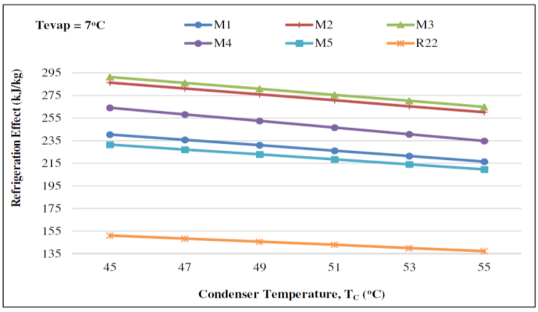

Fig. 11. Effect of Condenser temperature on Refrigeration effect

Figure 11 shows how condenser temperature affects R22 and refrigerant mixtures. Latent heat of evaporation decreases as condenser temperature rises, reducing cooling capacity.

5.3 Pressure ratio

Figure 12 shows how evaporator temperature affects refrigerant pressure ratios. As evaporating temperature rises, suction pressure increases while discharge pressure remains constant, reducing the compressor's pressure ratio. Even though mixture operating pressure is higher than R-22, pressure ratio is lower at various evaporator temperatures.Figure 12 shows that M3 has a high-pressure ratio due to its lower suction pressure at 7oC evaporator and 50oC condenser temperatures. M3, M2, M5, and M1 have higher pressure ratios than R22, while M4's is lower.

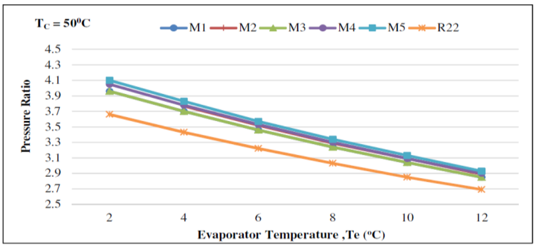

Fig. 12. Effect of Evaporator Temperature on Pressure Ratio

Figure 13 compares pressure ratios of R152a/R290/R600/R600a with R22 and condenser temperatures. Condenser temperature increases the pressure ratio. Increasing condenser temperature raises compressor discharge pressure, which raises compressor pressure ratio. Due to its low suction pressure, M3 has the highest-pressure ratio.

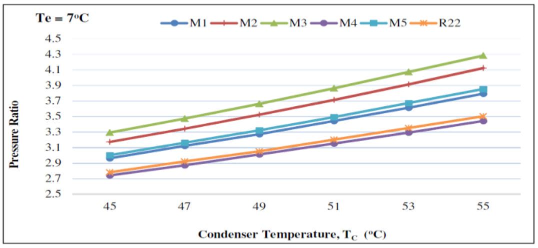

Fig. 13. Effect of Condenser temperature on Pressure ratio

5.4 Compressor Discharge Temperature

Figure 14 compares refrigerant discharge and evaporator temperatures. Evaporator temperature raises compressor discharge temperature. Figure shows that R22 has a high discharge temperature, while M3, M4, M2, M1 and M5 are 23, 99%, 17, 50%, 13,98% and 11,41% lower. Low compressor discharge temperature improves lubricant-refrigerant compatibility and compressor reliability.

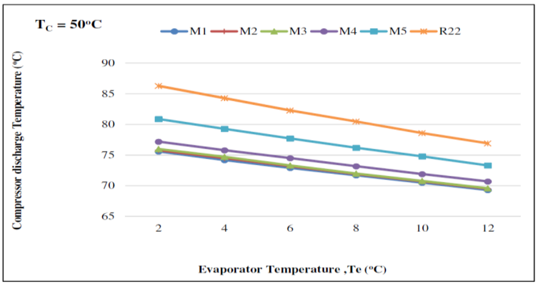

Fig. 14. Effect of Evaporator temperature on compressor dischargeTemperature

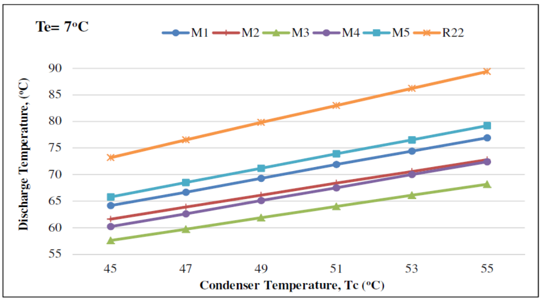

Fig. 15. Effect of Condenser temperature on Discharge temperature

The impact of changing the condenser's temperature on the compressor's discharge temperature is shown in Figure 15, for a range of mass ratios of R152a, R290, R600, and R600a to R22. As the condenser temperature rises, so does the temperature at the compressor's discharge.

5.5 Mass flow rate

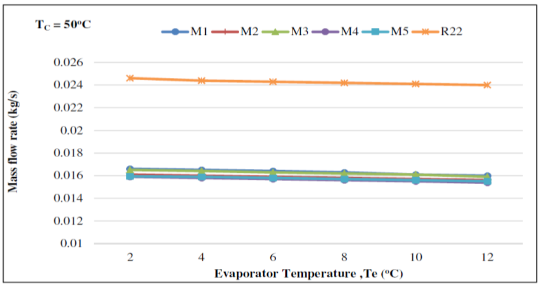

Fig. 16. Effect of Evaporator temperature on Mass flow rate

Figure 16 shows how evaporator temperature affects refrigerant mass flow rate. Due to high vapour density, R22 has a higher mass flow rate than new refrigerants. M3, M2, M4, M1 and M5 have mass flow rates 51.55 %, 50.01 %, 40.96 %, 38.44 % and 34.59 % higher than R22 at 7oC evaporator and 50oC condenser.

Figure 17 shows condensing temperature's effect on refrigerant and mixture mass flow rate. Figure 17 shows how condenser temperature affects refrigerant mass flow rate. R22 has the highest mass flow rate due to its high vapour density.

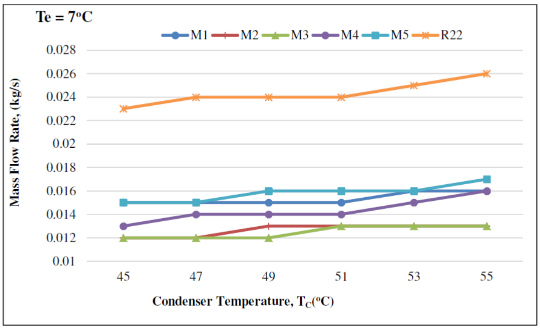

Fig. 17. Effect of Condenser temperature on Refrigerant mass flowRate

5.6 Compressor power

Figure 18 compares the mixtures of R152a/R290/R600/R600a and R22 compressor power with evaporator temperature. Increased evaporator temperature at constant condenser temperature reduces compressor power due to pressure ratio. Condenser temperature increases power consumption.

It is clear from looking at Figure 18 that the compressor power of M4 is considerably higher than that of the other models. It is possible to deduce from figure 5.13 that the compressor power of M3, M2, M5, and M1 is, respectively, 5.22%, 5.12%, 1.76%, and 0.88% lower than that of R22, while the compressor power of M4 is, in contrast, 2.6% higher.

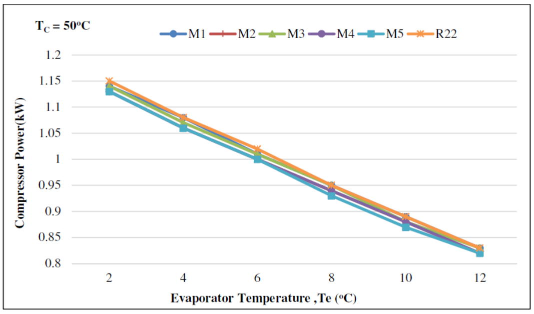

Fig. 18. Effect of Evaporator temperature on Compressor power

The effect of condenser temperature on the compressor power of varying mass proportions of R152a/R290/R600/R600a and R22 is depicted in Figure 19.

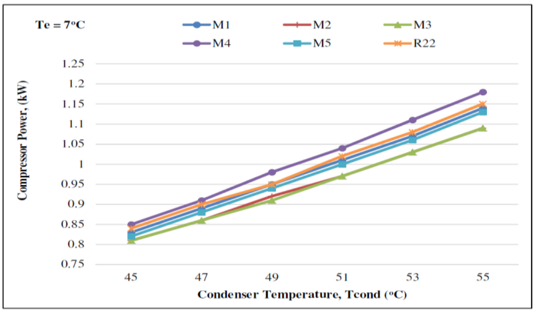

Fig. 19. Effect of Condenser temperature on Compressor Power

The increase in the temperature of the condenser causes an increase in the pressure ratio across the compressor, which in turn causes an increase in the power output of the compressor. On the other hand, it was discovered that the power of the compressor was decreasing and getting closer to that of R-22 as the HC content of the refrigerant mixture increased. It has also been observed that, of all the mixtures, M3 has the lowest amount of power consumption. When compared to the other models, Figure 19 reveals that the M4 compressor has the highest possible power output, measuring 1.2 kW.

6. Conclusion

The impact of the new mixtures was studied in an experimental study on a window air conditioner using the five selected refrigerant mixtures. The desired performance parameters, such as coefficient of performance, compressor power, refrigeration effect, mass flow rate, compressor discharge temperature and pressure ratio, and so on, were investigated and compared to the base refrigerant R22. Among the five alternative mixtures tested, the composition known as Mixture-3 performed the best.

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