AN EXPERIMENTAL ANALYSIS OF THE EFFECT OF REFRIGERANT CHARGE LEVEL AND OUTDOOR CONDITION ON A WINDOW AIR CONDITIONER

by

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R22 is an hydrochlorofluorocarbon widely used in refrigerant and air conditioning plants and although it has a low ozone depletion potential (0.05), it is necessary to consider the large amount that commonly escapes from commercial units to the atmosphere. This paper presents experimental investigation on the performance of a window air conditioner operated with R22 and the M20 (80% R407C and 20% HC blend by wt.) refrigerant mixture tested under different refrigerant charge levels and outdoor conditions. Experiments were conducted in accordance with the Bureau of Indian standards procedure in a psychrometric test facility. Capillary and charge optimization tests were conducted for the both R22 and the M20 refrigerant mixture based on maximum coefficient of performance. Refrigerant charge in the air conditioner was systematically varied and the influences of refrigerant charge quantities and outdoor conditions on system performance are studied for both R22 and the M20 refrigerant mixture. At each charge levels, the outdoor room conditions were changed in accordance with Bureau of Indian standards. It is observed that R22 is more sensitive to deviations in charge levels as compared to the M20 refrigerant mixture. A decrease in charge level of about 7% reduced the system refrigerating capacity by 11.3% with R22 while with the M20 refrigerant mixture it reduces by 6.9% only. Similarly an over charge by 7% reduces the refrigerating capacity of the system by 13.8% with R22 while with M20 it reduces by 6.5% only.

Key words: *refrigerant mixture, over and under charged, outdoor conditions, system performance*

Introduction

Several new environmental problems have been encountered in recent decades. Among these, global warming and ozone depletion caused mainly by CFC and HCFC refrigerants are major issues. The chlorine contained in CFC and HCFC is believed to be the major factor causing the ozone layer depletion. These effects were not considered when the refrigeration industry first adopted these refrigerants as working fluids in refrigerators, heat pumps, and air conditioners. R22 is widely used in refrigeration and air-conditioning plants and it is necessary

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to consider the large amount that commonly escapes from commercial units to the atmosphere. More over, its sales volume has been the largest among various refrigerants. Even though the ozone depleting potential of HCFC22 is not as high as other CFC, it still contains ozone depleting chlorine and hence the parties to the Montreal Protocol decided to phase out HCFC22 eventually and the regulation for the HCFC production has begun from 1996 in the developed countries. There are a wide variety of options available to replace both the HCFC refrigerants and equipment or systems. Thermodynamic similarity is important when retrofitting the existing system, otherwise capacity and efficiency may be lost. Initially, AREP has identified many numbers of alternative refrigerants. Among many refrigerants HFC and HC are better alternatives for R22. HC such as propane (R290) [1, 2] could be used in the air conditioning system. Purkayastha *et al.* [1] have compared the performance of R290 and commercial LPG and reported that the R290 COP was higher than that of R22 system. Devotta *et al*. [2] conducted experiment in a window air conditioner using R22 and R290 and concluded that the performance of R290 was better than that of R22. Devotta *et al*. [3] carried out theoretical performance study using R134a, R290, R407C, R410A, and three blends of R32, R134a, and R12. It was reported the R290 COP was marginally higher than that of R22. It also has zero ODP and very low GWP; however its flammability is a major draw back.

Another new refrigerant is R407C [3, 4] which is a ternary zeotropic mixture of R32, R125, and R134a (23%, 25%, and 52% by wt.), whose main thermodynamic properties are very close to those of R22 with the exception of the temperature gliding during the phase change process at a constant pressure. Devotta *et al*. [3] carried out theoretical performance study using R134a, R290, R407C, R410A, and three blends of R32, R134a, and R12 suitable for window air conditioner. It was reported that for retrofitting, R407C was probably the best candidate. Devotta *et al*. [4] investigated experimentally the performance of R22 and R407C in a window air conditioner and concluded that the COP of R407C was lower in the range of 7.9% to 13.5% than that of R22. However HFC are immiscible with the conventional mineral oil, which is used in hermetic compressor and thus a new generation of synthetic oils (usually Polyol ester based) have come into use. But POE is highly hygroscopic in nature.

Jabaraj *et al*. [5] studied the possibility of using R407C/R290/R600a refrigerant mixture as a substitute for R22 in a window air conditioner with mineral oil as compressor lubricant. HC blend percentage was varied from 10 to 25% in steps of 5% and evolved an optimal composition for the mixture (M20: R407C 80% and HC blend 20%) in respect of better COP and per day energy consumption. Jabaraj *et al*. [6] also reported that M20 could be a viable alternative for R22, since it had 9.5-12.7% higher refrigerating effect and 8.2-11.2% higher COP than that of R22. In order to properly use the new refrigerant, the thermodynamic, flow, and heat transfer properties are to be studied in retrofitted condition. In particular, a detailed understanding of the characteristics of the M20 refrigerant mixture is very important in undercharged and overcharged condition for air conditioning systems. In the present study experiments were carried out to quantify the deviation in performance of a residential 1.5 TR window air conditioner in under and overcharged conditions at different outdoor conditions with capillary tube as expansion for the M20 refrigerant mixture in retrofitted condition. The same is compared with that of R22 charged in the same window air conditioners.

Review of literature

The amount of refrigerant charge and outdoor conditions in the air conditioner are the primary parameters influencing energy consumption. Under charged or over charged into the system will degrade system performance and deteriorate system reliability [7-15]. In order to provide the rated performance (capacity and efficiency) through out its working, the system must be charged with optimum amount of refrigerant. However, it is difficult to determine the optimum charge due to its dependency on operating conditions and expansion device, compressor and heat exchanger used in the system. Estimating the refrigerant charge requires knowledge of the state (quality, temperature, pressure, *etc*.) of the refrigerant and geometric variables (tube length, diameter, *etc*.) throughout the system.

Robinson *et al*. [7] studied the effect of refrigerant charge on the performance of three blends of R134a and R32 with different mass proportion in an air-to-air heat pump. The COP and capacity variations were analyzed for different charge and outdoor conditions. Farzad *et al*. [8] quantified the influence of the refrigerant charge on the system performance variables such as capacity, flow rate, evaporator superheat, power consumption, and seasonal energy efficiency ratio were discussed with capillary tube as an expansion in a 3 TR split system air conditioner. It was reported that a larger degradation in system capacity was experienced in undercharged than in overcharged condition at the studied operating conditions. Choi *et al.* [9] investigated the effects of off design R22 refrigerant charge amount from –20% to +20% of full charge on the performance of water-to-water heat pump with electronic expansion valve (EEV) and capillary. It was reported that the system with capillary was more sensitive to off-design refrigerant charge level than that of system with EEV. The degradation of performance was higher at undercharged conditions than that at overcharged condition for capillary system. Choi *et al*. [10] investigated the effects of the expansion device on the performance of a water-to-water heat pump using R407C at various charging conditions. It was reported that the degradation of system performance was lower for the R407C system with EEV as compared to that of system with capillary tube.

O'Neal *et al.* [11] conducted an experiment on a 10.6 kW capacity split-system air conditioner with capillary tube expansion at different charge levels and different outdoor conditions and determined its effect on the system performance. It was observed that the degradation in performance for each of the variables was generally greater in undercharged than in overcharged condition. Corberan *et al*. [12] conducted an experiment on water-to-water heat pump and analyzed the effect of refrigerant charge variation on the performance of system. It was concluded that the system performance was highly dependent on refrigerant charge quantity. Bjork *et al.* [13] and Cho *et al*. [14] also reported that the system performance was affected with change in refrigerant charge. The above reviewed literatures reported that improper charging was detrimental to the performance of the systems. However, the intensity of degradation varies with the refrigerant used.

Most of the previous researches related to effects of refrigerant charge and outdoor conditions on the heat pump with capillary tubes and split-system air conditioners performance with capillary tubes. Jabaraj *et al*. [5, 6] have reported the M20 refrigerant mixture could be used in a window air conditioner designed for R22 as an alternative. It was also found that M20 performs better than that of R22. In the present scenario of preparing for a phase out of R22, studies on ozone friendly M20 refrigerant mixture are more appropriate in retrofitted condition since; previous studies reported that the system performance was dependent on types of refrigerant. An experimental study on the effects of charge deviations in an air conditioner having capillary tube as an expansion device is essential, to understand the system behavior when M20 is considered as a retrofit refrigerant.

The objective of the present study is to quantify the performance variation in a residential 1.5 TR window air conditioner in retrofitted condition charged with the M20 refrigerant mixture for ±7% refrigerant charge quantity than the optimum charge and compare the same with that of R22. The charge in the system is systematically varied and its effect on performance parameters is analyzed at different outdoor conditions prescribed in BIS standard [15] for testing of air conditioners.

Experimentation

The experimental facility mainly consists of the psychrometric test room and the test unit suitably instrumented to conduct the performance study.

Psychrometric test room

An experimental setup was constructed (as shown in fig. 1) that would facilitate performance assessment of a window air conditioner on various indoor and outdoor conditions in accordance with different standards (BIS and ASHRAE). The psychrometric room consists of two adjacent chambers to maintain indoor and outdoor conditions respectively. Both the rooms have separate AHU with a cooling coil (dehumidifier), air heater, and steam injection facility (humidifier) which are controlled/modulated by suitable feed back control system to maintain the required indoor and outdoor test conditions given in tab. 1.

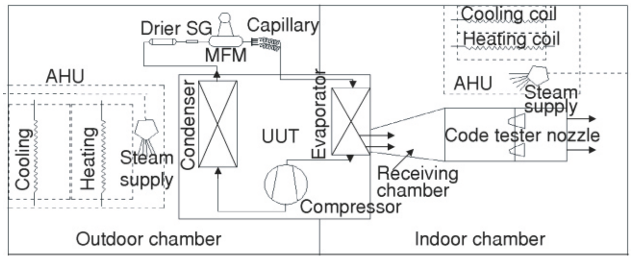


Figure 1. Schematic diagram of the experimental setup

Table 1. Operating conditions for the performance over a range charge quantity

	Standard test	Types	Outdoor conditions	
			DBT [°C]	WBT [°C]
1	BIS DT	Capacity rating	35	30
2	BIS ETA	Capacity rating	35	24
3	BIS ETA	Maximum	43	26
4	BIS ETB	Capacity rating	46	24

The facility has been designed to maintain individual temperature readings within the tolerance as prescribed in BIS [15] and ASHRAE [16] standards (±0.5 °C for dry bulb temperatures and ±0.3 °C for wet bulb temperatures). Six temperature sensors were strategically located in each room to the uniformity in room temperature within ± 0.5 °C. To measure the supply air flow rate from the test unit a code tester is available in the indoor room and the code tester design is based on ASHRAE standard 41.2-1987 [17]. This consists of a set of nozzles that can be suitably selected to allow the air to flow through the selected nozzle. The pressure drop across the nozzle is measured using a differential pressure transducer. There is an auxiliary blower driven with a variable frequency drive to maintain zero gauge pressure at the receiving chamber. This is done so that the test unit does not experience any resistance to throw the supply air due to upstream surging effect of the flow through nozzle. The code tester was connected with a suitable leak proof duct to the supply grill of the test unit.

Test unit

The test unit was suitably modified to connect temperature sensors (RTD PT100-class A ±0.15 °C accuracy) and pressure sensors (±0.1% accuracy) across each component. The mass flow rate of refrigerant was measured by a Coriolis type mass flow meter (±0.1% accuracy) connected in the liquid line. The entire flow lines along with components were properly insulated to avoid heat infiltration. The power consumed by the compressor was measured by a separate power meter (±0.25% accuracy). To optimize the capillary length, four lengths viz 0.8382 m, 0.762 m, 0.7112 m, 0.6604 m (1.6764 mm diameter), were fixed to a header. Suitable hand shut-off valves were used to select the required capillary to be included in the circuit. One sight glass was provided in the liquid line to check the condition of the condensed refrigerant in the circuit. The supply and return air temperatures (DBT and WBT) were measured by suitable RTD (class-A) fixed at appropriate locations.

Data logging

All measured data are logged into a PC through a suitable data logging system. Once steady-state condition is achieved all the data will be automatically logged into the system. The steady-state condition is manually confirmed by checking the uniformity in temperature indicated by all the room temperature sensors in accordance with the test conditions and the power consumption of the compressor.

Experimental procedure

The performance of the system was determined in accordance with the BIS [15] test conditions for residential sized air conditioner as given in tab. 1. The tests were conducted by varying two variables: (1) the outdoor room temperature and (2) the refrigerant charge. There were four different outdoor conditions and six charge levels for R22 and the M20 refrigerant mixture. Thus for both the refrigerant, 24 different combinations of test conditions were considered. The performance was assessed in a psychrometric test facility using air enthalpy method.

In air enthalpy test method, refrigerating capacity is determined from the difference in enthalpies obtained against DBT and WBT of air entering and leaving the test unit and the associated air flow rate under specified test conditions. The mass of air was calculated using the measured pressure drop across the nozzle and DBT/WBT of sample air in the code tester based on ASHRAE standard 41.2-1987 [17]. Refrigerant side measurements were also made to ensure that the maximum difference between the air side and refrigerant side capacity was less than 6% as prescribed in the standards. Properties of refrigerants were extracted from REFPROP [18].

To have a realistic comparison of the performance of the M20 refrigerant mixtures with a conventional refrigerant, the experiment was carried out initially with the conventional refrigerant R22. During experimentation with R22 the capillary length as recommended by the manufacturer (0.8382 m length and 0.001524 m diameter) was considered and the charge quantity was optimized as the system flow volume had changed due to the alterations made for fixing

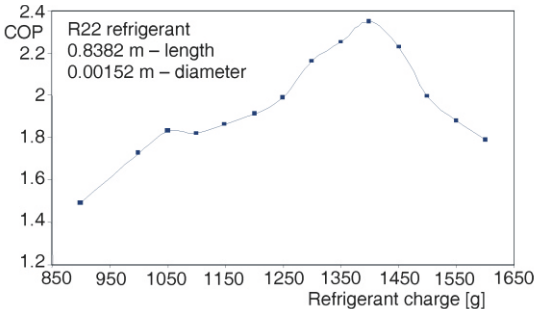


Figure 2. Variation of the COP with refrigerant charge quantity

the mass flow meter, drier, sight glass, and capillary header. Subsequently, the capillary and charge were optimized for the M20 refrigerant mixture. The refrigerant charge quantity and the capillary tube lengths were optimized for maximum COP at BIS DT test conditions as indicated in tab. 1. It is to be noted that the BIS DT is the test condition closet to the normal ambient temperature hence the charge quantity and capillary was optimized at BIS DT condition. The charge quantity of R22 was varied from 900 g to 1600 g in steps of 50 g, and as seen in fig. 2 with 1400 g charge, the system attained the maximum COP. With the optimum charge and capillary, the performance test was carried out for R22.

The charge quantity of the mixture required for the M20 system is not equal to that of R22, as the specific volume of M20 is different. Jabaraj *et al*. [5] have calculated the equivalent charge quantity based on the specific volume of M20 at the suction of the compressor for domestic air conditioners. The same procedure is followed in the present study for the calculation of the equivalent charge quantity. Considering the specific volume (REFPROP [18]) of the refrigerants, the equivalent quantity of R22 for 1 g of the R407C and HC blend is 1.037 g and 2.29 g, respectively.

Let *HFC* be the mass of R407C present in the mixture, *HC* be the mass of *HC* blend in the mixture, *HCFC* be the R22 charge quantity in a system, and *m* be the mass fraction of *HC* blend, then,

$$1.037HFC + 2.29HC = HCFC \tag{1}$$

$$HFC \quad \frac{1 \quad m}{m} \; HC \tag{2}$$

Using the above equations the mass of the R407C and HC blend present in unit mass of M20 was calculated. Before starting the experiment with the mixture, it was prepared separately in a cylinder. For the mixture, the equivalent charge quantity for 1400 g of R22 was obtained, along with the individual mass of the R407C and HC blend. The refrigerants were weighed individually in an electronic balance with an accuracy of ±1 g and filled in the cylinder with the help of a suitable charging manifold.

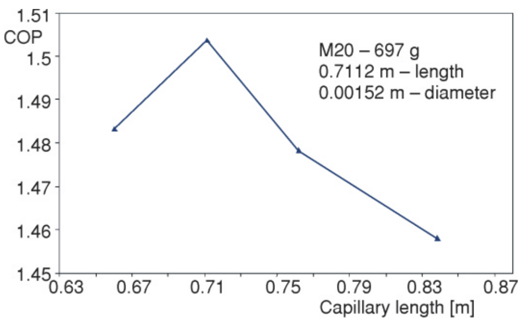


Figure 3. Variation of M20 COP with refrigerant charge

For M20, starting at 697 g, the capillary was optimized to be of 0.7112 m length as shown in fig. 3. Subsequently the charge was increased in steps of 39 g (which is the equivalent quantity for 50 g R22) for the same capillary of 0.7112 m. Even though the optimized capillary length and charge quantity are independent it can be judged that for a given refrigerant the relative trend remains fairly independent of the charge quantity. Hence, it was decided to consider only 0.7112 m length for all the other charge quantities. The charge quantity was increased until 1282 g, while the maximum COP was experienced at 1204 g itself as seen in fig. 4.

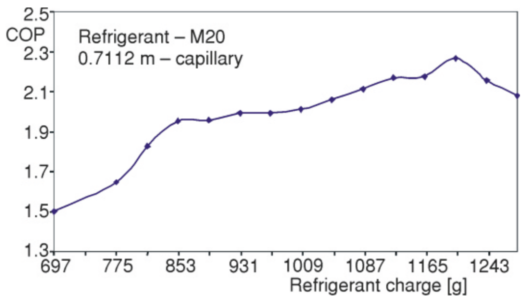


Figure 4. Variation of COP with M20 refrigerant charge quantity

However, for the optimized charge quantity of 1204 g the system was recharged and tested for the capillary length, viz., 0.6604 m, 0.7112 m, 0.762 m, and 0.8382 m. It was found that for 1204 g, 0.7112 m was the best capillary which confirmed that the initial judgment is correct. The observed maximum COP values are shown in fig. 3 in the first trial and in the recharge trial it is shown in fig. 5 for the same 0.7112 m capillary. Thus, 1204 g charge quantity with 0.7112 m capillary was considered as the optimum charge and capillary length for the M20 refrigerant mixture.

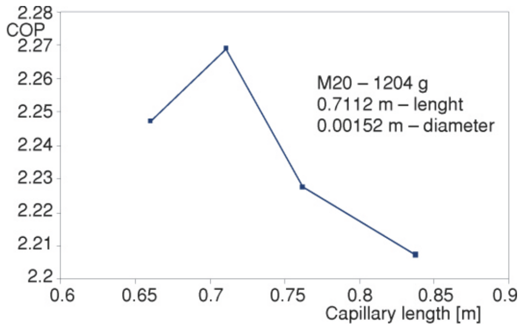


Figure 5. Variation of M20 COP with refrigerant charge

At the steady-state, the refrigerant mass flow rate, pressure, and temperature across the evaporator and condenser, compressor power, compressor dome temperature, DBT and WBT of return as well as supply air and mass flow rate of air, were measured for the various indoor and outdoor room conditions as given in tab. 1.

Before starting the experiment, the mixture was prepared separately in a cylinder. For the mixture, the equivalent charge quantity as against R22 charge quantity was obtained, in terms of the mass of R407C and the HC blend, as explained earlier. For the individual tests the charging was done from the lowermost level by carefully adding the mixture in required quantities alone. While adding charge enough care was taken to draw the refrigerant from the cylinder as liquid only to avoid composition shift.

Results and discussion

The performance of a window air conditioner with the M20 refrigerant mixture has been compared with the performance of R22 for different charge quantities and outdoor conditions while the indoor condition was maintained at 27 °C, 19 °C (DBT, WBT) BIS DT condition. For R22 the optimum charge is realized at 1400 g charge level for DT and ETA test conditions, while for ETAM and ETB, the optimum refrigerating capacity is realized at 1350 g charge level. For M20 refrigerant mixture, the optimum charge is realized at 1201 g charge level for DT and ETA test conditions where as for ETAM and ETB, the optimum charge is realized at 1163 g charge level. At each charge level the following performance parameters namely, refrigerant mass flow rate, refrigeration capacity, evaporator inlet/outlet temperature, evaporator and condenser inlet pressure, compressor power, and COP are compared.

Influences of refrigerant charge and outdoor condition on refrigerating capacity

Figure 6 shows the variation of R22 and the M20 refrigerant mixture refrigerating capacity under different refrigerant charge quantities and outdoor conditions. In under-charged conditions, the refrigerating capacity is reduced with decreasing refrigerant charge due to the reduction of the refrigerant mass flow rate (shown in fig. 7) and compressor efficiency resulting

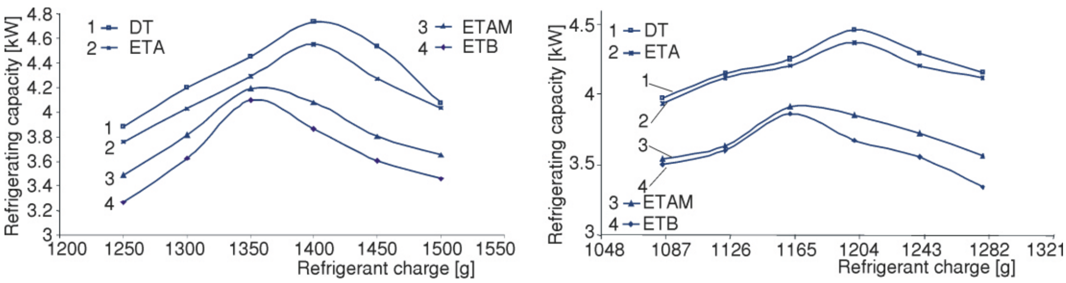


Figure 6. Variation of the R22 and M20 refrigeration capacity under different refrigerant charge quantities

from an increase in the suction temperature (shown in fig. 8). In an undercharged condition, the increase in the suction temperature at the compressor inlet with refrigerant charge yields higher specific volume which is reduces the mass flow rate in undercharged conditions.

In an undercharged condition the low evaporator inlet pressure (shown in fig. 10) as well as low mass flow rate of refrigerant leads to higher superheat prevailing in the evaporator. Since a major part of the evaporator is occupied by the superheated refrigerant. Due to this reason, the more significant reduction of evaporator efficiency occurred which leads to lower refrigerating capacity in undercharged conditions. It is evident from fig. 9 that the evaporator outlet temperature decreases with an increase in the charge levels and outdoor conditions. Therefore, the cooling capacity increases up to optimum refrigerant charge and then decreases with an increase in the refrigerant charge beyond optimum charge. In overcharged conditions, the refrigerating capacity is reduced due to a decrease of the temperature difference between the refrigerant and the air with the increasing refrigerant charge in the heat exchangers. In overcharged condition for the same indoor and outdoor condition, the evaporator inlet pressure is increased which reduces the temperature difference. The same results were also reported by O'Neal *et al.*[11], Farzad *et al.* [8], and Choi *et al.* [9, 10].

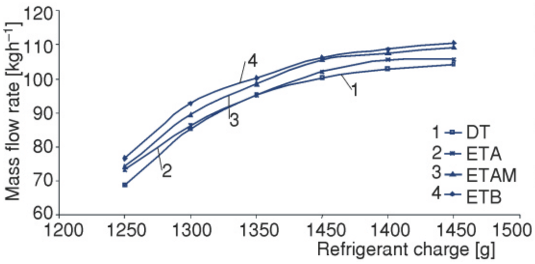
At the end section of the condenser the reduction of the heat transfer coefficient for the M20 refrigerant mixture would be more than that of R22, due to a lower saturation temperature of the M20 refrigerant mixture system (temperature glide) when compared with that of the R22 system. The capacity reduction in the M20 refrigerant mixture system in overcharged conditions slowly drops, because the overcharged refrigerant would enhance the heat transfer performance in the end section of the condenser and reduce the drop of temperature difference between the refrigerant and the air in the condenser. The same trend was reported by Choi *et al*. [10].

It is also observed that the optimum charge for test conditions DT and ETA are different from that of ETAM and ETB test conditions, since, the outdoor conditions for ETAM and ETB are higher than those of DT and ETA. With an increase in outdoor temperature, the optimum refrigerating capacity occurred at a lower charge for ETAM and ETB and the optimum refrigerating capacity occurred at 1350 g. It is observed that the maximum cooling capacity is dependent on the air temperature entering the condenser. Moreover as the inlet temperature and pressure of the evaporator are increased due to higher condenser inlet air temperature, the latter part of the evaporator is occupied by the superheated refrigerant, which leads to lower cooling capacity. The refrigerating capacity of the M20 refrigerant mixture realized the peak in the charge quantity of 1201 g for the DT and ETA test condition, whereas for ETAM and ETB, the optimum refrigerating capacity was realized at 1163 g. Thus, the charge that produced optimal performance at one temperature produced a sub-optimal performance at other temperatures. The same results were reported by Farzad *et al.* [8].

The refrigerating capacity of R22 in a 7% undercharged condition is 11.3% lower than that of the optimum, whereas the refrigerating capacity of the M20 refrigerant mixture is 6.9% lower than that of the optimum. It is also found that the refrigerating capacity of R22 in a 7% overcharged condition is 13.8% lower than that of the optimum, whereas the refrigerating capacity of the M20 refrigerant mixture is 6.5% lower than that of the optimum for the DT operating condition.

In an optimum charge condition, the refrigerating capacity of R22 at BIS DT condition is 18.4% higher than that of the BIS ETB condition, whereas the refrigerating capacity of the M20 refrigerant mixture is 17.6% higher than that of the BIS ETB condition. It is observed that the impact of outdoor condition on the performance of M20 refrigerant mixture is lower than that of R22. At higher operating conditions also the degradation in performance of the M20 refrigerant mixture is lower than that of R22, due the temperature glide in the heat exchangers.

Figure 7 shows the mass flow rate of R22 and the M20 refrigerant mixture under different charge levels and outdoor conditions. It is observed that the mass flow rate increased in undercharged conditions until the optimum charge level, and further increased for overcharged conditions beyond the optimum charge for both R22 and the M20 refrigerant mixture. The increment in the mass flow rate for the certain degree of undercharged conditions is higher than the increment experienced for the same degree of overcharged conditions.



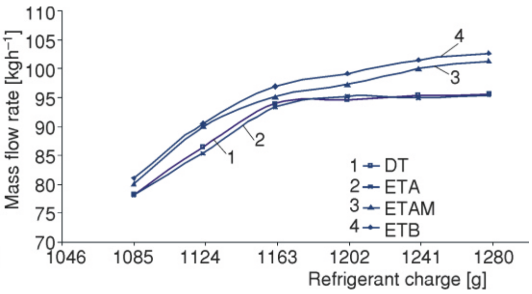


Figure 7. Variation of the R22 and M20 refrigerant mass flow rate under different operating conditions

It is quantified that in the DT operating condition, the mass flow rate of R22 at a 7% undercharged condition is 14.8% lower than that of the optimum condition, whereas the mass flow rate for the M20 refrigerant mixture is 8.7% lower than that of the optimum condition. The mass flow rate of R22 at a 7% overcharged condition is 4.2% higher than that of the optimum condition, whereas the mass flow rate of the M20 refrigerant mixture is 1% higher than that of the optimum condition for the same operating condition. The refrigerant flow rate increased with an addition of the charge amount at fixed air temperatures entering the condenser and evaporator. It is also found that in all the tested operating conditions the mass flow rate of the M20 refrigerant mixture is lower than that of R22.

In general, the refrigerant flow rate through the capillary tube is strongly dependent on the condensing pressure, while it is insensitive to the evaporating pressure due to chocking (Kuehl *et al.*, [19]). As the air temperature entering the condenser is increased, the mass flow rate passing through the capillary tube also increases, because the pressure difference between the capillary tube inlet and outlet increases. The same results were reported by Farzad *et al.* [8] and Choi *et al.* [9]. It is observed that the mass flow rate of R22 in 1400 g, DT operating condition is 5.7% lower than that of the ETB condition, where as the mass flow rate of the M20 refrigerant mixture in 1201 g DT operating condition is 3.1% lower than that of the ETB operating condition.

Figure 8 shows the variation of R22 and the M20 refrigerant mixture evaporator inlet temperature under different refrigerant charge levels and outdoor conditions. It is observed that the evaporator inlet temperature increases with an increase of the charge levels and outdoor conditions. Beyond the optimum charge level the superheat is also reduced as the refrigerant charge increased due to a rise of the mass flow rate, which leads to saturation at the end of the evaporator and the possible introduction of wet vapor into the compressor.

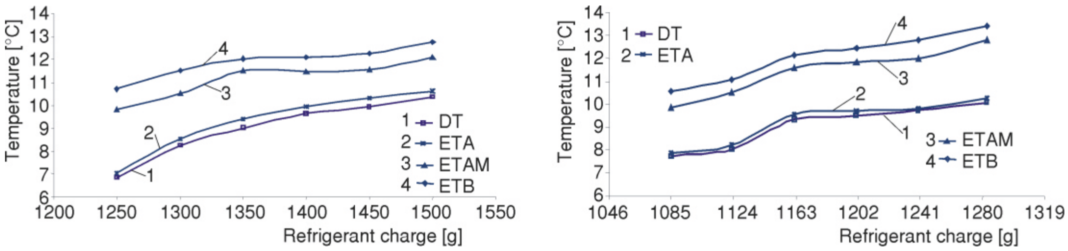


Figure 8. Variation of the R22 and M20 evaporator inlet temperature under different refrigerant charge quantities

The evaporator inlet temperature of R22 when 7% undercharged is 11.8% lower than that of the optimum condition whereas for the M20 refrigerant mixture the evaporator inlet temperature is 12.5% lower than that of the optimum condition in the DT operating condition. The evaporator inlet temperature of R22 at a 7% overcharged condition is 6.3% higher than that of the optimum condition whereas for the M20 refrigerant mixture the evaporator inlet temperature is 4.9% higher than that of the optimum condition in the same operating condition.

Figure 9 shows the variation of R22 and the M20 refrigerant mixture evaporator outlet temperature under different refrigerant charge levels and outdoor conditions. It was observed that the evaporator outlet temperatures of both the refrigerants are decreased in both the undercharged and overcharged conditions. It was inference that the evaporator outlet temperature of the M20 refrigerant mixture is slightly higher; due to this the DBT/WBT difference of air at the

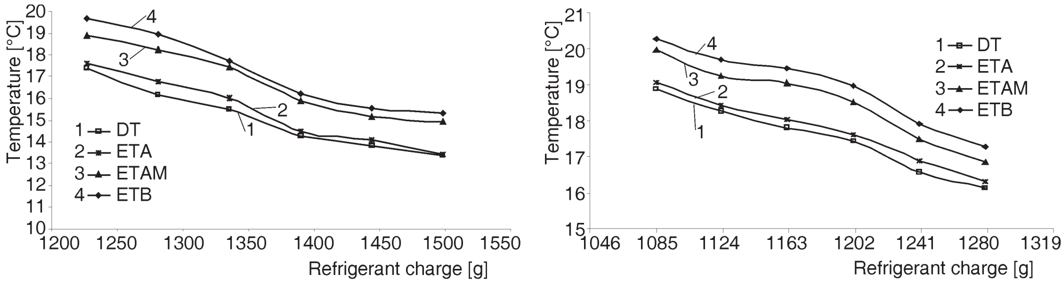


Figure 9. Variation of the R22 and M20 evaporator outlet temperature under different refrigerant charge quantities

outlet of the evaporator may be lowered than that of R22. This is attributes the lower refrigerating capacity of the M20 refrigerant mixture than that of R22.

The evaporator outlet temperature of R22 when 7% undercharged is 10.9% higher than the optimum condition whereas for the M20 refrigerant mixture the evaporator outlet temperature is 4.5% higher than that of the optimum condition in the DT operating condition. The evaporator outlet temperature of R22 at a 7% overcharged condition is 5.9% lower than that of the optimum condition, whereas the evaporator outlet temperature of the M20 refrigerant mixture is 7.5% higher than that of the optimum condition in the same operating condition.

Figure 10 shows the variation of R22 and the M20 refrigerant mixture evaporator inlet pressure under different refrigerant charge quantities and outdoor conditions. It is observed that the evaporator inlet pressure increases with an increase in the charge levels and outdoor conditions in undercharged conditions.

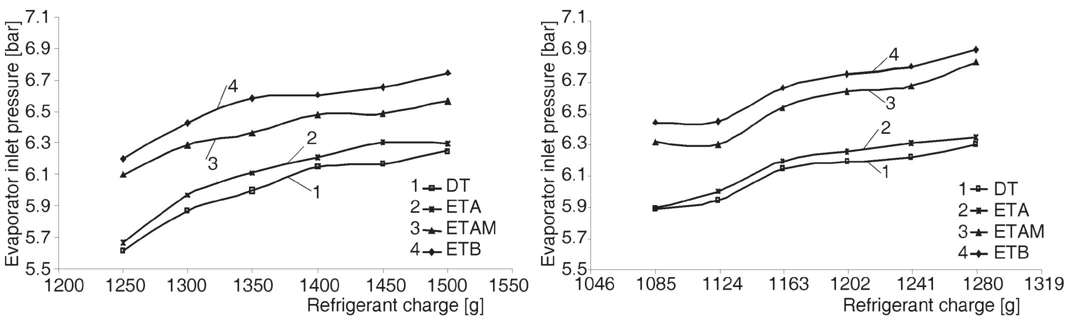


Figure 10. Variation of R22 and M20 evaporator inlet pressure under different refrigerant charge quantity

In the DT operating condition, the evaporator inlet pressure of R22 when 7% undercharged is 4.4% lower than that of the optimum condition, whereas for the M20 refrigerant mixture the evaporator inlet pressure is 3.9% lower than that of the optimum condition. The evaporator inlet pressure of R22 in a 7% overcharged condition is 1.7% higher than that of the optimum condition, whereas the evaporator inlet pressure of the M20 refrigerant mixture is 1.8% higher than the optimum condition in the same operating condition. In general, the increase of evaporator inlet pressure beyond the optimum condition will decrease the temperature difference between the refrigerant and air in overcharged condition. This is attributes the lower DBT/WBT difference across the evaporator and lower refrigerating capacity in the evaporator.

Figure 11 shows the variation of R22 and the M20 refrigerant mixture condenser inlet pressure under different refrigerant charge quantities and outdoor conditions. It is observed that the condenser inlet pressure is increased with an increase in the refrigerant charge and outdoor conditions. The main effects of the increased charge beyond the optimum, are the increased condensing pressure and the increased sub cooling due to the accumulation of refrigerant. Moreover, the temperature difference between the refrigerant and air is decrease in condenser and the condenser load decreases in overcharged condition. The higher condensing pressure leads to severe problems in the compressor life also. The condenser inlet pressure of R22 when 7% undercharged is 2.4% lower than that of the optimum condition, whereas for the M20 refrigerant mix-

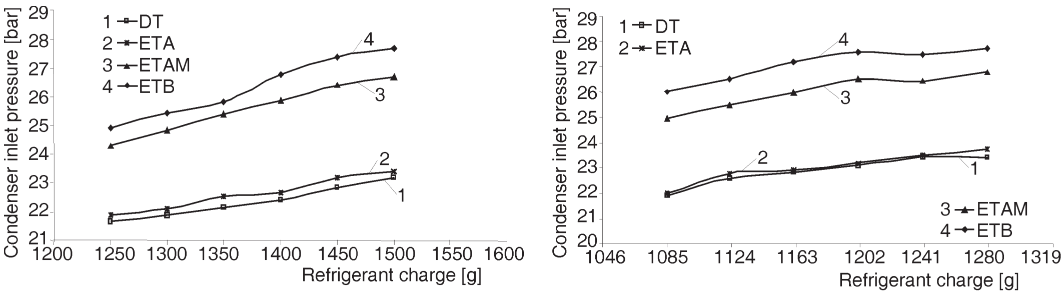


Figure 11. Variation of R22 and M20 condenser inlet pressure under different refrigerant charge quantities

ture the condenser inlet pressure is 2.1% lower than that of the optimum condition in the DT operating condition. The evaporator inlet pressure of R22 at a 7% overcharged condition is 3.4% higher than that of the optimum condition, whereas the evaporator inlet pressure of the M20 refrigerant mixture is 1.3% higher than that of the optimum condition in the same operating condition.

Figure 12 shows the variation of R22 and the M20 refrigerant mixture condenser inlet temperature under different refrigerant charge quantities and outdoor conditions. It is observed that the condenser inlet temperature is decreased with an increase in the refrigerant charge and outdoor conditions. The main effects of the decreased charge level below the optimum are the increased condensing temperature and the decreased sub cooling due to the scarcity of the refrigerant. The decreased sub cooling and higher condenser temperature leads to higher vapor en-

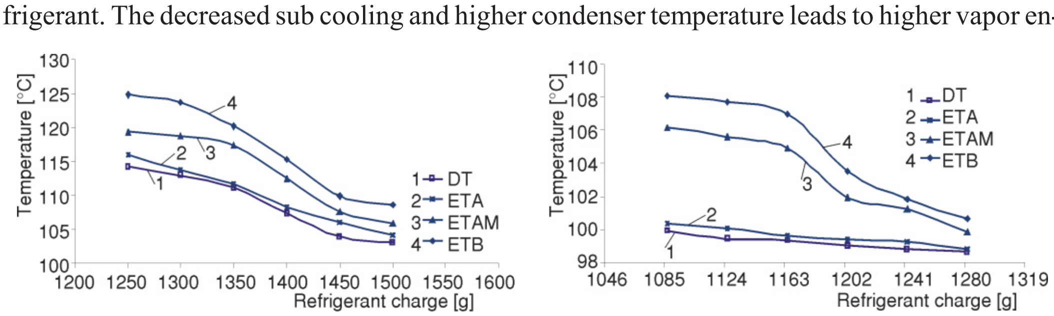


Figure 12. Variation of the R22 and M20 condenser inlet temperature under different refrigerant charge quantities

try into the evaporator. Due to this, the major part of the evaporator is occupied with superheated refrigerants in undercharged conditions. The condenser inlet temperature of R22 when 7% undercharged is 5% higher than the optimum condition whereas the condenser inlet temperature of R22 in a 7% overcharged condition is 4% lower than the optimum in the DT operating condition. But, for the M20 refrigerant mixture the condenser inlet temperature when 6.5% undercharged is 1% higher than the optimum condition whereas the condenser inlet temperature of R22 in a 6.5% overcharged condition is 1% lower than the optimum in the DT operating condition.

Influences of refrigerant charge and outdoor condition on compressor power

Figure 13 shows the variation in the power consumption of R22 and the M20 refrigerant mixture under different refrigerant charge levels and outdoor conditions. It is found that even in over charged conditions, the power consumption of the system increases due to a rise in the refrigerant flow rate and compression ratio. The increase of the suction pressure and temperature in the compressor with refrigerant charge, yields a lower specific volume and higher mass flow rate. Therefore, the compressor power consumption slowly but continuously increases with the refrigerant charge. The power consumption of R22 when 7% undercharged is 3.6% lower than that of the optimum charge condition, whereas for the M20 refrigerant mixture the power consumption is 3.2% lower than that of the optimum in the DT operating condition. It is also found that the power consumption of R22 in a 7% overcharged condition is 1.5% higher than that of the optimum condition, whereas the power consumption of M20 refrigerant mixture is 4.3% higher than that of the optimum in the DT operating condition.

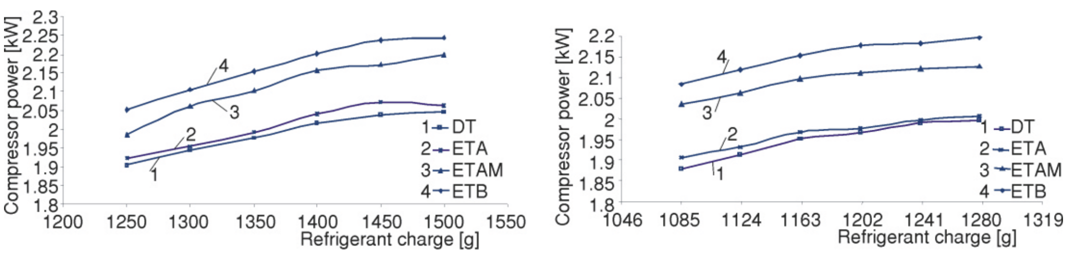


Figure 13. Variation of the R22 and M20 compressor power under different refrigerant charge quantities

The power consumption of the M20 refrigerant mixture is lower than that of R22 in all the charge levels and outdoor conditions, since the vapor specific volume of the M20 refrigerant mixture is lower than that of R22. The properties of the R22 and M20 refrigerant mixture were taken from REFPROP.

Influences of the refrigerant charge and outdoor condition on COP

Figure 14 shows the variation of R22 and the M20 refrigerant mixture COP under different refrigerant charge levels and outdoor conditions. Due to the lower mass flow rate and refrigerating effect as explained earlier, the COP of the system is lower than the optimum charge level in

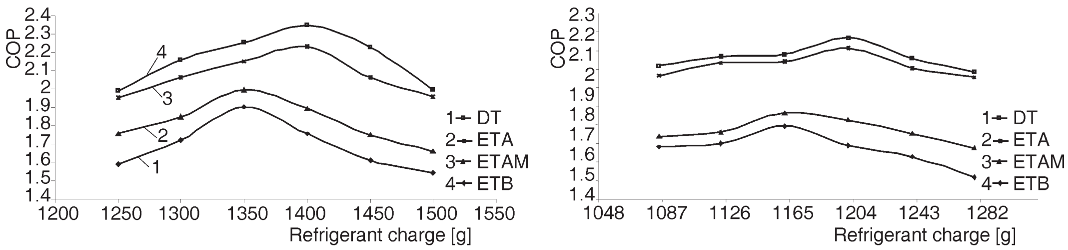


Figure 14. Variation of the R22 and M20 COP under different refrigerant charge quantities

undercharged conditions. The COP is further reduced than the optimum charge level, due to an increase in the pressure ratio and decrease in the refrigerating capacity in overcharged conditions.

It is also observed that as the air temperature entering the condenser increased, the COP significantly dropped at all charge amounts due to higher power consumption, lower refrigerating capacity and higher pressure ratio. It is observed that the COP of R22 in the 1400 g DT operating condition is 25.2% higher than that of the ETB operating condition, which shows the influence of outdoor conditions on the system performance. The outdoor conditions increase, the system performance decreases due to lower refrigerating capacity and higher power consumption.

It has been observed that with a change in refrigerating capacity of the M20 refrigerant mixture, the COP is gradually reduced with an increase of the charge amount, due to a continuous increase of power consumption, which is caused by a rise in the pressure ratio and a higher friction loss in the compressor. However, the COP degradation of the M20 refrigerant mixture system with respect to the charge amount in overcharged conditions is a little less pronounced than that of the R22 system, because the rise of the evaporator inlet temperature in the M20 refrigerant mixture capillary tube system is lower than that of the R22 capillary tube system.

The COP of R22 when 7% undercharged is 7.9% lower than that of the optimum condition, whereas for the M20 refrigerant mixture the COP is 3.9% lower than optimum in DT operating condition. It is also found that the COP of R22 in a 7% overcharged condition is 15.5% lower than that of the optimum, whereas the COP of the M20 refrigerant mixture is 7.7% lower than the optimum in the DT operating condition. The COP of the M20 refrigerant mixture at 1201 g DT operating condition is 25.6% higher than that of the ETB operating condition. It is observed that at the optimum condition the influence of outdoor condition on the system performance with the M20 refrigerant mixture is same as that of R22.

The operation of an air conditioner at elevated ambient temperatures inherently results in a lower COP. The COP relation, COP = $T_{\text{evaporator}}/(T_{\text{condenser}} - T_{\text{evaporator}})$, indicates that the COP decreases when the condenser temperature increases at a constant evaporation temperature. This theoretical indication derived from the reversible cycle is valid for all refrigerants. Due to this, the system COP decreases with the increase of condenser air inlet temperature.

Influences of the various operating condition on refrigerating capacity at optimum charge quantity

The performance of the air conditioner with the M20 refrigerant mixture has been compared with the performance of R22 under different indoor and outdoor conditions as prescribed in BIS standard. The BIS standard considers the domestic conditions (including DBT and WBT) as well as international climate conditions (export test conditions). For summer air conditioning, the entire range of DBT and WBT that globally prevail (all round the year), are by and large covered in the various test conditions prescribed in BIS standard. The BIS standard test conditions considered for the study are given in tab. 2.

Table 2. Indoor and outdoor air temperatures as prescribed in the BIS standard

	Test	Type	Indoor		Outdoor	
			DBT [°C]	WBT [°C]	DBT [°C]	WBT [°C]
1	BIS-DT	Capacity rating	27	19	35	30
2	BIS-DT	Maximum	35	24	46	27
3	BIS-ETA	Capacity rating	27	19	35	24
4	BIS-ETA	Maximum	32	23	43	26
5	BIS-ETB	Capacity rating	29	19	46	24

Figure 15 shows the comparison of the refrigerating capacity of both R22 and the M20 refrigerant mixture under the different operating conditions. The refrigerating capacity of R22 3.302 kW in the higher operating condition (BIS DTM), where as the M20 refrigerant mixture gives 3.159 kW in that operating conditions. In the maximum operating condition (ETA-M and DT-M) the M20 refrigerant mixture had 6.68% and 4.34% lower refrigerating capacity with respect to R22. The M20 refrigerant mixture had 6.644%, 9.69%, and 4.18% lower refrigerating capacity than R22 in capacity rating test conditions (DT, ETA, and ETB). Due to the higher vapor fraction at the inlet of the evaporator and the lower mass flow rate of M20, the refrigerating capacity is lower than R22 refrigeration system. However for a retrofit condition this drop in capacity is not very serious.

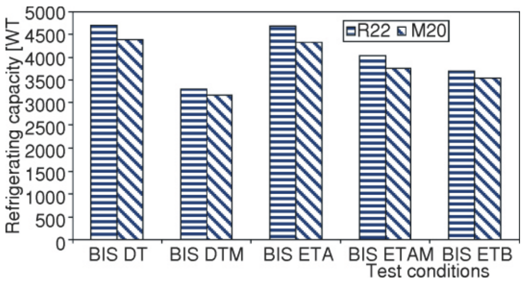


Figure 15. Refrigerating capacities of R22 and M20 under different test conditions

Uncertainty analysis

Experimentation using R22 and M20 refrigerant mixture included the measurement of air temperatures (DBT and WBT), pressure difference across the nozzles, power consumption, refrigerant temperature and pressures across each component of the systems and refrigerant mass flow rate. Refrigerating capacity and actual COP were calculated based on the measured parameters. The uncertainties (calculated using standard method – square root of the sum of squares) in refrigerating capacity, compressor power and COP were in the range 1.0% to 1.5%, 0.8% to 2.2%, and 2.5% to 4.0%, respectively.

Conclusions

A 5.25 kW window air conditioner with capillary tube expansion was used to compare the performance of the M20 refrigerant mixture with R22 over a range of refrigerant charge quantities and outdoor conditions. The refrigerant charge in the unit was systematically varied to determine and quantify its effect on the mass flow rate, refrigerating capacity, evaporator, and condenser inlet pressure, pressure ratio, power consumption, and COP. Based on the experimentation the following conclusions are drawn.

The refrigerating capacity of R22 in a 7% undercharged condition is 11.3% lower than that of the optimum, whereas the refrigerating capacity of the M20 refrigerant mixture is 6.9% lower than that of the optimum. It is also found that the refrigerating capacity of R22 in a 7% overcharged condition is 13.8% lower than that of the optimum, whereas the refrigerating capacity of the M20 refrigerant mixture is 6.5% lower than that of the optimum for the DT operating condition.

The power consumption of R22 when 7% undercharged is 3.6% lower than that of the optimum charge condition, whereas for the M20 refrigerant mixture the power consumption is 3.2% lower than that of the optimum in the DT operating condition. It is also found that the power consumption of R22 in a 7% overcharged condition is 1.5% higher than that of the optimum condition, whereas the power consumption of M20 refrigerant mixture is 4.3% higher than that of the optimum in the DT operating condition.

The COP of R22 when 7% undercharged is 7.9% lower than that of the optimum condition, whereas for the M20 refrigerant mixture the COP is 3.9% lower than optimum in DT operating condition. It is also found that the COP of R22 in a 7% overcharged condition is 15.5% lower than that of the optimum, whereas the COP of the M20 refrigerant mixture is 7.7% lower than the optimum in the DT operating condition.

The degradation of performance of the M20 refrigerants was lower for under charging than for overcharging than that of R22. It was found that the cooling capacity, compressor power, mass flow rate, and COP of the capillary system show dependent on refrigerant charge and outdoor conditions. M20 being considered as an alternative for R22, this study is a useful tool for analyzing the effect of refrigerant charge and outdoor conditions in retrofitted condition.

Acknowledgments

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Acronyms

AHU – air handling unit
AREP – alternative refrigerant evaluation program
ASHRAE – American society of heating, refrigerating and air-conditioning engineers
BIS – Bureau of Indian standards
CFC – chlorofluorocarbon
COP – coefficient of performance
DBT – dry bulb temperature, [°C]
DT – domestic capacity rating test condition
EEV – electronic expansion valve
ETA – export test A capacity rating test condition
ETAM – export test A maximum operating test condition
ETB – export test B capacity rating test condition
GWP – global warming potential
HC – hydrocarbon
HCFC – hydrochlorofluorocarbon
HFC – hydrofluorocarbon

LPG – liquefied petroleum gas
MFM – mass flow meter
ODP – ozone depletion potential
PC – personal computer
POE – polyolester
RTD – resistance temperature detector
SG – sight glass
TR – ton of refrigeration
UUT – unit under test
WBT – wet bulb temperature, [°C]

Subscripts

a – air
ac – actual
c – compressor
d – discharge
mea – measured
s – suction

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Appendix

Specifications of the components used in window air conditioner

Compressor

Hermetically sealed reciprocating type
Make: Kirloskar Copeland Ltd.
Model: KCH522HAE
Lubricant: Mineral oil
Cooling capacity: 5250 W (1.5 TR)
Rated power: 2000 W

Condenser

Plate fin and tube air cooled condenser
Number of rows: 3
Material: Copper
Copper tube size: diameter: 9.5 mm
Length per tube: 525 mm
Number of tubes: 48
Width: 87 mm
Height: 440 mm
Fins per inch: 12

Evaporator

Plate fin and tube – Dry expansion type
Number of rows: 3
Number of circuit: 2
Material: Copper
Copper tube size: diameter – 9.5 mm
Lenght per tube: 366 mm
Number of tubes/circuit: 24
Width: 87 mm
Height: 406 mm
Fins per inch: 12
Tube layout: staggered arrangement

Drier

Molecular sieve type

Capillary tube

R22:
Bore 1.542 mm and length 0.8382 m

M20 refrigerant mixture:
Bore 1.524 mm and length 0.7112 m

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