CHAPTER 5

RESULTS AND DISCUSSION

5.1 INTRODUCTION

The performance of the fin-and-tube evaporator is simulated for R22 and the M20 refrigerant mixture specific to the operating conditions of the window air conditioner. The simulation results enable to predict the refrigerating capacity that could be realized in the air conditioner, because the upstream and downstream flow conditions prevailing in the air conditioner are imposed as boundary conditions for the simulation.

The performance and heat transfer experimental studies have been conducted as discussed in chapter 4. A systematic comparison has been made between R22 and the M20 refrigerant mixture with reference to system performance parameters, such as refrigerating capacity, work of compression, COP etc. The experimental heat transfer coefficients are also compared with the theoretical predictions at all possible levels.

5.2 SIMULATION OF FIN-AND-TUBE EVAPORATOR

Simulations are performed for R22 and the M20 refrigerant mixture in the BIS DT capacity rating operating condition. The mass flow rate, pressure and temperature of the refrigerant and air that are needed as input for the simulation study are taken from the experimental data. The thermophysical properties of the refrigerants are taken from REFPROP 7.0. The output parameters are the pressure drop, saturation temperature, profile of the
temperature difference between the refrigerant and the air, the heat transfer rate, and air temperature in each row. These results are presented and discussed in this section.

The actual refrigerating capacity obtained from the experimental study carried out as explained in chapter 4, is used for the validation of the simulated heat transfer rate. The comparison of the predicted heat transfer rate with the experimental heat transfer rate is also discussed in this section.

The simulation of the fin-and-tube evaporator is carried out using ε-NTU section-by-section method for R22 and the M20 refrigerant mixture. As discussed in chapter 3, the general equations, pressure drop correlations and heat transfer coefficients for air, are in use while executing the program. MATLAB 6.0 is used to write the code for R22 and the M20 refrigerant mixture simulation. The simulations are carried out for the operating conditions of the air and the refrigerant as given in Table 3.1. In this simulation model, Lavin and Young (1965), Shah (1982), Gungor and Winterton (1986 and 1987), Jung et al (1989), Liu and Winterton (1991), and Choi et al (2000) heat transfer coefficient correlations are analyzed for the fin- and tube evaporator performance with R22, it being necessary to implement them in the model to facilitate comparisons and decide which one works best for the evaporator performance studied. A number of these correlations, found in the reviewed literature, have been implemented in the simulation model and studied. The Lavin and Young (1965) correlation has been modified with the present experimental data for R22 and the Lavin and Young modified correlation has been evolved. Their predicted heat transfer rate has been compared with the experimental heat transfer rate of R22 as given in Table 5.1. The results obtained are referred to the heat exchangers studied for this particular case, which means that the solution chosen is not necessarily better than the others, but is closer to the experimental measurements. The
predicted heat transfer rate is validated against the experimental heat transfer rate. It is observed from the predicted results that the heat transfer correlation that works best for the evaporator studied, is the Lavin and Young modified correlation. It is found that its heat transfer rate is 4.5% lower than that of the experimental results. Even though the considered correlations are evolved based on the R22 and a few other refrigerants’ heat transfer coefficients, they have over-predicted the experimental results.

Table 5.1 Predicted performance of the fin-and-tube evaporator with R22 using different correlations

<table>
<thead>
<tr>
<th>Correlation</th>
<th>Vapor fraction</th>
<th>T_{sat} K</th>
<th>T_{su} K</th>
<th>Heat transfer rate (W)</th>
<th>Experimental heat transfer rate (W)</th>
<th>Deviation %</th>
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<tr>
<td>Lavin and Young (1965)</td>
<td>1.00</td>
<td>277.8</td>
<td>283.9</td>
<td>2496.9</td>
<td>2354.6</td>
<td>+5.7</td>
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<tr>
<td>Shah (1982)</td>
<td>1.00</td>
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<td>285.1</td>
<td>2527.8</td>
<td></td>
<td>+6.8</td>
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<tr>
<td>Gungor and Winterton (1986)</td>
<td>1.00</td>
<td>278.3</td>
<td>285.2</td>
<td>2574.4</td>
<td>2354.6</td>
<td>+8.5</td>
</tr>
<tr>
<td>Gungor and Winterton (1987)</td>
<td>0.99</td>
<td>278.4</td>
<td>289.2</td>
<td>2548.1</td>
<td></td>
<td>+7.4</td>
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<tr>
<td>Liu and Winterton (1991)</td>
<td>0.78</td>
<td>278.2</td>
<td>…………</td>
<td>1611.9</td>
<td></td>
<td>-31.5</td>
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<tr>
<td>Choi et al (2000)</td>
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<td>289.8</td>
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<tr>
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<td>1.00</td>
<td>277.8</td>
<td>284.6</td>
<td>2247.9</td>
<td></td>
<td>-4.5</td>
</tr>
</tbody>
</table>

The Lavin and Young modified correlation has been chosen as the one to be finally implemented in the model because it covers the whole range of real appliance evaporator vapour fraction, heat flux and mass velocity ranges. The selected correlation has been implemented in the model and the simulated results are compared with those of the M20 refrigerant mixture in section 5.2.1.
For the M20 refrigerant mixture, Gungor and Winterton (1986 and 1987), Liu and Winterton (1991), Jung et al (1993), Wattelet et al (1994), and Choi et al (2000) heat transfer coefficient correlations are analyzed for the performance of the fin-and-tube evaporator. The above correlations have been implemented in the simulation model and their predicted heat transfer rate has been compared with the experimental heat transfer rate of the M20 refrigerant mixture as given in Table 5.2. For the simulation of the M20 refrigerant mixture the conditions of air and the refrigerant inlet pressure are considered as given in Table 3.1. The Gungor and Winterton (1987) correlation has been modified with the present experimental data for M20 and the Gungor and Winterton modified correlation has been evolved. The predicted heat transfer rate of M20 refrigerant mixture with all the considered correlations is validated against the experimental heat load. It is observed from the predicted results that the heat transfer correlation that works best for the evaporator studied is the Gungor and Winterton modified correlation, as its heat transfer rate is 5.2% lower than that of the experimental results.

Based on the results shown in Table 5.2, it is observed that the Gungor and Winterton modified correlation predicted the heat transfer rate closer to the experimental results than the other correlations. Hence, the Gungor and Winterton modified correlation is evolved based on the present experimental heat transfer coefficient vaporizing in a fin-and-tube evaporator. The Gungor and Winterton modified correlation has been chosen as the one to be finally implemented in the model, because it covers the whole range of qualities. The selected correlation has been implemented in the model and the simulated results are compared with those of the M20 experimental result in section 5.2.1.
Table 5.2  Predicted performance of the fin-and-tube evaporator with the M20 refrigerant mixture using different correlations

<table>
<thead>
<tr>
<th>Correlation</th>
<th>Outlet conditions</th>
<th>Experimental heat transfer rate (W)</th>
<th>Deviation (%)</th>
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<tr>
<td></td>
<td>Vapor fraction</td>
<td>$T_{sat}$ (K)</td>
<td>$T_{su}$ (K)</td>
</tr>
<tr>
<td>Gungor and Winterton</td>
<td>1.00</td>
<td>276.6</td>
<td>286.3</td>
</tr>
<tr>
<td>(1986)</td>
<td></td>
<td></td>
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<tr>
<td>Gungor and Winterton</td>
<td>1.00</td>
<td>278.2</td>
<td>288.3</td>
</tr>
<tr>
<td>(1987)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Liu and Winterton</td>
<td>0.918</td>
<td>273.6</td>
<td>.......</td>
</tr>
<tr>
<td>(1991)</td>
<td></td>
<td></td>
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</tr>
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<td>Jung et al (1993)</td>
<td>1.00</td>
<td>276.9</td>
<td>287.9</td>
</tr>
<tr>
<td>Wattelet et al (1994)</td>
<td>1.00</td>
<td>276.4</td>
<td>283.1</td>
</tr>
<tr>
<td>Gungor and Winterton</td>
<td>1.00</td>
<td>277.2</td>
<td>282.8</td>
</tr>
<tr>
<td>modified</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

5.2.1 Comparison of Simulated Results

The most suitable correlation namely Lavin and Young for R22 and Gungor and Winterton for M20 have been implemented in the simulation model and the performance of fin-and-tube evaporator has been predicted.

Figure 5.1 shows the pressure variations of R22 and the M20 refrigerant mixture along the refrigerant flow path. The predicted pressure at the exit of the M20 refrigerant mixture is 2.6% lower than that of R22. It is observed that the predicted outlet pressure of the M20 refrigerant mixture is lower than that of R22, even though the M20 refrigerant mixture enters at a higher pressure. The liquid and vapor density of the M20 refrigerant mixture is less than that of R22 (REFPROP 7.0). Liquid and vapor densities are important parameters in pressure drop correlations; owing to this the pressure
drop of the M20 refrigerant mixture is higher than that of R22. The higher pressure drop can lead to lower system performance.

Figure 5.2 shows the predicted saturation temperatures of R22 and the M20 refrigerant mixture along the refrigerant flow path. For R22 the total temperature drop including the bend effect in the two phase region is 1.78 °C. However, for the M20 refrigerant mixture the temperature glide is accompanied with a drop in temperature due to the pressure drop. Thus, the net temperature rise is only 4.28 °C, while the glide corresponding to the inlet pressure is 7.1°C.

Figure 5.1 Simulated pressures of the refrigerants along the refrigerant flow path
Figure 5.2 Simulated temperature of the refrigerant along the refrigerant flow path

Figure 5.3 shows the simulated temperature difference between the refrigerant and the air between the rows for both the refrigerants. In this figure F, S and T represent first, second and third row respectively. For the same evaporator inlet air temperature, the M20 refrigerant mixture shows 10.1% to 20.1% higher temperature difference than that of R22 in the I row, while in the II and III row the temperature difference between the air and the refrigerant is initially lower, and higher after some sections than that of R22. In III row the temperature difference is decreased due to superheating occurred in the later part of the tubes. In general, the higher temperature difference offers better refrigerating capacity in an evaporator. The lower temperature drop experienced by the air indicates a poor overall heat transfer coefficient of the evaporator when the M20 refrigerant mixture is used in the existing system. Since air flow conditions at the inlet of the evaporator remain the same, this directly points out to the poor heat transfer coefficient of the M20 refrigerant mixture for the prevailing flow conditions.
Figure 5.3  Simulated temperature differences between the refrigerant and the air

Figure 5.4 shows the comparison of the simulated heat transfer rate in each row of both the refrigerants. In the I row, the higher heat transfer rate is obtained for the M20 refrigerant mixture, since the temperature difference between the refrigerant and the air in I row is higher than that of R22. The heat transfer rate of the M20 refrigerant mixture in the first row is 4.5% higher than that of R22. In II and III rows the heat transfer rate of the R22 is 18.5% and 19.6% respectively, higher than that of the M20 refrigerant mixture. As shown in Figure 5.2, the M20 refrigerant mixture has a temperature glide of 4.2 K, even though the pressure drop occurs while flowing through the tube. The temperature difference between the air and refrigerant is decreased in the II and III rows, due to the increased equilibrium temperature of the M20 refrigerant mixture. Due to this, the heat transfer rate of the M20 refrigerant mixture in II and III rows is lower than that of R22.
Figure 5.4  Comparison of the simulated heat transfer rate in each row

Figure 5.5 shows the comparison of the simulated and experimental heat transfer rate in a fin-and-tube evaporator in BIS DT capacity rating operating condition as given in Table 3.1. It is observed that the simulated heat transfer rate of the R22 and M20 refrigerant mixture is 4.5 and 5.2% respectively, lower than that of experimental results. It is reported in section 5.3.7.7 that the Gungor and Winterton modified correlation predicted the heat transfer coefficients of the M20 refrigerant mixture in the deviation range of -25% to +15%. It is also reported in section 5.3.7.7 that the Lavin and Young modified correlation predicted the experimental results in the deviation range of -25% to +20%. This is inline with the difference in heat transfer rate predicted.
Figure 5.5 Comparison of the simulated and experimental heat transfer rate in the evaporator

Figure 5.6 shows the comparison of the simulated air temperature in each row of the fin-and-tube evaporator for R22 and the M20 refrigerant mixture. The air enters in the same condition into both the evaporators, while the leaving air temperature in each row does not remain the same. From the figure it is observed that the temperature of the leaving air in each row of the M20 refrigerant mixture is slightly higher than that of R22. This is attributed to the increasing trend of the equilibrium temperature of the M20 refrigerant mixture. The slightly higher air temperature of the M20 refrigerant mixture indicates a lower refrigerating capacity than that of R22.
Figure 5.6 Comparison of the temperature of air leaving each row

5.3 EXPERIMENTAL STUDY

The performance of the air conditioner with the M20 refrigerant mixture has been experimentally studied and compared with that of R22 under different indoor and outdoor conditions. The following performance parameters, namely, Refrigerating capacity, compressor power, COP, refrigerant mass flow rate, condenser inlet pressure, pressure ratio, compressor dome temperature, and per day energy consumption are compared.

5.3.1 Capillary and Charge Optimization Test

The results of these tests are presented in section 4.4.1 itself as it forms the basis of conducting the rest of the experimentation. The capillary length and charge quantity were optimized for the maximum COP.
5.3.2 Capacity Assessment Test

The results of the capacity assessment test conducted with the optimized capillary and charge quantity at the selected operating conditions prescribed in BIS and ASHRAE standards are reported here.

Figure 5.7 shows the condenser inlet pressure of R22 and the M20 refrigerant mixture under the different operating conditions considered. The condenser inlet pressure of the M20 refrigerant mixture is higher in the range of 1% to 4.6% than that of R22 under all operating conditions. The discharge pressure of the M20 refrigerant mixture is 19.54 bar in a lower operating condition (TB) and 28.1 bar in a higher operating condition (BIS DTM). The discharge pressure at the maximum operating condition has not exceeded the manufacturer’s design limit and hence retrofitting can be done.

![Condenser inlet pressures of R22 and the M20 refrigerant mixture under different test conditions](image)

**Figure 5.7** Condenser inlet pressures of R22 and the M20 refrigerant mixture under different test conditions
Figure 5.8 shows the pressure ratio comparison of R22 and the M20 refrigerant mixture under different operating conditions. In all operating conditions the pressure ratio of the M20 refrigerant mixture is more or less the same as that of R22, but in the TB condition the pressure ratio is 1.5% higher than that of R22. In all operating conditions the evaporator pressure of the M20 refrigerant mixture is higher than that of R22; due to this the pressure ratio of the M20 refrigerant mixture is more or less the same as that of R22. Hence the M20 system characteristics would be more or less similar to that of R22 system.

![Pressure ratio chart](image)

**Figure 5.8** Pressure ratio of R22 and the M20 refrigerant mixture under different test conditions

Figure 5.9 shows the refrigerant mass flow rate of R22 and the M20 refrigerant mixture under different operating conditions. The mass flow rate of R22 varies in the range of 91.8 kg hr\(^{-1}\) to 110.4 kg hr\(^{-1}\), but for the M20 refrigerant mixture the mass flow rate varies in the range of 87.2 kg hr\(^{-1}\) to 99.2 kg hr\(^{-1}\). The M20 refrigerant mass flow rate is lower in the range of 5.0%
to 11.3% than that of R22. The lower mass flow rate can be attributed to its higher specific volume as compared to R22 for which the compressor was originally designed. However in spite of low flow rates received the drop in system refrigerating capacity is very marginal.

![Graph showing refrigerant mass flow rate](image)

**Figure 5.9** Mass flow rate of R22 and the M20 refrigerant mixture under different test conditions

Figure 5.10 shows the comparison of the refrigerating capacity of both R22 and the M20 refrigerant mixture under different operating conditions. The refrigerating capacity of R22 is 4.918 kW in the lower operating condition (TB) and 3.302kW in the higher operating condition (BIS DTM), whereas the M20 refrigerant mixture gives 4.622 kW and 3.159 kW respectively in those operating conditions. Experimental results show that the M20 refrigerant mixture has 6.0% and 7.5% lower refrigerating capacity with respect to R22 in TB and TA test conditions respectively. In the maximum operating condition (ETAM and DTM), the M20 refrigerant mixture has 6.7% and 4.3% lower refrigerating capacity with respect to R22. The M20
refrigerant mixture has 6.6%, 7.7% and 4.2% lower refrigerating capacity than R22 in the 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 the M20 refrigerant mixture, the refrigerating capacity is lower than that of the R22 refrigeration system. However for a retrofit condition this drop in capacity is not very serious.

![Graph showing refrigerating capacities of R22 and M20](image)

**Figure 5.10  Refrigerating capacities of R22 and M20 under different test conditions**

Figure 5.11 shows the power consumed by the compressor with R22 and the M20 refrigerant mixture in the system. The power consumed by the R22 compressor is in the range of 1853W to 2329W, while that with the M20 refrigerant mixture is in the range of 1787W to 2244W. Under the different operating conditions the power consumed by the M20 refrigerant mixture is in the range of 3.2% to 4.9% lower than that of R22. One of the major reasons for the lower compressor power is the increase in the refrigerant saturation temperature at the suction of the compressor and the decrease in the saturation temperature at the compressor discharge. Therefore, the compressor operates
over a lower ‘temperature lift’, which reduces energy consumption. It is to be noted that the power consumed by the M20 refrigerant mixture did not exceed the limit as prescribed by BIS-1391-1992.

![Bar Chart: Compressor power of R22 and the M20 refrigerant mixture under different test conditions](image)

**Figure 5.11 Compressor power of R22 and the M20 refrigerant mixture under different test conditions**

Figure 5.12 shows the variation of the COP for both R22 and the M20 refrigerant mixture under different operating conditions. The COP of R22 is 2.31 and 2.80 under the TA and TB operating conditions, but the COP of the M20 refrigerant mixture is 2.21 and 2.59, which is 7.8% and 4.4% lower than that of R22. In the capacity rating test conditions (BIS), the COP of the M20 refrigerant mixture is 2.2%, 4.9% and 1% lower in DT, ETA and ETB conditions respectively, than that of R22. In all the conditions studied the COP of the M20 refrigerant mixture is lower in the range of 1% to 7.8% than that of R22; this can be attributed to the lower refrigerating capacity and mass flow rate. However, the difference is very marginal.
From the above results it can be understood that the technical viability of M20 as a substitute for R22 is proved beyond doubt. However the performance of M20 at retrofit condition is marginally lower than that of R22.

Figure 5.12b shows the comparison of 1/COP and 1/Refrigerating capacity of both R22 and M20 under different operating conditions. The 1/COP of both the refrigerants is in increasing trend with increasing 1/Refrigerating capacity. As evident from Figure 5.10 and Figure 5.12a the refrigerating capacity and COP of M20 refrigerant mixture is lower than that of R22 under different operating condition. Due to the higher vapor fraction at the inlet of the evaporator and the lower mass flow rate of the M20 refrigerant mixture, the refrigerating capacity is lower than that of the R22 refrigeration system and as a consequence the COP is lower than that of R22 refrigeration system. Under the different operating conditions the 1/COP of M20 is 1.0% to 4.2% higher than that of R22.

Figure 5.13 shows the variation of pressure drop in the evaporator with R22 and M20 refrigerant mixture under different operating conditions. It can be seen that the pressure drop experienced by M20 is lower than that of R22 in the range of 1.4% to 6.0% for the various operating conditions. This can possibly reduce the work of compression as it is evident from Figure 5.11. However it is to be noted that work of compression also depends on the pressure ratio and other thermodynamic properties of the fluid. The reduction in pressure drop could be attributed to the lower liquid density of M20 refrigerant mixture as compared to R22.
(a) COP of R22 and the M20 refrigerant mixture under different test conditions

(b) Comparison of 1/COP and 1/Refrigerating capacity of both R22 and M20 under different operating conditions

Figure 5.12 COP of R22 and the M20 refrigerant mixture under different test conditions
Figure 5.13  Comparison of the pressure drop in evaporator

The compressor dome temperature plays a major role in the compressor performance and life. Figure 5.14 shows the compressor dome temperature with R22 and the M20 refrigerant mixture as refrigerant under different operating conditions studied. It is observed that the compressor dome temperature for the M20 refrigerant mixture is higher in the range of 0.5% to 3.8% than that of R22. The maximum dome temperature is experienced at BIS DTM condition. For the M20 refrigerant mixture the maximum dome temperature is 59.0°C, but for R22 the dome temperature is 57.7°C. Even though the compressor dome temperature while using the M20 refrigerant mixture is higher than that of R22, it is well within the upper limit (60°C) specified by the manufacturer.

Figure 5.15 shows the variation of the condenser inlet temperature under different operating conditions. It is observed that the discharge temperatures of R22 are 4.0% to 9.8%, higher in all operating conditions than those of M20 refrigerant mixture system. A lower discharge temperature is useful for better lubricant characteristics and oil stability. At lower temperatures, any failure in the worn rings, acid formation and oil breakdown would be avoided. A lower discharge temperature also indicates a lower motor winding temperature. The result has always been a machine that has
endured, has resulted in excellent valve life, has demanded minimal maintenance and has performed consistently. This means that when M20 is used the reliable life of the compressor is likely to be longer as compared to that of R22.

Figure 5.14 Comparison of compressor dome temperature under different operating conditions

Figure 5.15 Comparison of condenser inlet temperature under different operating conditions
Experimental results reported by Devotta et al (2005a) that the R407C’s cooling capacity was lower in the range of 2.1 to 7.9%, COP was lower in the range of 7.9 to 13.5% and the compressor power was higher in the range of 6 to 7% than those of R22 in capacity rating operating conditions. But the present study shows that, in the capacity rating test conditions (BIS), the COP of the M20 refrigerant mixture is 2.2%, 4.9% and 1% lower than that of R22 in DT, ETA and ETB conditions respectively. Thus, the M20 refrigerant mixture can be used as an alternative for R22 without any modification in the system other than changing the capillary and charge quantity. Considering R407C, this mixture will be a better retrofit alternative for R22 as R407C is compatible with POE oil only, which is highly hygroscopic and expensive. The M20 refrigerant mixture is compatible with mineral oil and hence POE oil can be dispensed with.

5.3.3 Pull Down Test

Any refrigerant has to be assessed for its cooling rate before being considered as an alternative. Figure 5.16 shows the temperature variation inside the indoor chamber during the pull down test. The pull down time for the M20 refrigerant mixture is 2366sec, and for R22 it is 1786sec. The pull down time for the M20 refrigerant mixture is 24.5% higher than that of R22. The higher pull down time may be attributed to the lower refrigerating capacity prevailing in the evaporator. It is to be recalled that this refrigerant can cool better if the condenser length is increased for its effective condensation and in turn better mass flow rate.

Figure 5.17 shows the compressor power variation during the pull down test. Even though the power consumed by the compressor with the M20 refrigerant mixture is lower than that with R22, the pull down time is higher than that of R22. It is also noted that the power drops instantaneously in the beginning, then rises and then again drops slowly. This rise is due to the influence of the increased local outdoor temperature which is due to the heat
rejected by the condenser of the test unit. However, this is subsequently taken care of by the outdoor room AHU as evinced by the subsequent drop in the compressor power.

Figure 5.16 Variation of temperature with time during pull down test

Figure 5.17 Variation of compressor power with time during pull down test
5.3.4 Per-day Energy Consumption Test

Figure 5.18 shows the comparison of energy consumption of R22 and the M20 refrigerant mixture under different operating conditions. The BIS ETA, BIS ETAM and BIS ETB test conditions are selected as these conditions cover the normal and extreme indoor and outdoor air temperatures. It is observed, that for the M20 refrigerant mixture, the per day energy consumption is higher than that of R22 in the range of 7% to 18.4% under different test conditions. This is because of the lower refrigerating capacity of the M20 refrigerant mixture system.

![Comparison of energy consumption](image)

**Figure 5.18 Comparison of per day energy consumption under different operating conditions**

Jabaraj et al (2006 and 2007) have reported the performance of the same M20 refrigerant as against R22 with enhanced refrigerating capacity, shorter pull down period and lower per day energy consumption. It is noted that the condenser length was increased by 19% to achieve the recommended degree of sub-cooling for the system without increasing the condenser pressure. However, in the present work the same condenser length is maintained as a retrofit measure. Hence, the condensation was not complete leading to vapor entry into capillary. This has resulted in a lower mass flow rate leading to poor heat transfer characteristics, refrigerating capacity, and in
turn, lower COP for the system. Thus, the present work clearly proves the workability of the M20 refrigerant mixture as a retrofit refrigerant with a marginal loss in performance as discussed. Jabaraj et al (2006 and 2007) reported that there was an increase in the refrigerating capacity, which outweighed (because of the higher mass flow rate) the increase in compressor power consumption resulting in higher COP. In the present work, the decrease in refrigerating capacity outweighs the decrease in compressor work (due to the lower mass flow rate) resulting in lower COP. Hence, for the new system it is better to consider enhanced condenser length, though the same condenser can be used for retrofitting.

5.3.5 Performance Over a Range of Charge Quantity

As already discussed it is not unusual to have a deviation in the charge quantity while retrofitting. The performance of a window air conditioner with the M20 refrigerant mixture has been compared with the performance of R22. At each charge level the outdoor conditions were changed as given in Table 5.3 while the indoor condition was maintained at 27°C, 19°C (DBT, WBT). The following performance parameters, namely, refrigerant mass flow rate, evaporator and condenser inlet pressure, evaporator inlet and outlet temperature, condenser inlet temperature, compressor power, Refrigerating capacity and COP are compared.

Table 5.3 Test conditions for performance over a range of charge quantity

<table>
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<tr>
<th>Sl.No.</th>
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<th>Outdoor conditions</th>
</tr>
</thead>
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<td></td>
<td></td>
<td>DBT (°C)</td>
</tr>
<tr>
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<td>BIS DT</td>
<td>Capacity Rating</td>
<td>35</td>
</tr>
<tr>
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<td>BIS ETA</td>
<td>Capacity Rating</td>
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<tr>
<td>4</td>
<td>BIS ETB</td>
<td>Capacity Rating</td>
<td>46</td>
</tr>
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</table>
5.3.5.1 Influences of Refrigerant Charge and Outdoor Condition on Refrigerating Capacity

Figure 5.19 shows the variation of system refrigerating capacity with refrigerant charge quantity of R22 and M20 under different outdoor conditions. In all practical cases, the cooling capacity is proportional to the refrigerant mass flow rate. In an undercharged condition, the increase in the evaporator outlet temperature as seen in Figure 5.20 yields a higher specific volume at the compressor suction and results in a lower mass flow rate as seen in Figure 5.21. In an undercharged condition the low evaporator inlet pressure (shown in Figure 5.23) 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 the evaporator efficiency decreases drastically leading to lower refrigerating capacity in undercharged conditions. It is evident from Figure 5.20 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. The reduction in refrigerating capacity can be attributed to the increase in condenser pressure that can cause excessive flashing in the capillary leading higher dryness fraction at the inlet of evaporator. The refrigerating capacity is also increased because of higher mass flow rate but beyond the optimum charge level it can be inferred that the drop in refrigerating capacity due to excessive flashing in the capillary outweighs the effect due to increased mass flow rate and hence the refrigerating capacity drops. The same e results were also reported by O’Neal and Farzad (1990), Farzad and O’Neal (1991) and Choi and Kim (2002 and 2004).
Figure 5.19 Variation of refrigerating capacity with respect to refrigerant charge

As compared to R22 the refrigerating capacity reduction in the M20 refrigerant mixture system in the overcharged condition happens slowly, because the increased mass flow rate 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 and Kim (2004).

It is also observed that the optimum charge for test conditions DT and ETA is 1400g while for ETAM and ETB test conditions it is 1350g. Since, the outdoor conditions for ETAM and ETB are higher than those of DT and ETA. It is observed that the maximum cooling capacity is dependent on
the air temperature entering the condenser. As the condenser inlet air temperature increases the rate of condensation decreases which would increase the condenser pressure resulting in a higher mass flow rate of the refrigerant as shown in Figure 5.21. However the higher condensing pressure causes more flashing of refrigerant in the capillary resulting in a higher dryness fraction at the evaporator inlet which drastically reduces the refrigerating effect. The drop in refrigerating effect outweighs the rise in mass flow rate resulting in a reduced refrigerating capacity for higher condensed inlet air temperature conditions namely ETAM and ETB. The above phenomenon is also true with increasing charge quantity. Thus because of the combined effect as the charge quantity is increased the optimum level is attained at a lower charge level itself when the condenser inlet air temperature is higher.

The refrigerating capacity of the M20 refrigerant mixture realized the peak in the charge quantity of 1201g for the DT and ETA test condition, whereas for ETAM and ETB, the optimum refrigerating capacity was realized at 1163g itself. Thus, the charge that produced optimal performance at one temperature produced a sub-optimal performance at other ambient conditions. The same results were reported by Farzad and O’Neal (1991). Making a comparison between optimum charge level of R22 and M20; the lower charge level of M20 is because of its higher specific volume and increased saturation pressure at the condenser.

It is quantified that in the Domestic Test (DT) at a 7% undercharged condition, the refrigerating capacity of R22 is 11.3% lower than that of the optimum condition, whereas the refrigerating capacity of the M20 refrigerant mixture is 6.9% lower than that of the optimum condition. Similarly, at a 7% overcharged condition the refrigerating capacity of R22 is 13.8% lower than that of the optimum condition, whereas the refrigerating capacity of the M20
refrigerant mixture is 6.5% lower than that of the optimum condition. Thus the impact of charge quantity deviations is less pronounced in M20 system as compared to that of R22 system.

Figure 5.20 shows the variation of R22 and the M20 refrigerant mixture evaporator outlet temperature under different refrigerant charge levels and outdoor conditions. It is quantified that in the Domestic Test (DT) at a 7% undercharged condition, the evaporator outlet temperature of R22 is 10.9% higher than that of the optimum condition whereas the evaporator outlet temperature of the M20 refrigerant mixture is 4.5% higher than that of the optimum condition. Similarly, at a 7% overcharged condition the evaporator outlet temperature of R22 is 5.9% lower than that of the optimum condition, whereas the evaporator outlet temperature of the M20 refrigerant mixture is 7.5% lower than that of the optimum condition. In undercharged condition the charge quantity variation gives a lesser impact on the evaporator outlet temperature when M20 is charged as compared to R22. As the charge quantity increases the refrigerating capacity increases resulting in a lower evaporator outlet temperature.

As the refrigerant charge increases, the mass flow rate increases and the evaporator pressure also increases as seen in Figure 5.23. Hence the evaporator inlet temperature also increases as shown in Figure 5.22 corresponding to the saturated condition of the refrigerant. However the outlet temperature is influenced by the superheating as well as glide experienced in the case of zeotropic mixtures. It can be seen that in BIS DT operating condition R22 undergoes a maximum superheating temperature of 31.8°C, while M20 undergoes a maximum superheating temperature of (including glide) of 21.6°C at their respective lowest charge quantities considered. This temperature rise decreases as the charge quantity increases. The drop in evaporator outlet temperature can be attributed to the fact that higher mass
Flow rates have greater potential to absorb the latent heat and hence the temperature rise is minimal. In the case of R22 the evaporator outlet temperature values reaches a plateau after the optimum charge level while for M20 it further continues to decrease. This indicates that an overcharging of M20 can lead to liquid entry in the suction line.

Figure 5.20 Variation of evaporator outlet temperature with respect to refrigerant charge

Figure 5.21 shows the mass flow rate of R22 and the M20 refrigerant mixture under different charge levels and outdoor conditions. The mass flow
rate increases as the charge quantity increases which is more significant in the undercharged condition than in the overcharged condition.

Figure 5.21 Variation of refrigerant mass flow rate with respect to refrigerant charge

It is quantified that in the Domestic Test (DT) at a 7% undercharged condition, the mass flow rate of R22 is 14.8% lower than that of the optimum condition, whereas the mass flow rate of the M20 refrigerant mixture is 8.7% lower than that of the optimum condition. Similarly, at a 7% overcharged condition the mass flow rate of R22 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. The charge quantity variation gives a lesser impact on the mass flow rate when M20 is charged as compared to R22. It is also found that in all the tested operating conditions the mass flow rate of the M20 refrigerant mixture is lower in the range 1.5% to 28% 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 as reported in Kuehl and Goldschmidt (1991). 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 and O’Neal (1991) and Choi and Kim (2002). It is observed that the mass flow rate of R22 with 1400g charge quantity (optimum charge), DT operating condition is 5.7% lower than that of the ETB condition, where as the mass flow rate of the M20 with 1201g charge quantity (optimum charge) DT operating condition is 3.1% lower than that of the ETB operating condition. It is inferred that the outdoor condition variation gives a lesser impact on the mass flow rate when M20 is charged as compared to R22.

Figure 5.22 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.

It is quantified that in the Domestic Test (DT) at a 7% undercharged condition, the evaporator inlet temperature of R22 is 11.8% lower than that of the optimum condition whereas the evaporator inlet temperature of the M20 refrigerant mixture is 12.5% lower than that of the optimum condition. Similarly, at a 7% overcharged condition the evaporator inlet temperature of
R22 is 6.3% higher than that of the optimum condition, whereas the evaporator inlet temperature of the M20 refrigerant mixture is 4.9% higher than that of the optimum condition. In overcharged condition the charge quantity variation gives a lesser impact on the evaporator inlet temperature when M20 is charged as compared to R22.

![Figure 5.22](image)

**Figure 5.22** Variation of evaporator inlet temperature with respect to refrigerant charge

Figure 5.23 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 5.23** Variation of evaporator inlet pressure with respect to refrigerant charge

As seen in Figure 5.23, it is quantified that in the Domestic Test (DT) at a 7% undercharged condition, the evaporator inlet pressure of R22 is 4.4% lower than that of the optimum condition, whereas the evaporator inlet pressure of the M20 refrigerant mixture is 3.9% lower than that of the optimum condition. Similarly, at a 7% overcharged condition the evaporator inlet pressure of R22 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. The charge quantity variation gives
almost the same impact on the evaporator inlet pressure when M20 is charged as compared to R22.

Figure 5.24 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 due to the accumulation of refrigerant. The higher condensing pressure leads to severe problems in the compressor life.

Figure 5.24 Variation of condenser inlet pressure with respect to refrigerant charge
It is quantified that in the Domestic Test (DT) at a 7% undercharged condition, the condenser inlet pressure of R22 is 2.4% lower than that of the optimum condition, whereas the condenser inlet pressure of the M20 refrigerant mixture is 2.1% lower than that of the optimum condition. Similarly, at a 7% overcharged condition the condenser inlet pressure of R22 is 3.4% higher than that of the optimum condition, whereas the condenser inlet pressure of the M20 refrigerant mixture is 1.3% higher than that of the optimum condition. The charge quantity variation gives a lesser impact on the condenser inlet pressure when M20 is charged as compared to R22.

Figure 5.25 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.

It is quantified that in the Domestic Test (DT) at a 7% undercharged condition, the condenser inlet temperature of R22 is 5.0% higher than the optimum condition, whereas the condenser inlet temperature of M20 is 4.0% lower than the optimum condition. Similarly, at a 7% overcharged condition the condenser inlet temperature of R22 is 1% lower than the optimum condition, whereas the condenser inlet temperature of the M20 refrigerant mixture is 1% lower than the optimum in the DT operating condition. The charge quantity variation gives a lesser impact on the condenser inlet temperature when M20 is charged as compared to R22.
Figure 5.25 Variation of condenser inlet temperature with respect to refrigerant charge

5.3.5.2 Influences of Refrigerant Charge and Outdoor Condition on Compressor Power

Figure 5.26 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 higher specific volume and
mass flow rate. Therefore, the compressor power consumption slowly but continuously increases with the refrigerant charge.

![Graph showing variation of compressor power with respect to refrigerant charge](image)

**Figure 5.26 Variation of compressor power with respect to refrigerant charge**

It is quantified that in the Domestic Test (DT) at a 7% undercharged condition, the power consumption of R22 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. Similarly, at a 7% overcharged condition the power consumption of R22 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. The charge
quantity variation gives a lesser impact on the compressor power consumption when M20 is charged as compared to R22.

The power consumption of the M20 refrigerant mixture is lower in the range 2.1% to 10.0% 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.

5.3.5.3 Influences of the Refrigerant Charge and Outdoor Condition on COP

Figure 5.27 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 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 quantities due to higher power consumption, lower cooling capacity and higher pressure ratio. It is observed that the COP of R22 in the 1400g DT operating condition is 25.2% lower than that of the ETB operating condition.

It is found that for the M20 refrigerant mixture in undercharged conditions, due to the lower mass flow rate and refrigerating effect, the COP of the system is lower than that of the optimum charge level. 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.

![Graph](image)

**Figure 5.27 Variation of COP with respect to refrigerant charge**

It is quantified that in the Domestic Test (DT) at a 7% undercharged condition, for R22 the COP is 7.9% lower than that of the optimum condition, whereas for the M20 refrigerant mixture the COP is 3.9% lower than optimum. Similarly, at a 7% overcharged condition the COP of R22 is 15.5% lower than that of the optimum, whereas the COP of the M20 refrigerant mixture is 7.7% lower than the optimum. The COP of the M20 refrigerant
mixture at 1201g DT operating condition is 25.2% lower than that of the ETB operating condition. The charge quantity variation gives a lesser impact on the COP when M20 is charged as compared to R22.

The operation of an air conditioner at elevated ambient temperatures inherently results in a lower coefficient of performance (COP). The COP relation, \( \text{COP} = \frac{T_{\text{evap}}}{(T_{\text{cond}} - T_{\text{evap}})} \), 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.

5.3.6 Overall Heat Transfer Coefficient

This section deals with the overall heat transfer coefficients of R22 and the M20 refrigerant mixture in the evaporator of a window airconditioner operated at standard indoor and outdoor conditions. The experimental overall heat transfer coefficients of R22 and the M20 refrigerant mixture are compared. Experiments are performed in the test conditions given in Table 4.2 using R22 and the M20 refrigerant mixture. The optimized capillary and optimum charge quantity are maintained (as discussed in chapter 4) during experimentation. Based on the measured parameters, viz., the pressures and temperatures across the evaporator; the LMTD, heat flux, and the overall heat transfer coefficients of R22 and the M20 refrigerant mixture are calculated and compared.

If the heat transfer rate is lower than the cooling capacity of the refrigerant, the evaporator temperature and the pressure decreases leading to frost formation and further reduction in the cooling rate. This can cause liquid entry into the compressor suction which is detrimental to the compressor life and performance. On the other hand, if the cooling capacity of the refrigerant is lower than the possible heat transfer rate, the desired cooling of air will not
be achieved. Hence this matching is to be assessed while retrofitting the airconditioner with the new refrigerant.

Figure 5.28 shows the comparison of the evaporator inlet pressure for R22 and the M20 refrigerant mixture under all the operating conditions. It is observed that the evaporator inlet pressure of the M20 refrigerant mixture is higher in the range of 1% to 3% than that of R22 for the operating conditions considered. Even though the operating pressure is higher for the M20 refrigerant mixture, the heat transfer is less as evident in the later graphs of this section. This can be attributed to the poor heat transfer coefficient experienced possibly because of the lower flow rate of the M20 refrigerant mixture.

Figure 5.29 shows the difference in DBT of air across the evaporator for different test condition while using R22 and the M20 refrigerant mixture. It is observed that the temperature difference of R22 is 1.5% to 4.5% higher than that of the M20 refrigerant mixture. The higher DBT difference with R22 results in better refrigerating capacity than that with M20 refrigerant mixture as is evident from Figure 5.10. The lower temperature drop experienced by the air indicates a poor overall heat transfer coefficient of the evaporator when the M20 refrigerant mixture is used. Since air flow conditions remain the same, this directly points out the poor heat transfer coefficient of the M20 refrigerant mixture for the prevailing flow conditions. However, the overall loss in the refrigerating capacity is only less than 10% as discussed in section 5.3.
Figure 5.28 Comparison of the evaporator inlet pressure for different test conditions

Figure 5.29 Comparison of the difference in DBT of air across the evaporator for different test conditions
Figure 5.30 shows the difference in WBT of air across the evaporator for different test conditions while using R22 and the M20 refrigerant mixture. It is observed that the WBT difference with R22 is 2.9% to 12.7% higher than that with M20 refrigerant mixture. This reduced drop in WBT result in lesser dehumidification effect and refrigerating effect also. Due to the temperature glide, the equilibrium temperature of the M20 refrigerant mixture is continuously increased from the inlet of evaporator to outlet. Hence, the wall temperature of the M20 refrigerant mixture does not remain the same as with R22. This can result in decreased DBT as well as WBT difference across the evaporator. These higher DBT and WBT differences offer better refrigerating capacity and accordingly the ON cycle time with the R22 refrigerant is less than that with the M20 refrigerant mixture, which also resulted in reduced energy consumption per day.

![Figure 5.30 Comparison of the difference in WBT of air across the evaporator for different test conditions](image-url)
Figure 5.3 shows the comparison of the LMTD realized with R22 and the M20 refrigerant mixture in the evaporator. The LMTD is higher with M20 as the refrigerant. It is observed that the LMTD with M20 is 2.0% to 6.0% higher than that with R22. This is because of the glide experienced by the refrigerant leading to a smaller temperature difference between the hot and cold fluid at the exit for the parallel flow configuration. However, inspite of higher LMTD the refrigerating capacity is lower because of the higher coil temperature due to the temperature glide leading to a lower drop in WBT as well as DBT. A lesser drop in WBT indicates a lesser latent heat removal and a corresponding reduction in the refrigerating capacity.

![Graph showing comparison of LMTD of R22 and M20](image)

**Figure 5.31 Comparison of the calculated LMTD of R22 and M20**

Figure 5.32 shows the comparison of the calculated vapor fraction of R22 and the M20 refrigerant mixture under the various operating conditions. It is observed that the vapor fraction of M20 refrigerant mixture that enters the evaporator is higher than that of R22 under all the operating conditions.
conditions. Generally, in the capillary tube the two phase flow is mainly influenced by the liquid and vapor specific volumes. The degree of sub-cooling at the inlet to the capillary also significantly influences the exit dryness fraction. In the case of the M20 refrigerant mixture, the temperature glide and the latent heat of condensation increases the condenser load, leading to incomplete condensation as seen in Table 5.4. Hence, the entry to the capillary itself is in the two-phase region, which has been calculated by the measured pressure and temperature at the outlet of the condenser.

**Table 5.4 Condenser outlet conditions**

<table>
<thead>
<tr>
<th>Test conditions</th>
<th>R22</th>
<th>M20</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$P_c$ (bar)</td>
<td>$T_{sat}$ ($^\circ$C)</td>
</tr>
<tr>
<td>BIS DT</td>
<td>21.73</td>
<td>55.0</td>
</tr>
<tr>
<td>BIS DTM</td>
<td>27.11</td>
<td>65.0</td>
</tr>
<tr>
<td>BIS ETA</td>
<td>21.83</td>
<td>55.2</td>
</tr>
<tr>
<td>BIS ETAM</td>
<td>25.43</td>
<td>62.2</td>
</tr>
<tr>
<td>BIS ETB</td>
<td>26.57</td>
<td>64.2</td>
</tr>
<tr>
<td>ASHRAE A</td>
<td>21.83</td>
<td>55.5</td>
</tr>
<tr>
<td>ASHRAE B</td>
<td>18.41</td>
<td>47.67</td>
</tr>
</tbody>
</table>

This is reflected as a higher dryness fraction at the entry to the evaporator as compared to the case with R22. The liquid and vapor specific volumes of the M20 refrigerant mixture are higher than those of R22. Due to the absence of sub cooling and the higher liquid and vapor specific volumes of the M20 refrigerant mixture, the vapor fraction at the entry of the evaporator is higher than that of R22. This is the main drawback with the M20 refrigerant mixture while considering as an alternative to R22 for retrofitting. However, if the condenser length is increased as pointed out in
Jabaraj et al (2006 and 2007) the better condensation can make the M20 refrigerant mixture superior to R22. Needless to say that in spite of the above drawback it can be used as a retrofit refrigerant with a 7.7% reduction in refrigerating capacity and 4.4% reduction in COP.

![Vapor fraction comparison](image)

**Figure 5.32 Inlet vapor fraction of R22 and M20 under different operating conditions**

Figure 5.33 shows the variation of the heat flux for R22 and the M20 refrigerant mixture under the various operating conditions. It is found that the heat flux decreases with an increase in the outdoor air temperature. The heat flux of the M20 refrigerant mixture is lower in the range of 3.1% to 8.7% than that of R22, since the vapor fraction at the entry to the evaporator is higher than that of R22. It is also observed that in BIS ETB and BIS DTM operating conditions the heat flux difference between the M20 refrigerant mixture and R22 is lower than that in other operating conditions. It can be inferred that the loss in the refrigerating capacity of M20 is lower at higher ambient conditions compared to other ambient conditions.
Test conditions
Heat flux (W/m²)

Figure 5.33  Calculated evaporator heat flux with R22 and M20 under different operating conditions

It can also be seen that the heat flux realized, is mainly dependent on the outdoor conditions rather than the indoor conditions. The fact that the condenser temperature and exit subcooling depends on the condenser inlet air temperature is ascribed to this phenomenon. As the degree of subcooling decreases with increasing condenser inlet air temperature, the mass flow rate of the refrigerant increases and the dryness fraction at the inlet of the evaporator also increases leading to poor refrigeration effect. It is also found that the Refrigerating capacity decreases as the magnitude of the drop in refrigeration effect outweighs the rise in the mass flow rate. The net effect is manifested as reduced heat flux which is clearly evinced in these graphs. It is to be noted that the influence of the condensing temperature on the refrigerating capacity in terms of the heat flux drop could be realized only through a heat transfer study in a real appliance and not in conventional heat transfer study test sections and this is the main highlight of this study. The evaporator heat flux realized with M20 refrigerant is lower than that with
R22. It can be seen that the reduction in evaporator flux ranges from 3.1% to 8.7% for M20 as compared to R22.

Figure 5.34 shows the comparison of the experimental overall heat transfer coefficients of R22 and the M20 refrigerant mixture. It is observed that the overall heat transfer coefficient decreases with the increase of outdoor conditions which is the same for both R22 and the M20 refrigerant mixture. An increase in the outdoor temperature increases the condenser pressure leading to an increase in the saturation temperature of the refrigerant in the evaporator as well as the vapor fraction at the entry of the evaporator. The increase in evaporator inlet pressure with increase of condenser pressure is also observed in Figure 5.23.

![Overall heat transfer coefficients of R22 and M20](image_url)

**Figure 5.34 Overall heat transfer coefficients of R22 and M20**

As the inlet vapor fraction increases, the heat flux decreases in the evaporator due to lesser wetted perimeter inside the coil and the lower latent heat absorption rates. It is found that the overall heat transfer coefficient of
the M20 refrigerant mixture is lower in the range of 7.3% to 20.8% than that of R22 under all operating conditions. This can be attributed to the lower heat flux experienced with the M20 refrigerant mixture because of the lower flow rates due to the limited condenser capacity. Further, the LMTD of the evaporator with the M20 refrigerant mixture is also higher than that with R22 and due to all these above reason the overall heat transfer coefficient of M20 refrigerant mixture is lower than that of R22.

5.3.7 Local Heat Transfer Coefficient

The heat transfer coefficient and its distribution in the evaporator tube is strongly influenced by pressure, mass velocity, vapor fraction, wall superheat, properties of the refrigerant, the tube wall material, geometry of the tube, thermal conductivity of the tube wall, liquid composition of refrigerants and flow patterns. Due to the numerous factors affecting the heat transfer coefficient, the recommended correlations that have been developed for other refrigerants and certain thermal and geometric boundary conditions may not predict the local heat transfer coefficients of the M20 refrigerant mixture. In the case of pure substances like R22, aspects of the boiling heat transfer are well explored and reasonably good correlations have been developed for the design of efficient heat exchangers. However, with mixtures, the boiling heat transfer is more complicated. Due to this complexity, the heat transfer problems with mixtures have received less attention than those with pure substances. The lack of such design data as heat transfer coefficients, especially in flow boiling heat transfer, is now one of the limitations in the design of efficient heat exchangers with the M20 refrigerant mixture. Hence, experiments are performed in the appliance itself at the standard test conditions given in Table 4.1 using R22 and the M20 refrigerant mixture.

In this section, the experimentally estimated local heat transfer coefficients of R22 and the M20 refrigerant mixture are discussed, and based
on measured parameters, the influences of the indoor and outdoor air
temperature on the heat transfer rate and local heat transfer coefficients are
also analyzed. The predicted heat transfer coefficients using the existing
correlations are compared with the experimental heat transfer coefficients.
Based on the comparison, the Lavin and Young (1965) and Gungor and
Winterton (1987) correlations are modified to fit the present experimental
heat transfer coefficients data of R22 and the M20 refrigerant mixture,
respectively. A multiple regression analysis is adopted to evolve the intercepts
and exponents of the modified correlations.

Tables 5.5 and 5.6 show the experimentally determined mass flow
rate, inlet evaporator pressure and the calculated mass velocity, heat flux,
vapor fraction entry into the evaporator of R22 and the M20 refrigerant
mixture under different indoor and outdoor temperatures. The comparison of
the experimental heat transfer coefficients is carried out in test conditions,
heat flux, evaporator inlet pressures and mass flow rate.

Table 5.5 Measured parameters for R22 and the M20 refrigerant mixture

<table>
<thead>
<tr>
<th>Properties</th>
<th>TA</th>
<th>TB</th>
<th>BIS DT</th>
<th>BIS DTM</th>
<th>BIS ETA</th>
<th>BIS ETAM</th>
<th>BIS ETB</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass flow rate (kg hr⁻¹)</td>
<td>R22</td>
<td>104.8</td>
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<td>105.5</td>
<td>110.4</td>
<td>104.5</td>
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<td>92.97</td>
<td>99.18</td>
<td>92.60</td>
<td>99.06</td>
<td>94.42</td>
</tr>
<tr>
<td>M20</td>
<td>92.9</td>
<td>7.10</td>
<td>6.26</td>
<td>6.65</td>
<td>6.83</td>
<td>6.95</td>
<td></td>
</tr>
<tr>
<td>Evaporator pressure (bar)</td>
<td>R22</td>
<td>6.22</td>
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<td>7.10</td>
<td>6.29</td>
<td>6.43</td>
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<td>R22</td>
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<td>M20</td>
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<td>2</td>
<td>7</td>
<td>4</td>
<td>5</td>
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<td>2</td>
<td>7</td>
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<td>5</td>
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<tr>
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<td>2</td>
<td>7</td>
<td>4</td>
<td>5</td>
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</table>
Table 5.6 Calculated parameters of R22 and the M20 refrigerant mixture

<table>
<thead>
<tr>
<th>Test conditions</th>
<th>Vapor fraction</th>
<th>Mass velocity (kgm⁻²s⁻¹)</th>
<th>Heat flux (W m⁻²)</th>
<th>Heat flux rank</th>
<th>Mass velocity rank</th>
<th>Vapor fraction rank</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>R22</td>
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<td>R22</td>
<td>M20</td>
<td>R22</td>
<td>M20</td>
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<tr>
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<td>8682</td>
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<tr>
<td>ASHRAE B</td>
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<td>0.402</td>
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<td>9780</td>
<td>9217</td>
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<tr>
<td>BIS DT</td>
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<td>0.428</td>
<td>256.6</td>
<td>226.2</td>
<td>9379</td>
<td>8684</td>
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<td>BIS DTM</td>
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<td>0.479</td>
<td>268.6</td>
<td>241.3</td>
<td>7049</td>
<td>6829</td>
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<tr>
<td>BIS ETA</td>
<td>0.272</td>
<td>0.433</td>
<td>254.4</td>
<td>225.3</td>
<td>9286</td>
<td>8473</td>
</tr>
<tr>
<td>BIS ETAM</td>
<td>0.293</td>
<td>0.449</td>
<td>265.4</td>
<td>241.0</td>
<td>8439</td>
<td>7765</td>
</tr>
<tr>
<td>BIS ETB</td>
<td>0.350</td>
<td>0.463</td>
<td>257.9</td>
<td>229.7</td>
<td>7717</td>
<td>7450</td>
</tr>
</tbody>
</table>

5.3.7.1 Flow Pattern Map

The inner wall temperature of the evaporator, Twi, is estimated from the measured outside wall temperature by applying the one-dimensional, radial, steady-state heat conduction equation for a hollow cylinder, assuming uniform heat generation within the tube wall and an adiabatic condition on the outside of the tube. The corrections made by this procedure depend on the heat flux of the experimental tests. Since the heat transfer process depends upon the flow regime, to analyze the heat transfer coefficients, one must first estimate which flow pattern is present. It is also known that the flow pattern in an evaporator tube plays an important role in the characteristics of heat transfer, and pressure drop as well. The heat transfer coefficient generally keeps changing as the flow pattern changes along an evaporator tube.

It was reported in Lee et al (2003) that the refrigerant flow in an evaporator with multi-pass circuits will have the same pressure drop across each flow circuit. Hence, the same refrigerant flow can be assigned to each path circuit of an evaporator when a capillary tube is connected directly to the
refrigerant flow inlet. Based on the above, the heat flux and mass velocities were divided equally among the two circuits in the studied fin-and-tube evaporator.

Most of the reviewed literature reported that annular flow regimes prevail in the evaporator in airconditioning appliances. In the present work, the heat transfer characteristics study is conducted in a psychrometric test facility as per BIS and ASHRAE test conditions. A knowledge of the flow pattern prevailing in the evaporator is important for the present study, since in horizontal flow boiling, the heat transfer characteristics are largely affected by the flow pattern, viz, annular or stratified flow. The flow patterns could be characterized as a function of the mass flow rate, vapour fraction and fluid properties. The measured parameters and the calculated vapor fraction for R22 and the M20 refrigerant mixture are tabulated in Tables 5.5 and 5.6 respectively. Depending on the test conditions, a flow pattern map is developed for R22 and the M20 refrigerant mixture in a fin-and-tube horizontal evaporator tube. Figures 5.35 and 5.36 show the flow pattern map for R22 and the M20 refrigerant mixture based on the Kattan-Thome-Favrat (1998) flow pattern map. Though the flow pattern maps for different conditions vary, the maps corresponding to the extreme heat flux conditions (viz TB and BIS DTM) presented in Tables 5.5 and 5.6 are presented for discussion. It could be seen that the annular flow regimes prevail in extreme conditions and hence the discussion will be tenable for the full range of test conditions. The most common commercial evaporators are those in which the refrigerant boils inside the tubes. At higher vapor qualities the forced flow boiling inside a tube is in the annular regime, which is characterized by a liquid film around the tube perimeter and a vapor core.
For the R22 refrigerant, it is found from experimentation that the mass velocities varied between 223.5 and 268.5 kg m$^{-2}$ s$^{-1}$, and the heat fluxes between 7049 and 9780 W m$^{-2}$. It is found that the inlet vapour fraction varies between 0.23 and 0.39 for the operating conditions considered. For the R22 refrigerant, with increasing mass velocity and vapor fraction, the flow pattern changes from intermittent to annular. The more uniform distribution around the periphery can be attributed to the pattern of the annular flow in the R22 refrigerant evaporator tube. With the annular flow in the tube, the top and side portions of the tube section are wetted. Therefore, the more uniform distribution and higher values of the heat transfer coefficients are obtained in the tube.

For the M20 refrigerant mixture, it is found from experimentation that the mass velocities varied between 212.2 and 241.3 kg m$^{-2}$ s$^{-1}$, and the heat fluxes between 6829 and 9217 W m$^{-2}$. It is found that the inlet vapour
fraction is around 0.4 for all operating conditions. The flow regimes prevailing in the evaporator are presented in Figure 5.36. It can be observed from Figure 5.36 that the annular flow pattern is experienced inside the fin-and-tube evaporator for the M20 refrigerant mixture.

![Flow pattern map for M20 in a 9.524 mm tube](image)

**Figure 5.36 Flow pattern map for M20 in a 9.524 mm tube**

The transitions of flow pattern occur at lower mass velocity and vapor fraction in the R22 evaporator tube than in the M20 refrigerant mixture evaporator tube. The flow patterns observed in the experiment are annular flow regimes for both R22 and the M20 refrigerant mixture. It can be seen from Figure 5.35 that intermittent flow occurs until a vapour fraction of 0.359 while for the M20 refrigerant mixture the intermittent flow occurred until a vapour fraction of 0.372. In the M20 refrigerant mixture tube, the annular flow regime prevails at all the realized G and x values. Further these flow regimes in each case will be referred to for later discussions.
5.3.7.2 Temperature Distribution

Figure 5.37 shows the comparison of the calculated refrigerant temperature (Tr), inner wall temperature (Twi) and vapour fraction (x) with the number of tubes in a fin-and-tube evaporator in BIS DT operating condition for R22 and the M20 refrigerant mixture. Since the trend remained the same irrespective of the test condition, the BIS DT condition which is the closest to the typical tropical weather, is considered for the discussion of the local heat transfer coefficient. In real appliances, the refrigerant enters the evaporator at a vapor fraction greater than 0.2 in all operating conditions as given in Table 5.6; due to this, the nucleate boiling effect diminishes and the convective boiling region dominates as the stable annular flow develops, as shown in Figure 5.35. It can be seen that the vapor fraction entry into the evaporator with R22 is lower than that of the M20 refrigerant mixture as discussed in the section 5.3.6.

It is observed that the measured wall temperature of R22 is initially increased in the first row and slightly decreased in the second and third rows. Moreover the wall temperature with R22 is lower than that with M20 refrigerant mixture. The lower wall temperature will offer better DBT and WBT differences across the evaporator. Hence a better heat transfer rate and heat transfer coefficient can be expected with the R22 refrigerant. The higher wall temperature with M20 can be attributed to higher vapour fraction entry into the evaporator and temperature glide.
Figure 5.37  Comparison of temperature distribution and vapor fraction along the refrigerant flow path

For R22, due to the increasing wall temperature, the temperature difference between the refrigerant and the wall is lower in the first row. But, in the second and third rows slightly decreasing trend is observed, as the vapor fraction increases because of the reduced liquid film thickness. For the M20 refrigerant mixture, the temperature difference between the inlet and outlet of the first and second row is more or less the same, while in the third row, the temperature difference is very low.

For R22 the saturation temperature decreases along the length, because of the pressure drop in a tube during boiling, but the temperature increases at the outlet of the evaporator due to the superheated condition. The temperature of the zeotropic refrigerant mixture increases with the increase in the vapor fraction, because of temperature gliding effects. However, there are other cases where the temperature of a zeotropic refrigerant mixture decreases when the temperature drop due to the pressure drop is greater than the temperature gliding effect.
The interruption of the liquid film in an annular flow is known as the dryout phenomenon, and it is attributed to the effect of evaporation and hydrodynamic forces that cause the entrainment of droplets from the liquid film. In heat flux controlled systems, the beginning of dryout is accompanied by a decrease in the heat transfer coefficient, and consequently, by an increase in the wall temperature. It is found from Figure 5.37, that the wall temperature starts increasing due to dryout phenomenon and super heating of the refrigerant beyond 18 tubes with the M20 refrigerant mixture and beyond 21 tubes with R22 (refer Figure 4.13 for tube numbering).

Figure 5.38 shows the comparison of the measured air temperature at the inlet (Tdbti) and outlet (Tdbto) along the refrigerant flow path in a fin-and-tube evaporator in the BIS DT operating condition for R22 and the M20 refrigerant mixture. It can be seen that the air inlet temperature in the first row remain the same, while the exit temperature profile has a slope and the variation along the refrigerant flow path depends upon the refrigerant flow configuration for both the R22 and the M20 refrigerant mixture.

For R22, this is mainly caused by a changing refrigerant heat-transfer coefficient, which is greater at higher refrigerant qualities. For the zeotropic mixture, the slope of the air temperature is more pronounced because of the glide of the mixture’s saturation temperature during a phase change. It is observed from Figure 5.38, that the temperature difference between the inlet and outlet of the air for R22 is very low in the third row than that of the M20 refrigerant mixture. Moreover, the air temperature difference in the second row of R22 is lower than that of the M20 refrigerant mixture. Therefore, increased wall temperatures are observed in the present study and the same results are evident from the literature Barbosa and Hewitt (2001), due to the mass transfer resistance offered by the components of the M20 refrigerant mixture.
5.3.7.3 Variation of Vapor Fraction at the Evaporator Inlet

Figures 5.39 and 5.40 show the variation of vapor fraction along the refrigerant flow path for R22 and the M20 refrigerant mixture respectively, under different operating conditions. Three operating conditions based on the vapor fraction ranking shown in Table 5.6 viz minimum, maximum and mid ranking are considered. Accordingly TB, BIS ETA and BIS DTM are discussed. It is observed that the calculated vapor fraction entry into the R22 evaporator is lower than that of the M20 refrigerant mixture. As explained in section 5.3.6, due to the absence of sub-cooling at the condenser exit and higher liquid and vapor specific volumes of the M20 refrigerant mixture, the vapor fraction for M20 refrigerant mixture at the entry of the evaporator is higher than that of R22.
Figure 5.39  Calculated vapor fractions along the refrigerant flow path for R22 under different operating conditions

Figure 5.40  Calculated vapor fractions along the refrigerant flow path for M20 under different operating conditions
A higher vapor fraction entry into the evaporator may reduce the wetted perimeter inside the evaporator tube and in turn the heat transfer coefficient decreases. Hence poorer heats transfer rate leading lower refrigerating effect prevailing in the fin-and-tube evaporator with the M20 refrigerant mixture. It can be also seen that as the condenser inlet air temperature increases the dryness fraction at the inlet increases due to increased flashing in the capillary.

5.3.7.4 Heat Transfer Rate

Figures 5.41 and 5.42, illustrate the heat transfer rate prevailing in the tubes of the evaporator with R22 and the M20 refrigerant mixture respectively. In the subsequent figures F, S and T represent first, second and third row respectively and the tube numbering in each row increases in the direction of refrigerant flow (i.e. top to bottom in the I row and bottom to top in the II and III row). It is observed that the heat transfer rate decreases as the air travels across the rows, because the temperature difference between the air and the refrigerant decreases. It is observed that the heat transfer rate in each row is not identical but varying in nature, depending on path configuration (shown in Figure 4.13). Moreover, the heat transfer rate is not evenly distributed in each row in parallel-cross flow type evaporator with R22 and the M20 refrigerant mixture and the same is also reported by Lee et al (2003) with reference to the R22 evaporator. It can be also seen that the heat transfer rate is more influenced by the air temperature rather than the condition of refrigerant.

The heat transfer rate in the third row in the parallel-cross flow configuration with R22 is not higher than that in the second row even though the third row has same surface area compared to the first row. It is found that the heat transfer rate decreases in the order of ASHRAE B > BIS ETA > BIS DTM operating conditions. The heat transfer rate of the BIS ETB operating
condition is lower than that of the ASHRAE B operating condition, due to higher evaporating pressure and condensing pressure. The evaporating pressure and condensing pressure increase depending on the evaporator and condenser inlet air temperature.

![Graph showing heat transfer rate variation of R22 under different operating conditions in the evaporator tubes](image)

**Figure 5.41 Calculated heat transfer variation of R22 under different operating conditions in the evaporator tubes**

It is found from Figure 5.42 that the M20 refrigerant mixture had 6.0%, 9.7% and 4.2% lower average heat transfer rate than that of the R22 system in ASHRAE B, BIS ETA and BIS DTM test conditions respectively. Due to the higher vapor fraction at the inlet of the evaporator and the lower mass flow rate of the M20 refrigerant mixture, the heat transfer rate is lower than that of the R22 system. It is also recalled that more number of tubes are occupied with the vapor (superheated) refrigerant in the M20 refrigerant mixture evaporator than that of the R22 evaporator as seen in Figure 5.37.
Figure 5.42 Calculated heat transfer variation of M20 under different operating conditions in the evaporator tubes

Figures 5.43 and 5.44 show the calculated heat transfer variation along the refrigerant flow path for R22 and the M20 refrigerant mixture respectively, under different operating conditions. It is found from Figures 5.43 and 5.44, that the heat transfer rate decreases with increasing inlet vapor fraction, since the pressure of the refrigerant and the wetted perimeter of the tube decreases with an increase in the inlet vapor fraction. It is also observed that the heat transfer rate in the tubes after I row increases and decreases alternatively because of zigzag configuration. It can be concluded that the heat transfer rate in each row depends on the path configuration and the heat transfer rate is not evenly distributed in the tubes of each row in the parallel-cross flow evaporator with R22.

Takamatsu et al (1993) have reported that for a parallel flow configuration, the heat transfer rate decreases with vapor fraction. It was explained that the temperature difference between the refrigerant and the air decreases for the parallel flow. Hence, the heat transfer rate decreases with
vapor fraction for the parallel flow. Moreover, it was explained, that compared to the results for pure refrigerants, the change in heat flux with vapor fraction is high for the parallel flow, since the refrigerant temperature increases due to temperature glide.

![Graph showing calculated heat transfer variation of R22 along the refrigerant flow path under different operating conditions.](image)

**Figure 5.43** Calculated heat transfer variation of R22 along the refrigerant flow path under different operating conditions
5.3.7.5 Heat Transfer Coefficient

The heat transfer coefficient variation for different 1) Heat flux condition 2) Mass flux condition 3) Evaporator inlet pressure conditions are discussed in this section. As shown in Tables 5.5 and 5.6 the above parameters in each test condition are evaluated and ranked. The results of these test conditions for maximum, mid and minimum values of these parameters are considered for discussion. As the operating conditions are changed in real time, these parameters change along with the heat transfer coefficient.

5.3.7.5.1 Variation of the Heat Transfer Coefficient with Heat Flux

The trend of the local heat flux along the length in a typical evaporator possesses a varying nature as discussed already. The magnitude of the heat flux depends on the flow pattern of the refrigerant inside the
evaporator, and the evaporator air inlet temperature. The heat transfer coefficient can vary locally due to mal-distribution of air temperature and phase change on the refrigerant side. The temperature glide causes a change in the temperature difference between the air and the refrigerant side. The zigzag flow of the refrigerant between the second and third rows of the evaporator coil influences the local heat transfer coefficient on the tube side. Hence, the variations of the heat transfer coefficient with respect to the tube number (considered consecutively and row wise) are presented separately to unravel the trend in the variation. The vapor fraction is not considered as the primary variable to avoid confusion that may arise due to the zigzag flow configuration. However the vapor fraction increase is directly dependent on the consecutive tube numbering scheme. In this scheme, there is also an additional advantage to present the heat transfer coefficient drop realised after complete vaporization of the refrigerant.

The experimental local heat transfer coefficients of R22 are shown against tube numbers along the refrigerant flow path in Figure 5.45. In a parallel flow, the temperature difference between the air and the refrigerant side decreases and the observed results for the considered heat flux conditions also reveal that the heat transfer coefficient decreases with increasing vapour fraction.

Takamatsu et al (1993) have observed in a parallel flow that the experimental heat transfer coefficients of R22 decrease with increasing vapour fraction. It can be also seen that as the heat flux reduces the heat transfer coefficient is also decreasing. The zigzag flow path between II and III row yields zigzag (up and down) variation in the heat transfer coefficient realised in the tubes considered sequentially along the flow. Hence the variation in the row wise numbering scheme is also shown for better understanding.
The variation of the local heat transfer coefficient prevailing inside the refrigerant tube in the row numbering scheme is shown in Figure 5.46. It is observed that as the heat flux increases, the heat transfer coefficient also increases and the heat transfer coefficient decreases from row to row as the flow proceeds. The heat transfer coefficient can vary locally due to the variation of the temperature difference air and refrigerant prevailing between rows and phase change occurring on the refrigerant side.

The variation is also not steady because of the phase change experienced by the refrigerant in a zigzag flow across the second and third rows and the mal-distribution of air prevailing in the evaporator. These variations significantly affect the refrigerating capacity and are realised in a set-up like this with the appliance itself modified to be the experimental setup. The experienced drop in the local heat transfer coefficient along the flow direction is a characteristic of the heat transfer coefficient in parallel flow
configuration as reported in the literature by Takamatsu et al (1993). Even within the row the heat transfer coefficient decreases because of the increase in dryness fraction leading to reduced wetted perimeter. It is interesting to note that S₇, S₈, T₇ and T₈ tubes experience the lowest heat transfer coefficient as they are in the end of the flow path where the refrigerant is superheated.

![Graph](image-url)

**Figure 5.46 Experimental heat transfer coefficient of R22 under different heat fluxes**

The experimental heat transfer coefficients of the M20 refrigerant mixture against tube numbers for the same conditions along the refrigerant flow path, are shown in Figure 5.47. However, the heat flux values differ from those of R22 because of the different system behaviour with the M20 refrigerant mixture. In the parallel flow, the calculated results for these heat flux conditions reveal that the heat transfer coefficient decreases with tube number along the refrigerant flow path similar to the R22 results, as the convective evaporation governs the heat transfer in the high-vapour fraction region. Takamatsu et al (1993) have concluded that for the parallel flow test
section the heat transfer coefficient decreased and the convection becomes predominant at higher vapour fraction. Jung and Radermacher (1993) also reported that nucleate boiling was suppressed at vapor fraction greater than 20% for all mixtures, and convective evaporation becomes the main heat transfer mechanism.

![Graph showing heat transfer coefficient along the refrigerant flow path](image)

**Figure 5.47** Experimental heat transfer coefficient of M20 along the refrigerant flow path under different heat fluxes

It can be seen that maximum heat flux experienced with M20 is 9217 W m\(^{-2}\) as against 9780 W m\(^{-2}\) in the case of R22 which is 5.7% lower. Correspondingly the heat transfer coefficient is also lower for M20 and it varies in the range of 13.8% to 41.8% for the maximum heat flux condition. The same trend prevails for the other minimum and mid heat flux conditions also. The reduced heat transfer coefficient for M20 can be attributed to the entry dryness fraction being 0.428 against 0.228 for R22. This can be improved by providing a higher mass flow rate possible by increasing the condenser length.
The heat transfer coefficient varies in a zigzag fashion along the flow path because of the zigzag configuration between row II and III. The heat transfer coefficient with a row wise numbering scheme is also shown in Figure 5.48. It can be seen individually in II row and III row there is a marginal rise in heat transfer coefficient within the row. This can be attributed to the temperature glide experienced by the vaporizing refrigerant mixture.

Figure 5.48 represents the variations of the heat transfer coefficient of the M20 refrigerant mixture in the different tubes of each row under different heat flux conditions. The temperature gliding due to phase changes in the refrigerant mixture components decreases the temperature difference between the air and the refrigerant side. The temperature gliding effect of the M20 refrigerant mixture can also increase the local heat transfer coefficient within the row. Consequently the heat transfer rate also decreases marginally in the second and third rows which are reflected in the estimated heat transfer coefficient also. This decreasing phenomenon does not occur in the R22 evaporator; rather it increases marginally due to a drop in the refrigerant temperature, as the saturation pressure decreases due to friction. The slump in the heat transfer coefficient at S₇, S₈, T₇ and T₈ are due to extreme superheating over and above the temperature glide experienced. This high degree of superheating can be seen in Figure 5.37 also. It is to be noted that a higher mass flow can solve all these issues of poor heat transfer coefficient while using M20. Increasing the condenser length can enable realising a higher mass flow rate.
Figure 5.49 represents the comparison of the variations of the heat transfer coefficients of R22 and the M20 refrigerant mixture under different heat flux conditions evaporator between each row. It is found that the heat transfer coefficients of the M20 refrigerant mixture in the different operating conditions considered, are lower in the range of 13.8\% to 41.8\%, 38.7\% to 57.4\%, and 19.4\% to 55.9\% respectively, than those of R22. It can be seen that for the same operating conditions R22 has got a higher heat flux than the M20 refrigerant mixture. This can be a causative factor for the higher heat transfer coefficient in R22. Further, the mass transfer resistance experienced in a zeotropic mixture can also be ascribed as a reason for the poorer heat transfer coefficient of the M20 refrigerant mixture.
It is observed that the difference between R22 and the M20 refrigerant mixture heat transfer coefficients is quite pronounced. Further, the flux variation influences the heat transfer coefficient more significantly in the case of R22 than it is in the M20 refrigerant mixture. It is known fact that the heat transfer coefficient of a zeotropic mixture is less than the weighted average of the heat transfer coefficient of the individual components at the same operating conditions due to the mass transfer resistance experienced at the liquid vapor interface. The same was also explained in the reviewed literature Jung et al (1989), Murata and Hashizume (1993) and Choi et al (2003) also. In a zeotropic refrigerant system, the more volatile component in the liquid evaporates more rapidly than the less volatile component according to the phase equilibrium. As the quality is increased, the liquid phase becomes richer in the less volatile component while the vapor phase is richer in the more volatile component. Thus, the concentration of the more volatile component in the bulk liquid is lower than that at the gas-liquid interface.
while the concentration of the more volatile component at the gas-liquid interface is lower than that in the bulk vapor. This concentration difference makes the diffusion of the more volatile component from the bulk liquid to the interface and from the interface to the bulk vapor difficult causing a reduction in the heat and mass transfer rate. The more the compositions in the liquid and vapor differ, the more the heat and mass transfer coefficients are reduced. This effect outweighs the effect due to the variation in the heat flux and hence the heat transfer coefficient variation due to flux variation is not phenomenal in M20 as in R22.

5.3.7.5.2 Variation of the Heat Transfer Coefficient with Mass Velocity

Generally, the heat transfer coefficient is directly proportional to the mass velocity which can be explained in many ways. An increased mass velocity can result in an annular flow pattern, which is the most preferred flow for better heat transfer. A higher mass velocity can increase turbulence in the liquid phase, and the disturbance by the interfacial waves also cause heat transfer enhancement by increasing the mixing effect. However, in a real appliance, this mass velocity is decided by the operating conditions which also have other counter veiling effects on the heat transfer rate. As the air temperature entering the condenser is increased, the mass flow rate passing through the capillary tube also increases. But the higher air inlet temperature in the condenser will affect liquid sub-cooling, that reduces the heat transfer rate in the evaporator, because the higher vapour fraction entry into the evaporator leads to reduced wetted perimeter and in turn poor heat transfer coefficient. Similar effects can be expected when the evaporator inlet temperature also changes. Thus, in a real appliance the heat transfer coefficient decreases as the mass flow increases (because of ambient conditions), which is unlike the trend normally reported in literature based on a study in standard test sections.
Hence, in the present study, the experiments are carried out as per the standard test condition for a real appliance. In this section, to analyze the variation of the heat transfer coefficient with mass velocity, three test conditions are selected based on their mass velocity ranking, viz the maximum, minimum and the mid-ranking cases and the results are presented against the respective mass velocities realized.

Figures 5.50 and 5.51 show the variation of the heat transfer coefficient along the refrigerant flow path for different mass flow rates of the R22 and M20 refrigerant mixtures, respectively. It can be seen that as the mass velocity increases the heat transfer coefficient decreases in both the cases. In the case of R22 as the flow proceeds, by and large, the heat transfer coefficient tends to decrease. But for the M20 refrigerant mixture, within the row the heat transfer coefficient increases which has been discussed in section 5.6.5. Due to the lower heat transfer rate, the heat transfer coefficient of R22 is decreased in the higher mass flow rate also.

![Figure 5.50](image.png)

**Figure 5.50** Experimental heat transfer coefficient of R22 along the refrigerant flow path under different mass flow rates
Figure 5.51 Experimental heat transfer coefficient of M20 along the refrigerant flow path under different mass flow rates

It is observed that the experimental heat transfer coefficients for a refrigerant mixture increase for lower mass velocity for both R22 and the M20 refrigerant mixture. It is found that the heat transfer coefficients of the M20 refrigerant mixture in the considered operating conditions, are lower in the range of 43.8% to 57.9%, 19.9% to 55.6%, and 27.2% to 55.3% respectively, than those of R22.
Figure 5.52  Comparison of the experimental heat transfer coefficient of both R22 and M20 under different mass flow rate

5.3.7.5.3 Variation of the Heat Transfer Coefficient with Evaporator Inlet Pressure

The local heat transfer coefficient as a function of the evaporating pressure at the inlet is shown in Figure 5.53 for R22. It can be seen from Figure 5.53 that the heat transfer decreases as the evaporating pressure increases, the temperature difference between the air and refrigerant side decreases with increasing evaporating pressure. Lee et al (2003) also reported that the heat transfer coefficient decreases with an increase in the evaporating pressure. The general trend is that the two phase heat transfer coefficients decrease as the evaporation temperature increases. The major terms that affect the heat transfer coefficient as the evaporating pressure increases are properties of the refrigerant and reduced pressure.
Jung and Radermacher (1993) have reported that when nucleate boiling is suppressed, the heat transfer coefficient is directly proportional to $k_i^{0.6} \left( \frac{C_p}{\mu_i} \right)^{0.4}$ and reduced pressure ($P_{\text{red}}$). As temperature increases, the property term $k_i^{0.6} \left( \frac{C_p}{\mu_i} \right)^{0.4}$ increases while the reduced pressure ($P_{\text{red}}$) decreases. The effect of the latter, however, is stronger than that of the former, resulting in a decrease in the heat transfer coefficients as the temperature increases. Thus the heat transfer coefficient of R22 decreases with the evaporating pressure which is exactly opposite to that realized in conventional test sections. This is primarily because, the heat flux in the present set-up is not an imposed constant heat flux but a varied heat flux that prevails in the actual appliance also.

The heat transfer coefficient of the M20 refrigerant mixture as a function of the evaporating pressure is shown in Figure 5.54. It is observed that the two-phase heat transfer coefficients decrease as the evaporating pressure
pressure increases. The heat transfer reduction due to the concentration is significant at high pressures.

![Figure 5.54 Experimental heat transfer coefficient of M20 along the refrigerant flow path under different evaporator pressures](image)

This can be explained by the fact that the number of vapor bubbles being formed per unit surface area increase with the pressure. The vapor bubbles are more densely packed, and there are fewer surfaces available for the mass flow rate in the liquid space. Thus, the delivery of the low-boiling components is more strongly impeded, which leads to a further reduction in the heat transfer. Similar results have been found for many other liquid mixtures also.

Figure 5.55 shows the comparison of the experimental heat transfer coefficients of R22 and the M20 refrigerant mixture under different evaporator pressures. It can be seen that the heat transfer coefficient of the M20 refrigerant mixture is lower than that of R22. The superposition of heat transfer and mass transfer phenomena associated with the evaporation of a
mixture, decreases the heat transfer coefficient, compared to those pertaining to pure liquids.

For refrigerant mixtures, the boiling site density is affected by the mixture composition. Therefore, it is expected that the heat transfer for mixtures will be suppressed when compared with that for pure refrigerants under similar experimental conditions. The same phenomenon was also discussed in Choi et al (2003). The non-linear property variation with respect to mixture composition and mass transfer resistance near the interface, are causes for the degradation of the heat transfer coefficients. Figure 5.55 shows the comparison of heat transfer coefficients of the M20 refrigerant mixtures with that of R22. It is found that the heat transfer coefficients of the M20 refrigerant mixture in the considered operating conditions are lower in the range of 13.8% to 41.8%, 38.7% to 57.4%, and 19.4% to 55.9% respectively, than those of R22.
5.3.7.6 Comparison of the Existing Correlation with the Present Work

Many correlations for local heat transfer coefficients have been proposed for the design of evaporators. These correlations are classified into the following four types:

i) **Additive type**: The two-phase heat transfer coefficient is expressed in an additive form of correlation containing both the forced convection and nucleate boiling terms. The Gungor and Winterton (1986), Jung et al (1993) and Choi et al (2000) correlations are additive type correlations.

ii) **Asymptotic type**: The asymptotic model in which both the nucleate and convective terms are substituted with a suitable index based on the influence of the individual terms. The Liu and Winterton (1991) and Wattelet et al (1994) correlations come under the asymptotic type.

iii) **Based on property group**: The two phase heat transfer coefficient is defined as a function of the liquid heat transfer coefficient and non-dimensional numbers such as the Boiling number or Martinelli parameter. The Gungor and Winterton (1987) and Lavin Young (1965) correlations come under the property group.

iv) **Based on Froude number**: The two phase heat transfer coefficient is decided either by forced convective boiling or the nucleate boiling term which is dominant and the dominance is decided by the calculated Froude number. The Kandlikar (1990), and Shah (1982) correlations come under the correlations based on the Froude number.

The existing correlations are chosen because of their general acceptance and the broad data base of the refrigerant characteristics used in
their formulation. While conducting the simulation study it is necessary to consider a suitable correlation for the prediction of the behavior of fin-and-tube evaporator with R22 as well as the M20 refrigerant mixture. This would enable effective and meaningful comparison between the two refrigerants for the fin-and-tube evaporator. Hence, the above correlations are tried for predicting the heat transfer coefficients of both the refrigerants and their respective experimental results are compared. The closely predicting correlations in both the cases (R22 and M20) have been identified and suitably modified for the best fit of the experimental results.

5.3.7.6.1 Comparison of the Existing Correlations with the R22 Data

A comparison of the experimental heat transfer coefficients with the predicted heat transfer coefficients using the existing correlations such as Lavin and Young (1965), Shah (1982), Gungor and Winterton (1986 and 1987), Kandlikar (1990), Liu and Winterton (1991), Jung et al (1993), Wattelet et al (1994) and Choi et al (2000) correlations marked as LY, Shah, GW_86, GW_87, Kandlikar, LW, Jung, Wattelet, and Choi respectively, are shown in Figures 5.56, 5.57 and 5.58. These figures show the deviation between the experimental and predicted heat transfer coefficients using the above correlations in the same test conditions.

In Figure 5.56 the Lavin and Young (1965), Gungor and Winterton (1987), Shah (1982), and Kandlikar (1990) correlations are compared with the experimental results. It is observed that the majority of data lie within the range of +60% to +80% deviation. However the Lavin and Young (1965) correlation data lie within the deviation range of -20 to +65%. The Gungor and Winterton (1987) correlation has been developed for flow boiling in vertical, horizontal tubes and annuli, using the data base consisting of 4202 data for saturated boiling and 946 data points for sub-cooled boiling. The Kandlikar (1990) correlation is evolved using a data base consisting of 5236 data points from 24 experimental investigation results covering 10 working
fluids, mostly for vertical and horizontal tubes. The Lavin and Young (1965) correlation is tested for R12 and R22 data, and it is claimed that it is a suitable correlation for both the refrigerants. The Shah (1982) correlation is evolved based on 800 data points of water, R11, R12, R22, R113, and Cyclohexane for vertical and horizontal flow configurations.

![Graph showing deviation of predicted HTC from experimental HTC]

**Figure 5.56 Mean deviation of the predicted local heat transfer coefficient of R22**

Shah (1982), Gungor and Winterton (1987) and Kandlikar (1990) correlations are called as general correlations since all of them have been developed based on the extensive data base covering different refrigerants in a wide range of operating conditions such as mass velocity, heat flux, pressure, quality and tube diameter. Due to their generality, these correlations show large deviation with the experimental heat transfer characteristics of a particular refrigerant and specified operating condition in an appliance.
Figure 5.57 shows the deviation of the predicted heat transfer coefficients using the existing correlations such as Gungor and Winterton (1986), Jung et al (1993) and Choi et al (2000) with the experimental heat transfer coefficients of R22. Among these three correlations, the Gungor and Winterton (1986) correlation has been evolved using a data base from 28 authors’ experimental results consisting of over 4300 data points covering seven fluids, mostly for saturated boiling in vertical and horizontal tubes. The Choi et al (2000) correlation has been evolved based on the data of R32, R134a, R32/134a, R407C and R22 in a horizontal smooth tube. Experiments were conducted at average temperatures of -12°C to 17°C, with the mass velocity of 240 to 1060 kg m⁻² s⁻¹, and the heat flux of 4.1 to 28.6 kW m⁻². The Jung et al (1993) correlation has been evolved based on more than 3000 data of R22, R114, R12, R152a and their mixtures at various concentrations. The heat flux was varied from 10.0 to 45.0 kW m⁻², and the mass velocity from 250 to 720 kg m⁻² s⁻¹. The above three correlations have considered the nucleate and convective terms for the prediction of two phase heat transfer coefficients. But, as discussed in the previous section, in the considered operating conditions, the convective heat transfer phenomenon dominates in the fin-and-tube evaporator. Due to this, the predicted results show a deviation in the range of +30% to +90%.
Figure 5.57 Mean deviation of the predicted local heat transfer coefficient of R22

Figure 5.58 shows the deviation of the predicted heat transfer coefficients of R22 using the existing Liu and Winterton (1991) and Wattelet et al (1994) correlations with the experimental heat transfer coefficients. Among these two correlations, the Liu and Winterton (1991) correlation is evolved using the saturated boiling data base, containing 4202 data points for saturated boiling and 991 data points for sub-cooled boiling. These data were collected from 30 different literature sources involving nine different fluids. The Wattelet et al (1994) correlation is based on the data of R12, R134a and a mixture of R22/R124/R152a (52%/33%/15%) in a single-tube horizontal evaporator for both annular and wavy-stratified flows. The operating parameters were varied in the range of mass velocity, 25 to 100 kg m$^{-2}$ s$^{-1}$; heat flux, 2 to 10 kW m$^{-2}$; vapor fraction, 10% to 90%; and saturation temperature, -15°C to 5°C. Even though the heat flux and vapor fraction considered in the Wattelet et al correlation are within the present experimental
input conditions, the Wattelet et al correlation largely over predicted the experimental heat transfer coefficients. It is observed that the Liu and Winterton (1994) correlation predicts the experimental data in the range of +35 to +68% deviation.

**Figure 5.58 Mean deviation of the predicted local heat transfer coefficient of R22**

Considering the above correlations, most of the heat transfer studies were carried out in a tube-in-tube heat exchanger with parallel or counter flow configurations. Electrically heated or auxiliary heated test sections were used and the studies were, by and large, done at constant heat flux or constant wall temperature conditions, and heat flux conditions were externally imposed. But, in real appliances, the mass flow rate of the refrigerant, evaporator pressure, evaporator heat flux and vapor fraction of the refrigerant at entry and outlet, are decided by the combined characteristics of the compressor and other components in the prevailing operating conditions (i.e., indoor and outdoor air temperatures). Furthermore, the developed correlations are not for
a fin-and-tube evaporator with different tube configurations. The up and down flow of the refrigerant in the circuit, bends in the circuit, and air being used as the heat transfer fluid, are not taken into account for the existing correlations. Due to the above, in real appliances (the window air conditioner), the predicted heat transfer coefficient does not match with the experimental values, and it shows such large deviations. However to evolve a correlation for R22 specific to the heat transfer coefficient in a fin-and-tube evaporator one of the existing correlations is to be chosen for suitable modification to fit the present experimental data.

In Figure 5.59 the experimental heat transfer coefficients of R22 are compared with the well known correlations such as the Lavin and Young (1965), Shah (1982), Gungor and Winterton (1986), Gungor and Winterton (1987) and Choi et al (2000), correlations. Comparison has been made for an operating pressure of 6.263 bar, heat transfer rate of 2345 W and mass velocity of 255 kg m$^{-2}$s$^{-1}$ in a 9.52mm diameter tube.

It is worth noting that the difference between the correlations is not too large compared to that with the experimental results. It is also observed that the Lavin and Young (1965) correlation predicts the heat transfer coefficient of R22 with minimum deviation from the experimental data. Hence, the Lavin and Young (1965) correlation is considered for the above modification.
Figure 5.59  Comparison of the experimental and predicted heat transfer coefficient for R22

5.3.7.6.2 Comparison of the Existing Correlations with the M20 Data

This section presents the variation of the predicted heat transfer coefficient using different published correlations with the experimental heat transfer coefficients of the M20 refrigerant mixture. A few of the existing correlations are chosen for their features of dealing with refrigerant mixtures, their generality, and the number of data considered for their evaluation. A comparison of the experimental heat transfer coefficients with the predicted heat transfer coefficients by correlations such as Shah (1982), Gungor and Winterton (1986 and 1987), Liu and Winterton (1991), Jung et al (1993), Wattelet et al (1994) and Choi et al (2000) marked as Shah, GW_86, GW_87, LW, Jung, Watt and Choi respectively, are shown in Figures 5.60, 5.61 and 5.62.

Figure 5.60 shows the deviation of the predicted heat transfer coefficients of the Gungor and Winterton (1986), Jung et al (1993) and
Choi et al (2000) correlations with the experimental heat transfer coefficients of the M20 refrigerant mixture. The Choi et al (2000) correlation is evolved based on the experimental evaporative heat transfer coefficients of R32, R134a, R32/134a, R407C and R22 in a horizontal smooth tube, and the effect of heat flux and the temperature difference between the dew point and bubble point, are considered by a proposed simple form of correction factor for mixtures incorporated in this correlation. The correlation by Gungor and Winterton (1986) is developed from a large data base including water, refrigerants and ethylene glycol, and it has been widely used to predict the experimental heat transfer coefficients of all the refrigerants. In this analysis, as reported in Choi et al (2000) the Thome and Shakir correction factor for mixtures is incorporated with the Gungor and Winterton (1986) correlation. The Jung et al (1993) correlation is evolved based on the experimental data of more than 3000 local heat transfer coefficients of R22, R114, R12, R152a and their mixtures at various concentrations.

The Choi et al (2003) correlation is evolved based on the R407C data; it shows a large deviation with M20 refrigerant mixture heat transfer coefficient; even though R407C occupies a major part in the M20 refrigerant mixture. The Gungor and Winterton (1986) correlation with the correction factor also over predicted the experimental data. The Jung et al (1993) correlation is evolved based on the binary mixtures’ data; however, their correlation largely over predicted the present experimental data.
Figure 5.60 Mean deviation of the predicted local heat transfer coefficient of M20

Figure 5.61 shows a deviation of the predicted results of the Liu and Winterton (1991) and Wattelet et al (1994) correlations with the experimental heat transfer coefficients of the M20 refrigerant mixture. Among these two correlations, the Liu and Winterton (1991) correlation is evolved using the saturated boiling data base, containing 4202 data points for saturated boiling and 991 data points for sub cooled boiling. The Wattelet et al (1994) correlation is based on the data of R12, R134a and a mixture of R22/R124/R152a for both annular and wavy-stratified flows. The operating parameters were varied in the range of mass velocity, 25 to 100 kg m$^{-2}$ s$^{-1}$; heat flux, 2 to 10 kW m$^{-2}$; and saturation temperature, -15°C to 5°C. It is observed that the predicted results of both correlations largely over predicted the experimental heat transfer coefficients in the range of 60% to 80% deviations. Even though the Wattelet et al (1994) correlation is evolved based
on the mixture data, it over predicted the present experimental data. Moreover these two correlations predicted the two phase heat transfer coefficient by considering the convective and nucleate boiling components.

![Figure 5.61 Mean deviation of the predicted local heat transfer coefficient of M20](image)

Figure 5.61 Mean deviation of the predicted local heat transfer coefficient of M20

Figure 5.62 shows the deviation of the predicted heat transfer coefficients of the Gungor and Winterton (1987) and Shah (1982) correlations with the experimental heat transfer coefficients of the M20 refrigerant mixture. The predicted results of these correlations without any correction factor for mixtures are in the range of -45% to +70% deviations with the experimental heat transfer coefficients. These correlations predicted the heat transfer coefficients closer to the present experimental results than those of other correlations considered in this study.

All the considered correlations strongly over predicted the experimental data. The discrepancy in the prediction can be attributed to the
following. None of these correlations normally take into account the change in the physical properties of the mixtures with composition and retardation in the liquid-vapor mass transfer mechanisms. The developed correlations are not for a fin-and-tube evaporator with different tube configurations. The up and down flow of the refrigerant in the circuit, bends in the circuit, and air being used as the heat transfer fluid, are significant factors which may not be taken into account for the existing correlations. Due to the above, in real appliances, the heat transfer coefficient in window airconditioners does not match with the predicted values and it shows such large deviations from the predicted heat transfer coefficients of the M20 refrigerant mixture.

![Graph](Image)

**Figure 5.62** Mean deviation of the predicted local heat transfer coefficient of M20
The experimental heat transfer coefficient is compared with all the correlations as shown in Figure 5.63 to assess their suitability to estimate the local heat transfer coefficient. Comparison has been made for an operating pressure of 6.29 bar, heat transfer rate of 2163.5 W and mass velocity of 225.5 kg m\(^{-2}\)s\(^{-1}\) in a 9.52mm diameter tube. It is also confirmed from Figure 5.63, that the Shah (1982) and Gungor and Winterton (1987), correlations are closer to the experimental heat transfer coefficients than other correlations. Among these two correlations, the Gungor and Winterton (1987) correlation is selected and further modified to fit the present experimental data using the multiple regression method. The poor agreement of these correlations with the experimental results led to the development of a modified model. The development of a modified correlation for the M20 refrigerant mixture and the comparison of the predicted heat transfer coefficients using the modified correlation with the present experimental data are presented in the next section.

![Figure 5.63](image_url)  
*Figure 5.63  Comparison of the experimental and predicted heat transfer coefficient for M20*
5.3.7.7 Modification of Existing Correlations

Based on the study conducted in the previous section, the Gungor and Winterton (1987) and Lavin and Young (1965) correlations are modified to fit the heat transfer coefficient experimental data of M20 refrigerant mixture and R22 respectively. The intercepts and exponents of the dimensionless groups are fitted with respect to these databases using the multiple regression technique.

The original form of the Gungor and Winterton (1987) and Lavin and Young (1965) correlations are shown in equations (5.1) and (5.3). They are suitably modified for the M20 refrigerant mixture and R22 respectively, in terms of the dryness fraction ratio and density ratio in equations (5.1) and (5.3).

\[ h_{tp} = E h_i \]  \hspace{1cm} (5.1)

where

\[ E = 1 + 3000 \times Bo^{0.86} + 1.12 \left[ \frac{x}{1-x} \right]^{-0.75} \left[ \frac{\rho_l}{\rho_v} \right]^{0.41} \]  \hspace{1cm} (5.2)

\[ h_{tp} = h_i 3.79 \left( \frac{1+x}{1-x} \right)^{1.16} Bo^{0.1} \]  \hspace{1cm} (5.3)

where \( h_i \) is the liquid phase heat transfer coefficient, which is calculated using the Dittus-Boelter correlation. The accuracy of these two correlations is mainly based on the Dittus-Boelter correlation, for the calculation of turbulent single phase heat transfer coefficients. The value of the intercept and exponents based on multiple regression analysis of the experimental data are given in Table 5.7. The modified forms of the Gungor and Winterton and Lavin and Young correlations are shown in equations (5.4) and (5.5).

\[ \left[ \frac{h_{tp}}{h_i} - 1 - 3000 \times Bo^{0.86} \right] = A \left[ \frac{x}{1-x} \right]^a \left[ \frac{\rho_l}{\rho_v} \right]^b \]  \hspace{1cm} (5.4)
\[
\frac{h_{tp}}{h_i} = A \left( \frac{1 + x}{1 - x} \right)^a Bo^b \tag{5.5}
\]

**Table 5.7 Intercept and exponents for the modified correlation**

<table>
<thead>
<tr>
<th>Modified correlations</th>
<th>A</th>
<th>a</th>
<th>b</th>
<th>Mean Dev. (%)</th>
<th>R^2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gungor and Winterton</td>
<td>0.234</td>
<td>0.653</td>
<td>0.481</td>
<td>16</td>
<td>0.963</td>
</tr>
<tr>
<td>Lavin and Young</td>
<td>211.942</td>
<td>0.541</td>
<td>0.56</td>
<td>5</td>
<td>0.861</td>
</tr>
</tbody>
</table>

The comparison between the experimental results and predicted results of the modified equation for R22 is shown in Figures 5.64 and 5.65. Figure 5.64 shows the deviation of the predicted heat transfer coefficients of the modified Lavin and Young (1965) correlation with the experimental heat transfer coefficients of R22. It is observed that the modified Lavin and Young (1965) correlation predicted the experimental results in the deviation range of -25% to +20%. The mean deviation has been reduced to 5% for R22 when using the Lavin and Young modified model.

![Figure 5.64 Mean deviation of the predicted local heat transfer coefficient of R22](image-url)
Figure 5.65 presents the original and modified Lavin and Young (1965) model in the same figure, together with the experimental results of R22. The comparison has been made for an operating pressure of 6.26 bar, heat transfer rate of 2330 W and mass velocity of 255 kg m\(^{-2}\)s\(^{-1}\) in a 9.52mm diameter tube. It is found that a good prediction of the experimental results is obtained for R22 using the modified Lavin and Young (1965) correlation and the deviation is within 5%. The modified Lavin and Young (1965) correlation is recommended for the range of applicability shown in Table 5.8.

![Comparison of experimental and predicted heat transfer coefficient of R22](image)

**Figure 5.65  Comparison of the experimental and predicted heat transfer coefficient of R22**

**Table 5.8 Range of applicability for the Lavin and Young modified correlation**

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Ranges</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass velocity</td>
<td>225 to 270 kgm(^{-2})s(^{-1})</td>
</tr>
<tr>
<td>Vapor fraction</td>
<td>0.2 to 1.0</td>
</tr>
<tr>
<td>Heat flux</td>
<td>7049 to 9780 W m(^{-2})</td>
</tr>
<tr>
<td>Pressure</td>
<td>5.42 to 7.10 bar</td>
</tr>
</tbody>
</table>
The test conditions are arrived at, based on the calculated mass velocity, vapour fraction, heat flux and measured pressure at the inlet of the evaporator as given in Tables 5.5 and 5.6.

The comparison between the experimental results and predicted results of the modified equation for the M20 refrigerant mixture are shown in Figures 5.66 and 5.67. Figure 5.66 shows the deviation of the predicted heat transfer coefficients of the modified Gungor and Winterton (1987) correlation with the experimental heat transfer coefficients. It is observed that the modified Gungor and Winterton (1987) correlation predicted the heat transfer coefficients of the M20 refrigerant mixture in the deviation range of -25% to +15%. The mean deviation has been reduced to 16% for the M20 refrigerant mixture when using the Gungor and Winterton modified model over the original one.

Figure 5.66 Mean deviation of the predicted local heat transfer coefficient of M20
Figure 5.67 presents the comparison of the predicted heat transfer coefficient based on the original and modified Gungor and Winterton (1987) correlations with the experimental heat transfer coefficient. Comparison has been made for an operating pressure of 6.29 bar, heat transfer rate of 2163.5 W and mass velocity of 225.5 kg m$^{-2}$s$^{-1}$ in a 9.52mm diameter tube. It is observed that a good prediction of the experimental results is obtained for the M20 refrigerant mixture and the deviation is within 16%. The modified Gungor and Winterton (1987) correlation is recommended for the range of applicability shown in Table 5.9.

![Figure 5.67 Comparison of the experimental and predicted heat transfer coefficient of M20](image)

**Table 5.9** Range of applicability for the Gungor and Winterton modified correlation

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Ranges</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass velocity</td>
<td>212 to 245 kg m$^{-2}$s$^{-1}$</td>
</tr>
<tr>
<td>Vapor fraction</td>
<td>0.4 to 1.0</td>
</tr>
<tr>
<td>Heat flux</td>
<td>6829 to 9217 W m$^{-2}$</td>
</tr>
<tr>
<td>Pressure</td>
<td>5.59 to 7.17 bar</td>
</tr>
</tbody>
</table>
The test conditions are arrived at based on the calculated mass velocity, vapour fraction, heat flux and measured pressure at the inlet of the evaporator as given in Tables 5.5 and 5.6.

In this section, the experimentally estimated local heat transfer coefficients of R22 and the M20 refrigerant mixture are discussed, and based on the measured parameters, the influences of indoor and outdoor air temperature on the heat transfer rate, and local heat transfer coefficients are also analyzed. The experimental heat transfer coefficients of the M20 refrigerant mixture are compared with those of R22. It is found that the heat transfer coefficients of the M20 refrigerant mixture in all the studied operating conditions are lower in the range of 13.6% to 55.9% than those of R22. Further the experimental heat transfer coefficients of both the refrigerants are compared with the existing correlations, based on the grouping of the correlations. Based on the comparison, two existing correlations are selected and modified to fit the present experimental data for the operating conditions considered in this study. The Gungor and Winterton (1987) and Lavin and Young (1965) correlations are modified to fit the M20 refrigerant mixture and R22 experimental heat transfer coefficients respectively.

5.4 GUIDELINES FOR THE DESIGN OF WINDOW AIRCONDITIONER

The following are the design guidelines for the design of window airconditioner operated with pure and refrigerant mixture.

Fin-and-Tube evaporator and condenser:

1. Section-by-section analysis using $\varepsilon$-NTU method.
2. Bend pressure losses are to be considered.
3. Correlation assessment study for both heat transfer coefficient, pressure drop are to be carried out.

4. Continues variation of air and refrigerant temperature, temperature difference between air and refrigerant, heat transfer rate are to be calculated.

5. Selection of tube flow configuration either parallel, counter, z-type or zig zag type.


7. Wet and non-wetted condition of evaporator to be considered.

Capillary:

1. Entrance effect on refrigerant pressure drop is to be considered.

2. Effect of subcooling by considering metastable region.

3. Calculation of pressure drops in single and two phase region.

4. Assessment study on selection of two phase viscosity and friction factor model to be carried out.

5. Effect of inlet and outlet pressure on choked flow to be considered.

6. Heterogeneous flow mode to be considered.

Compressor:

1. Energy balance in compressor.

2. Calculation of pressure rise and pressure drop inside the compressor with change in evaporator and condenser air inlet temperature.

3. Calculation of mass of the refrigerant with change in evaporator and condenser air inlet temperature.
4. Work of compression, discharge temperature with change in evaporator and condenser air inlet temperature are to be calculated.

5. An unsteady state analysis to evaluate the combined effect of heat transfer inside the cylinder and the valve dynamics on the refrigerant behavior for alternative refrigerants to R22.

5.5 SCOPE FOR FUTURE RESEARCH

i) The condensation characteristics of the M20 refrigerant mixture can be studied comprehensively under different mass velocity, heat flux and saturation temperatures to evolve its respective heat transfer coefficients and correlate a separate heat transfer coefficient correlation.

ii) The experiment can be further extended to different flow configurations of a fin-and-tube evaporator and condenser, so that the required charge quantity, capillary length and condenser length can be recommended more accurately for the specific cases.

iii) The experiment can be further extended to a fin-and-tube condenser with micro-fin tubes instead of a smooth tube, so that the capacity of the M20 refrigerant mixture may be increased, which can be ascertained with a theoretical study also.

iv) The thermodynamic properties of M20 refrigerant mixture can be developed and the entropy generation analysis of the window air conditioner using M20 could be carried out.