CHAPTER 2

LITERATURE SURVEY

2.1 INTRODUCTION

Due to the ongoing global phase out of R22, which is still the most widely used refrigerant around the world, there is a need to replace this refrigerant in window airconditioning applications. A thorough literature review of the R22 replacement options for window airconditioning applications is necessary and it is carried out in this chapter. Moreover, it is observed from literature that the M20 refrigerant mixture is one such alternative with specific advantages as discussed in the previous chapter. It is also observed that the M20 refrigerant mixture works with mineral oil while R407C demands POE oil as the compressor lubricant. In the present chapter, literatures related to a performance study using the R407C and M20 refrigerant mixture is also reviewed.

The thermo-physical properties of refrigerants will play a major role in the performance of vapor compression systems. In general, system simulations are used to understand the relationship between refrigerant properties and system performance. A brief literature review is to be carried out to find out the parameters and methods to be considered for the proper simulation of a fin-and-tube evaporator used in a window airconditioner. For a more accurate comparison of refrigerants, information on the heat transfer characteristics of the M20 refrigerant is required in practically realized heat flux conditions. Furthermore, procedures for measuring the local heat transfer
coefficients in flow boiling are needed for design purposes using the M20 refrigerant mixture. The possibility of improving the performance (Refrigerating capacity) by varying the charge quantity is also investigated to maximize the COP of the M20 refrigerant mixture in retrofit condition.

In this chapter a comprehensive survey of the previous studies on the simulation of a fin-and-tube evaporator, the performance of alternative refrigerants, the heat transfer coefficients of pure and refrigerant mixtures, and the performance deviation of refrigerants over a range of charge quantities in an air conditioning system, are presented.

2.2 ALTERNATIVE REFRIGERANTS FOR R22

Domanski et al (1991) have reported a comparable evaluation of isobutane (R600a), propane (R290), R134a, R22, R410A, and R32 in an optimized fin-and-tube evaporator and analyzed the impact of the evaporator effects on the system COP. The fin-and-tube evaporator circuits were optimized using a non-Darwinian Learnable Evolution model. All the simulations were carried out at the evaporation temperature of 7°C and the condenser temperature of 38°C and 45°C. It was found from a theoretical cycle analysis that the COP of R600a was 5.3% better than that of R22 and the COP of R32 was 5.1% lower than that of R22. The COP of R290 was better than that of R22 by up to 3.5%, while including evaporator effect.

Kondepudi (1993) conducted an experiment in a 2TR split system air conditioner using five different refrigerant blends of R32 with R134a and R152a as drop-in alternatives to R22. All the tests were conducted in accordance with ARI rating conditions and the capacity, efficiency, and seasonal efficiency were calculated. The experimental results were compared with the simulated results obtained from the existing literature. The experimental results showed that the R32/R134a blends performed within 2%
of efficiency and with comparable capacity to that of R22. The 40%/60% blend of R32/R134a performed the best in a non-optimized system.

Radermacher and Jung (1993) have theoretically calculated the COP and seasonal performance factor for binary and ternary refrigerants. The following binary and ternary mixtures were considered as substitutes to R22: such as R32/R134a, R32/R152a, R32/R134, R32/R124, R143a/R134a, R143a/R152a, R143a/R124, R125/R134a, R125/R152a, R125/R124, R32/R152a/R134a, R32/R152a/R134 and R32/R152a/R124. Simulations were carried out in accordance with the ASHRAE (1983) testing and rating procedure. The COP of the best ternary mixture R32/R152a/R124 was 13.7% higher and the volumetric capacity was 23% lesser than that of R22. The COP of the best binary mixture R32/R124 was 13.4% higher and the volumetric efficiency was 9.6% lesser than that of R22. The performance comparison of a pure refrigerant with R22 showed that R152a, R134 and R134a could be used as replacement refrigerants.

Bivens et al (1994) have conducted experiments in two split-system heat pumps and a window air-conditioning unit using R407C. The experiments were carried out in accordance with ARI test conditions (A, B, high-temperature and Low Temperature Heating). It was observed that the energy efficiency of R407C was 3% to 4% lower in the cooling condition and 5% to 6% lower in the heating condition than that of R22. It was also observed that the compressor pressure of R407C was 4% to 14% higher than that of R22 depending on the system and operating conditions. The benefits of liquid line/suction line heat exchange and counter flow air-to-refrigerant heat exchangers were investigated theoretically.

Kim et al (1994) have presented a performance evaluation of a generic heat pump using two refrigerant mixtures of R134a with R290 and R600a; R290/R134a (45/55 by mass %) and R134a/R600a (80/20 mass %).
The experimental results were compared with those of pure R12, R134a, R22 and R290 at high temperature cooling and heating conditions. It was found that the COP of R290/R134a was lower than that of R22 and R290. The COP of R134a/R600a was higher than that of R12 and R134a. The capacity for R290/R134a was higher than that of R22 and R290 and R134a/R600a shown higher capacity than that of R12 and R134a.

Ravindra (1995) conducted a capacity and performance testing of a packaged airconditioner in a psychrometric test facility. The construction details of the psychrometric test facility and the DBT and WBT measurements inside the rooms were explained as directed by the standards. Air enthalpy or the psychrometric test method was explained for the calculation of the capacity of the system. The construction of an airflow tester was discussed for the measurement of the quantity and dry-bulb and wet-bulb temperatures of the air discharged from the UUT. The parameters to be controlled and the other parameters measured during the test were listed out in that work. The procedure for capacity rating and other tests as described in IS: 8148-1976 was explained.

Richardson and Butterworth (1995) have investigated experimentally the performance of hydrocarbon refrigerants in a hermetic vapor-compression system. Experiments were conducted using R12, propane and a binary mixture of propane/isobutene in various proportions. It was observed that the COP of the propane/isobutene (56/44%) mixture was higher than that of R12, while the propane/isobutene (43/57%) mixture COP was better at the evaporation temperature of above -10°C.

Wel et al (1997) have reported the drop-in system performance comparison of refrigerants R22 and R407C. The experiments were carried out in accordance with standard rating conditions, overload, and low temperature conditions. In order to reduce the system discharge pressure, the capillary
length was reduced and it was proved that cutting the length of the capillary tube could not effectively reduce the condensing pressure at the optimum performance point. It was found that the cooling capacity of R407C was 3% lower and the COP was 4% lower than that of R22. It was observed that the condensing pressure for R407C was 200 to 300 kN m\(^{-2}\) higher than that of R22.

Mei et al (1998) have performed a drop-in test with R407C in an off-the-shelf 2TR window air conditioner having an energy efficiency ratio of 10 using a parallel-cross-flow (PCF) and counter-cross-flow (CCF) evaporator. The test results indicated that at the ARI rated indoor and outdoor conditions, the cooling capacity was 8% higher and the COP about 3.8% higher for the CCF evaporator than that of the PCF evaporator. The experimental results also showed that the latent load for the CCF was 30.6% higher than that for the PCF evaporator using R407C and was 16.6% better than operations using R22.

Purkayastha and Bansal (1998) have presented an experimental study on the performance of R22, HC290 refrigerants and a liquefied petroleum gas mix. The experiments were carried out in a heat pump test facility with a condenser capacity of 15kW and the condensing temperatures were held constant at 35, 45 and 55°C, while the evaporating temperatures were varied from -15°C to 15°C. The mass flow rate and compressor discharge temperature of the mixture were lower than that of R22. It was found that the performance of the LPG mix was better than that of HC290 at higher condensing temperature, but lower at lower condensing temperature. The discharge temperature of HC290 and the LPG mix was much lower than that of R22 over the entire range of operating conditions.

Yang Zhao et al (1999) have investigated experimentally and theoretically the performance of some new refrigerant mixtures
R32/125/152a, R125/290, R32/290, R32/125/290 as alternatives for R22. The operating conditions cover the evaporator temperature range of -35°C to 10°C and condenser temperature ranges from 30°C to 60°C. It was concluded that the performance of the substitute R32/125/152a mixtures was close to that of R22 over the entire range of operating conditions, which had better efficiency under varying operating conditions.

Jung et al (2000) have tested 14 refrigerant mixtures composed of R32, R125, R134a, R152a, R290 (propane) and R1270 (propylene) in a breadboard heat pump of 3.5 kW capacity, in an attempt to find out R22 substitutes used in residential airconditioners. Water was used as a heat transfer fluid in the evaporator and condenser and all the tests were conducted as per ARI test conditions. It was observed that the ternary mixtures composed of R32, R125, and R134a had a 4.5% higher COP and capacity than that of R22. Another ternary mixture containing R125, R134a and R152a had both lower COP and capacity than that of R22. R32/R134a binary mixtures show a 7% increase in COP with a similar capacity to that of R22, while R290/R134a azeotrope shows a 3 to 4% increase in both COP and capacity. The compressor discharge temperatures of the mixtures tested were 15.9 to 34.7°C lower than those of R22.

Devotta et al (2001) have identified HFC134a, HC290, R407C, R410A and three blends of HFC32, HFC134a and HFC125 refrigerants and assessed their suitability as alternatives to R22 for air conditioner. The NIST CYCLE-D software was used for the comparative thermodynamic analysis. The variation of pressure ratio, discharge temperature, specific compressor displacement, compressor power and COP of all the selected refrigerants were compared with those of R22 refrigerant at different evaporating and condensing temperatures. It was concluded that the characteristics of R290 are very close to those of R22, and the compressors require very little
modification. It was concluded that for retrofitting, R407C was probably the best candidate.

Devotta et al (2005a) have presented the experimental performance analysis of a 1.5TR window air conditioner retrofitted with R407C, as a substitute for R22. All the tests were conducted as per the Indian Standard 1391 (1992) Part 1. It was observed from the experimental results of R407C, that the 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. The discharge pressures for R407C were higher in the range of 11 to 13% than those of R22. The simulation was carried out using the EVAP-COND (NIST) software and the results were compared with the experimental results.

Devotta et al (2005b) have presented the experimental performance study of a window air conditioner with propane (HC290), as a drop-in substitute to R22. Experimental results showed that HC290 had 6.6% to 9.7% lower cooling capacity, 2.8% to 7.9% higher COP, 12.4–13.5% lower energy consumption with respect to R22. The discharge pressures for HC290 were lower in the range 13.7–18.2% and the pressure drop was lower than those of R22 in both the heat exchangers. The simulation was carried out using the EVAP-COND (NIST) software and the results were compared with the experimental results. It was also concluded that the outlet temperatures of air for R22 and HC290 in both heat exchangers are nearly the same.

Chen (2008) has developed four sets of comparable R410A and R22 split-type residential air conditioners model using a design and development simulation software and studied their performance. The cooling capacity, energy efficiency ratio, annual power consumption of airconditioner and the global warming impact of refrigerants adopted by the airconditioner were compared. It was concluded that the adoption of R410A could be helpful for
airconditioner to decrease their heat exchanger size or improve their operation efficiency for power saving.

It was observed from literatures that some binary mixtures and ternary mixtures were considered as alternative refrigerants for R22. The performance of the system was characterized by mixture ratio, COP and evaporator air inlet temperature. Comparisons were made between the pure refrigerants and refrigerant mixtures on the basis of the COP. Moreover, it was observed that for retrofitting a R22 system, R407C is the best refrigerant, but POE oil had to be used instead of the conventional mineral oil. POE oil is highly hygroscopic. It was found that in all the conducted experimental works the performance of the R407C system was lower than that of R22. It was also observed that R290 can be used as an alternative for R22, even though its cooling capacity is lower than that of R22. The obtained results indicated that the COP of the system with R290 is better than that of R22. The main problem concerned with R290 is its flammability nature.

### 2.3 HFC/HC REFRIGERANT MIXTURE

Joseph Sekhar et al (2003) have conducted a performance test in a domestic refrigerator (165-litre) attached with a secondary refrigerant calorimeter charged with R12 and HFC134a/HC290/HC600a mixture without changing the mineral oil in the compressor. The system was operated with a thermostat setting of −15°C in the freezer, 0 to 7°C in the food compartment and 8 to 14°C in the crisper compartment. It was concluded that the energy consumption of the refrigerant mixture was 4 to 11 % lesser and the actual COP was 3 to 15% higher than that of R12. The power consumption was 2 to 3% higher than that of R12.

Joseph Sekhar et al (2004a) have carried out an experimental analysis in a 165 litre CFC12 household refrigerator retrofitted with an
eco-friendly refrigerant mixture of HFC134a/HC290/HC600a (M09) without changing the mineral oil. The experiment was conducted for more than 12 months at various ambient conditions of 24, 28, 32, 38 and 43°C and it was conformed that the new mixture was compatible to mineral oil. It was also found that the energy consumption of the new mixture was 4 to 11% lower than that of R12 and the actual COP was 3 to 8% higher than that of R12.

Joseph Sekhar et al (2004b) have studied experimentally the performance of an ozone friendly refrigerant mixture HFC134a/HC290/HC600a and R12 in a walk-in cooler, operating with an open type compressor with mineral oil as a lubricant. The no-load pull down time, motor power, energy consumption, ON and OFF cycle time, temperature distribution along the coil and COP test were conducted at the ambient temperature of 32°C± 1°C for both the refrigerants and the results were compared. It was observed that for the M09 refrigerant the actual COP was 9.5% higher, the energy consumption was 28% lesser and the compressor outlet temperature was 5-25°C lower than that of R12.

Jabaraj et al (2006 and 2007) have studied the possibility of using HFC407C/HC290/HC600a, a refrigerant mixture as a substitute for R22 in a window air conditioner, and to evolve an optimal composition of the mixture. Experiments were carried out in a room calorimeter set up fitted with 1050 W capacity window air conditioner. Condenser inlet air temperatures were held constant at 30, 35, 40 and 45°C, while evaporator inlet air temperatures were varied over a range viz. 21, 23, 25, 27 and 29°C. The HC percentage with R407C was varied from 10 to 25% in steps of 5%. It was observed that the new refrigerant mixtures demand longer condenser length for matching with R22 systems. The condenser tube length was increased by 19% to suit the mixtures as compared to R22. The performance analysis revealed that the new refrigerant mixture of M20 had 8.2 to 11.1% better actual COP and the energy
consumption was 5 to 10.5% lower than that of R22 at all considered condenser inlet air temperatures. The discharge temperature of M20 refrigerant was 3.7% to 11.5% lower and the energy consumption was 5.0% to 10.4% lower than that of R22.

The above reviewed literature revealed that M20 is one of the alternative refrigerants for R22. However, the present work is an attempt to make use of this M20 mixture without changing the condenser (as a retrofit). Hence, a detailed study under psychrometric test conditions as per BIS and ASHRAE standards is required. It is to be noted that the earlier work reported does not consider BIS and ASHRAE test conditions to maintain the WBT along with DBT.

2.4 PERFORMANCE OVER A RANGE OF CHARGE QUANTITY

O’Neal and Farzad (1990) have conducted an experiment on a 10.6 kW capacity split-system air conditioner with capillary tube expansion at different charge levels and different outdoor conditions, and determined its effect on the system performance. It was found that the degradation in performance for each of the variables was generally greater in undercharged than in overcharged conditions. Refrigerating capacity, power consumption, COP and seasonal COP were considered for the analysis. The results indicated that undercharging by as little as 5% can cause as much as a 6.1% drop in the seasonal coefficient of the performance.

Farzad and O’Neal (1991) have quantified the influence of the refrigerant charge on the performance of 3 TR split air conditioner with capillary tube expansion. The effects of the refrigerant charge on system variables such as capacity, flow rate, evaporator superheat, power consumption and seasonal energy efficiency ratio were discussed. It was
reported that a larger degradation in system capacity was experienced in undercharging than for overcharging in the studied operating conditions.

Robinson and O’Neal (1994) have studied the effect of refrigerant charge quantity on the performance of three blends of R134a and R32 with different mass proportion in an air-to-air heat pump. The COP and capacity variations were analyzed for different charge and outdoor air temperatures. For all the tests the indoor air temperature was maintained at 26.7°C, 19.4°C (DBT, WBT) while the outdoor room conditions were maintained at 27.8°C, 18.3°C (DBT, WBT), 35°C, 23.9°C (DBT, WBT) and 40.6°C, 28.3°C (DBT, WBT). It was observed that the optimum COPs of the mixtures were lower than that of R22. It was found that a shift in the charge produced the optimum COP for the 40.6°C outdoor temperature compared to either the 27.8°C or 35°C outdoor temperature.

Choi and Kim (2002) have investigated the effects of an off-design refrigerant charge on the performance of a water-to-water heat pump by varying the R22 refrigerant charge amount from -20% to +20% of full charge with electronic expansion valve and capillary. The water temperature entering the evaporator was kept at 25°C, and the condenser was maintained at 30°C, 34°C, 38°C, and 42°C. It was reported that the capillary system was more sensitive to an off-design refrigerant charge than that of EEV. The degradation of performance was higher in undercharged conditions than that at an overcharged condition for the capillary system.

Choi and Kim (2004) have investigated the effects of the expansion device on the performance of a water-to-water heat pump using R407C in various charging conditions. It was reported that the degradation of capacity and COP was lower for the R407C EEV system when the refrigerant charge deviated from the optimum charge as compared to the R22 and R407C capillary tube systems.
Corberan et al (2008) have conducted an experiment on a water-to-water heat pump and analyzed the effect of the refrigerant charge variation on the performance of the system. The influence of the charge on the capacity and COP of the unit in the given operating conditions was analyzed and then the causes of such influence. It was concluded that the system performance was highly dependent on the refrigerant charge quantity.

The reviewed literatures elucidated that an undercharge or overcharge of the refrigerant into the system will degrade its performance. In addition, the amount of refrigerant charge in the system is another primary parameter influencing energy consumption. Experiments were carried out by varying the refrigerant charge to characterize the effects of refrigerant charge quantity on the performance of the system. Moreover, it was found that the capillary tube system was relatively more sensitive to refrigerant charge and outdoor load conditions than other expansion devices. In general, for the capillary tube system the degradation of the performance was higher in undercharged conditions than that at overcharged conditions. Such charge deviations are not unusual when systems are retrofitted. Hence a study on the performance over a range of charge quantity is carried out.

2.5 HEAT TRANSFER CHARACTERISTIC STUDY IN A FIN-AND-TUBE EVAPORATOR

Sheffield et al (1989) have conducted an experimental study to investigate the contact conductance of plate finned tubes. The basic theory of thermal contact conductance supports the use of measurable parameters including interference, fin spacing, fin thickness, tube hardness, and tube diameter as prediction parameters. 31 coils were tested in a vacuum chamber. A correlation was developed based on the experimental database to predict the thermal contact conductance.
Webb (1990) has analyzed the heat transfer characteristics of heat exchangers having flat and wavy plate fins on a staggered array of circular tubes. 20 wavy plate fin geometrics and 7 flat plate fin geometrics were tested in different operating conditions. Based on the experimental results correlations were developed using the multiple regression technique to predict the air-side heat transfer coefficients as a function of the flow conditions and geometrics of the heat exchangers. The predictability of the evolved correlation was compared with the experimental results. It was concluded that the developed correlation predicts 98% of the flat fin data within a ±5% deviation and 88% of the wavy fin data within a ±5% deviation.

Wang et al (2000) have proposed a heat transfer and friction correlation for a fin-and-tube heat exchanger having plain fin geometry. A total of 74 samples were used to develop the correlation. The proposed heat transfer correlation had absorbed the contact conductance in the development of the correlation. The proposed heat transfer correlation could describe 88.6% of the database within ±15%, while the proposed friction correlation could correlate 85.1% of the database within ±15%. The mean deviation of the heat transfer correlation was 7.51%, while that for the proposed friction correlation was 8.31%.

Wang et al (2001) have studied experimentally the effect of circuitry on the refrigerant-side pressure drops of plate finned tube evaporators. Experiments were performed with counter cross, parallel-cross, and z-shape arrangements. The results showed that the parallel cross-flow circuit gives a lower pressure drop than other arrangements. Generally, the refrigerant-side pressure drops increase with air frontal velocities. However, for mass flux of 200 kg m$^{-2}$ s$^{-1}$ and parallel flow, the pressure drops decrease with increase of air frontal velocity. It was concluded that this unusual characteristic is most likely related to the flow pattern transition when subjected to heat addition.
Somchai et al (2002) have investigated experimentally the two phase heat transfer coefficient of R134a evaporating inside a fin-and-tube evaporator with plain fin geometry. The experimental apparatus consisted essentially of a well modified vapor compression refrigeration system. The test runs were done at average saturated refrigerant temperatures ranging between 4.0 and 9.0°C for evaporation and between 39.0 and 46.0°C for condensation. The refrigerant mass flow rates were between 7.6 and 9.0 g/s. The volumetric flow rate of air passing through the evaporator ranges between 0.25 and 0.5 m³ s⁻¹ and between 0.7 and 1.25 m³ s⁻¹ for the condenser. Based on the experimental data the tube side evaporation heat transfer coefficients were determined and a new correlation was proposed. The present results were compared with those calculated from the correlation reported in literature.

Most of the literature focused on a heat transfer and friction correlation study for the fin-and-tube heat exchanger having plain fin geometry. Based on the experimental results correlations were developed to predict the air-side heat transfer coefficients as a function of the flow conditions and geometrics of the heat exchangers. The limited literature investigated experimentally the two phase heat transfer coefficient characteristics of a refrigerant evaporating inside a fin-and-tube evaporator with plain fin geometry.

2.6 FIN-AND-TUBE HEAT EXCHANGER SIMULATION

Domanski (1991) has presented a simulation model of a plate fin-air-to-refrigerant heat exchanger used as an evaporator in residential airconditioning. The model was based on a tube-by-tube approach. The performance of each tube was analyzed separately by considering the cross flow heat transfer with the external air stream and the appropriate heat and mass transfer relationship. A comparison of the prediction and experimental
test was provided in that study. It was reported that air maldistribution might induce maldistributions of the refrigerant, which contributed to the performance degradation of the evaporator.

Bansal and Purkayastha (1998) have presented a steady state simulation model to predict the performance of alternative refrigerants in vapor compression refrigeration/heat pump systems. The presented model was based on the $\varepsilon$-NTU method following an elemental approach. The physical details of heat exchangers, compressor efficiency, mass flow rates, inlet temperatures to the evaporator and condenser, the pressure drops across the heat exchangers and the capacity of either the evaporator or condenser were given as the input parameters. The model results were validated with a wide range of experimental data of R22 and Propane on a heat pump test facility for a number of parameters, such as the COP, condenser capacity, mass flow rate of the refrigerant and compressor discharge temperature.

Corberan and Melon (1998) have developed a model to predict the behavior of finned tube evaporators and condensers working with R134a. The most recommended correlations in the reviewed literature were analyzed and compared with the experimental results. The experimental study to validate the model was carried out in a small air-conditioning unit with cross-flow air-refrigerant type heat exchangers. The experimental results were compared with the model predictions for thermal capacity, refrigerant superheat or subcooling, and tube-side pressure drop.

Domanski (1999) has presented a fin-and-tube evaporator theoretical model based on $\varepsilon$-NTU tube-by-tube scheme. The complexity of the refrigerant circuit, refrigerant distribution between all circuits and non-uniform air distribution were considered. The local parameters for each tube such as inlet and outlet quality, temperature, enthalpy, entropy, pressure drop,
mass flow rate for refrigerant and inlet and outlet temperature for air could be calculated using the developed model.

Horuz (1998) has investigated theoretically and experimentally the performance of evaporators using R134a, R717, R22, R502 and R12 refrigerants. During experimentation the following parameters were considered, such as air velocity, fin spacing, tube diameter, evaporator temperature, refrigerant type and frost height. A good agreement between the experimental and theoretical results was obtained and it was concluded that the theoretical heat transfer coefficients were 15 to 30 % higher than those of the experimental data. It was proved that the heat transfer varied more with the variation of the air mass flow rate passing through the evaporator coils.

Choi et al (1999) have presented a pressure drop correlation for evaporation and condensation in smooth and microfin tubes. The correlation was developed from a database consisting of pure and mixed refrigerants such as: R125, R134a, R32, R410A, R22, R407C and R32/R134a (25/75 % mass). The new correlation was obtained by replacing the friction factor and the tube-diameter in the Bo Pierre correlation with a friction factor derived from the pressure drop data for a micro-fin tube and the hydraulic diameter, respectively. The new correlation predicted the measured pressure drop a in smooth tube with an average residue of 15.0 %. In addition, the correlation was used to predict pressure drop data for the refrigerant/lubricant mixtures by using a viscosity-mixing rule.

Ould Didi et al (2002) have presented two-phase pressure drop data for evaporation in two horizontal test sections of 10.92 and 12.00 mm diameter for five refrigerants (R134a, R123, R402A, R404A and R502) over mass velocities from 100 to 500 kg m$^{-2}$ s$^{-1}$ and vapor qualities from 0.04 to 1.0. The experimental results were compared against seven two-phase frictional pressure drop prediction methods. It was concluded that the method
by Muller-Steinhagen and Heck was found to provide the most accurate predictions for annular flow and the method of Gronnerud was the best fit for both intermittent and stratified-wavy flows.

Lee et al (2002) have performed an experimental study of a fin-and-tube condenser using two different configurations of condenser paths (U and Z type) and two kinds of refrigerants R22 and R407C. A numerical code was developed, using a section-by-section analysis scheme in which mal-distribution on the air side and temperature gliding on the refrigerant side were considered along the tube-length direction. In the Z-type configuration R22 performed better than R407C, but no significant difference was found between results using either refrigerant in the U-type path configuration. The numerical results obtained with R22 were 10.1% higher than the experimental data; using R407C, the results were 10.7% lesser than experiment data.

Lee et al (2003) have developed a program to analyze the heat transfer characteristics of fin-and-tube evaporators using R407C and R22. The tube was segmented into several sections and the ε-NTU method was used. The temperature gradient in the flow direction and two dimensional air mal-distribution were traced, using a discrete pattern to simulate the continuous variation found in actual evaporators. Experiments were carried out with 45 real evaporators having two different refrigerant flow path configurations, using the R22 and R407C refrigerants. The local heat transfer predictions were verified by comparing the numerical and measured wall temperatures along the refrigerant flow path.

Choi et al (2003) have implemented an experimental investigation to determine the effect of a non-uniform refrigerant and air flow distributions on capacity degradation. The refrigerant distributions between individual refrigerant circuits were controlled to assess the potential to recover the lost capacity. The tests were performed on a three-circuit, three depth-row, finned-
tube evaporator and the refrigerant inlet quality, exit saturation temperature, and exit superheats for the individual circuits were controlled. The study showed that the capacity degradation due to refrigerant mal-distribution could be as much as 30%. The maximum capacity degradation was found to be 8.7% for the coil and air mal-distributions studied. A 4.0% capacity recovery was obtained by controlling the refrigerant distribution to obtain the target 5.6 °C at each circuit exit.

Domanski and Hermes (2006) have proposed a new correlation for a two-phase flow pressure drop in 180° return bends based on a total of 241 experimental data points for R22 and R410A. The following parameters were considered for the prediction of the correlation; smooth tubes with inner diameters \(D\) from 3.25 mm to 11.63 mm, bend radii \(R\) from 6.35 mm to 37.25 mm, and curvature ratios \(2R/D\) from 2.32 to 8.15. The correlation predicted all the data with a mean deviation of 15.7%.

The reviewed literatures revealed that many researchers have developed distributed parameter models to analyse the steady state performance of fin-and-tube heat exchangers. However, most of the models are only suitable for fin-and-tube heat exchangers with a simple tube arrangement. Domanski (1999) developed a tube-by-tube technology that is capable of analyzing the performance of a heat exchanger with complex refrigerant circuitry. This method was further developed by Lee et al (2002) to study two-dimension air distribution. The Lee et al (2002) model has been used in the present study for the simulation of a fin-and-tube evaporator using the new M20 refrigerant mixture.
2.7 LOCAL HEAT TRANSFER COEFFICIENT OF REFRIGERANTS

Shah (1982) has proposed a correlation for flow boiling in vertical and horizontal tubes and annuli. In this model the flow boiling regime was divided into three distinct regions: a nucleate-boiling-dominated regime, a bubble-suppression regime, and a convective-dominated regime. 800 data points from 18 independent experimental studies were used to evolve the correlation. The evolved correlation have been verified with some 3000 data points for 12 different fluids up to a reduced pressure of 0.89 and were found to be in good agreement with the experimental results.

Gungor and Winterton (1986) have developed a new general correlation for forced convection boiling and subcooled boiling with the aid of a large existing literature database. The data base consists of over 4300 data points covering seven fluids and 28 authors, mostly for saturated boiling in vertical and horizontal tubes, but with significant information also for subcooled boiling and for annuli. The data points taken from the literature consist of the experimentally measured values of the heat transfer coefficient and wall temperature as a function of pressure or saturation temperature, mass flux, heat flux and quality. It was concluded that the mean deviation between the predicted and measured boiling heat transfer coefficient was 21.4% for saturated boiling and 25.0% for subcooled boiling.

Gungor and Winterton (1987) have developed an improved general correlation for flow boiling in tubes and annuli, for both vertical and horizontal orientation by using a large data base. An attempt was made to collect data from a wide range of sources taken under a wide range of experimental conditions. The Birmingham data base consisting of 4202 data points for saturated boiling and 946 data points for subcooled boiling were
used for the development of the new correlation. A few of the existing correlations were compared with the data base and the present correlation.

Ross et al (1987) have determined experimentally the heat transfer coefficients for pure R152a and R13Bl, and for four mixtures of these refrigerants under a wide range of conditions in a horizontal stainless steel tube. The test was conducted for a heat flux of 10 to 95 kW m$^{-2}$, a mass flux of 150 to 1200 kg m$^{-2}$ s$^{-1}$, a composition of 0 to 1.0 and several intermediate values, a pressure of 1.7 bar to 8.0 bar and a quality of 0 to 1.0. The effect of full suppressed nucleate boiling was analyzed for pure and mixture refrigerants. It was explained that, based on the correlative evidence the full suppression of nucleate boiling was easier to achieve with mixtures than with pure fluids. The mixtures yielded sharply lower heat transfer coefficients than either pure refrigerant. The existing correlations were compared with the data successfully, for both pure and mixed fluids.

Jung et al (1989) have studied the mixture effects on horizontal flow boiling heat transfer, using more than 2000 local heat transfer coefficients of both an azeotropic R12/R152a mixture and non-azeotropic R22/R114 refrigerant mixture. In a convective evaporation region, a low mass transfer resistance was found for mixtures, because of the variation of physical properties due to mixing inside the tube. It was reported that the severe degradation of heat transfer with mixtures was found in a partial boiling region, due to the suppression of nucleate boiling at lower qualities, because of the loss of wall superheat with mixtures. Using the experimental results a correlation based on the supposition of Chen (1966) and using only phase equilibrium data to consider mixture effects, was developed with mean deviations of 7.2 and 9.6% for pure and mixed refrigerants.

Jung et al (1989) have reported an experimental study on horizontal flow boiling heat transfer for pure R22, R114 and their mixtures under a
uniform heat flux condition. More than 1200 local heat transfer coefficients were obtained for annular flow at a reduced pressure of 0.08, heat flux ranges from 10 to 45 kW m\(^{-2}\) and mass flow rates of 16 to 46 g s\(^{-1}\). The results indicated that a full suppression of nucleate boiling for pure and mixed refrigerants beyond transition qualities occurred and the majority of the data belongs to the convective evaporation region. It was found that the heat transfer coefficients of mixtures in the convective evaporation region were as much as 36% lower than the ideal values under the same flow condition. It was concluded that the non-ideal variations in physical properties account for 80% of the heat transfer degradation seen with mixtures and the other 20% (less than 10% of the heat transfer coefficient) was believed to be caused by the mass transfer resistance in the convective boiling region. It was measured that a composition variation of up to 0.07 mole fraction in the annular liquid film between the top and bottom of the tube, causes a corresponding circumferential variation of wall temperature with mixtures.

Kandlikar (1990) has refined the earlier correlation by expanding the data base to 5246 data points from 24 experimental investigations with the working fluid of water, R11, R12, R13B1, R22, R113, R114, R152a, nitrogen and neon. The proposed correlation along with the constants gave the mean deviation of 15.9% with water data and 18.8% with all refrigerant data and it also predicts the correct \(h_{\text{tp}}\) versus quality trend as verified with water and R113 data. It was suggested that the proposed correlation could be extended to other fluids by evaluating the fluid dependent parameter \(F_{\text{fl}}\) for that fluid from its flow boiling or pool boiling data.

Liu and Winterton (1991) have developed a correlation dependent on the Prandtl number in the convective term. Thus, an accurate predictive method covering a very wide range of parameters was constructed with an explicit nucleate boiling term and without boiling number dependence. Most
of the recent accurate correlations for predicting the heat transfer coefficient in saturated flow boiling, contain an empirical boiling number correction, since the boiling number correction allowed the enhancement of the forced convective heat transfer mechanisms arising from the generation of vapor in the boundary layer next to the wall. The proposed correlation was compared with the existing successful general correlations such as Shah (1982), Gungor and Winterton (1987 and 1986) and Chen (1966). The present correlation predicted much more evenly with all the fluids involved, than the other correlations.

Jung and Radermacher (1993) have reported a study of the prediction of the heat transfer coefficient and pressure drop of refrigerant mixtures on the same cooling capacity basis, assuming evaporation in horizontal tubes using prospective mixtures to replace R12 and R22. The obtained results indicated that nucleate boiling was suppressed at qualities greater than 20% for all mixtures, and convective evaporation becomes the main heat transfer mechanism. For the same capacity, the effect of volatility difference between the refrigerant mixtures and mass flow rates on the heat transfer coefficient was analyzed. The overall impact of heat transfer degradation due to heat transfer resistance which exists in the heat transfer fluid side and refrigerant side was also discussed. It was also concluded that the transport properties of the liquid affect heat transfer more than the other properties.

Murata and Hashizume (1993) have investigated experimentally the heat transfer coefficient and pressure drop of the pure refrigerant R123 and a mixture of R123 and R134a. It was found that the heat transfer coefficient of the mixture was lower than that for an equivalent pure refrigerant with the same physical properties, not only in the boiling dominant region but also in the convection dominant region. It was observed through the sight glass, that the flow pattern was annular or semi-annular in the region $x \geq 0.2$ except for a
mass flux of 100 kg m$^{-2}$s$^{-1}$. It was found that the heat transfer coefficients were dependent on heat flux in the low quality and low mass flux region, but the dependence was less significant in the high quality and mass flux region. Based on the experimental results correlations were proposed for pure and refrigerant mixtures.

Takamatsu et al (1993a) have carried out experiments on the boiling heat transfer of nonazeotropic refrigerant mixtures of R22/R114 flowing inside a 7.9 mm horizontal smooth tube. Using a water-heated, double tube type evaporator the local heat transfer coefficients were measured for both the counter and parallel flows. An empirical correlation equation was proposed as a sum of nucleate and convective boiling contribution for the data in the annular-flow regime, taking into account the mixture effect. It was reported that the mean deviation between the calculated and measured heat transfer coefficients was 8.9% for the present experimental data. The equation reveals that the convection heat transfer was dominant due to the suppression of nucleate boiling for the mixtures.

Takamatsu et al (1993b) have reported an experimental study of the boiling heat transfer of R134a, R22, R114 and R12 flowing inside a 7.9 mm inner diameter horizontal smooth tube. Using a water-heated, double-tube type evaporator, the local heat transfer coefficients were measured for both the counter and parallel flows. The experiments were conducted for the operating parameters of mass velocity 100, 200, 200 and 350 kg m$^{-2}$s$^{-1}$ and the reduced pressure varied from 0.13 to 0.23. A correlation based on the experimental results was proposed for the data in the annular-flow regime. It was reported that the mean deviation between the calculated and measured heat transfer coefficients was 12.2% for the present experimental data and 9.5% for the data available from literatures. The proposed correlation shows that the nucleate boiling was not fully suppressed even in the high-quality
region in the case of the counter flow, while convective evaporation was
dominant in the high-quality region with uniform heat flux conditions.

Torikoshi and Ebisu (1993) have investigated experimentally the
heat transfer and pressure drop characteristics of R134a and R32 and a
mixture of R134a/R32 (70%/30% by wt) in a smooth horizontal tube. The
experiment was carried out in a smooth copper tube of 8.70mm inner
diameter, 9.52mm outer diameter, and 4000mm long test section. In the
evaporation and condensation test, the mass flux was varied in the range of
40 kg m$^{-2}$ s$^{-1}$ to 280 kg m$^{-2}$ s$^{-1}$. The experimental results showed that the
evaporative heat transfer coefficient of R134a was lower by 15%, and for R32
it was higher by 55% than that of R22. The evaporation and condensation heat
transfer coefficients of R32/R134a were lower than that of R22.

Wattelet et al (1994) have reported the experimental heat transfer
coefficients for R12, R134a and a mixture of R22/R124/R152a
(52%/33%15%) in a single-tube horizontal evaporator 2.43 m long and
7.04 mm inside diameter for both annular and wavy-stratified flow. The
operating parameters were varied as follows: mass flux, 25 to 100 kg m$^{-2}$ s$^{-1}$,
heat flux, 2 to 10 kW m$^{-2}$, quality, 10% to 90%; and saturation temperature,-15°C to 5°C. It was reported that the heat transfer coefficient for R134a was
higher than that of R12 and the heat transfer coefficient for a mixture was
lower than that of R12 at low mass flux condition. A new correlation was
developed using an asymptotic form.

Darabi et al (1995) have reported a comprehensive review of the
flow boiling correlations for smooth and augmented tubes for both pure fluids
and mixtures. Discussions on the correlations for smooth tubes and enhanced
tubes were carried out. The pure refrigerants’ correlations of Chen (1966),
Shah (1982), Bjorge et al (1982), Gungor and Winterton (1986), Liu and
Winterton (1991), Klimenko (1988, 1990), Kandlikar (1990), and Steiner and
Taborek (1992) were reviewed for their fitness to predict the data and deviation between them.

Wang and Chato (1995) have reviewed more than 60 articles related to pool boiling, forced convective evaporation and the falling film evaporation of mixtures. The reasons for the degradation of heat transfer with mixtures, the circumferential variation of wall temperatures, the suppression of nucleate boiling, the improved peak heat flux, methods for enhancement of pool and flow boiling, the heat transfer coefficient and pressure drop correlations of mixtures were discussed.

Wijaya and Spatz (1995) have measured the heat transfer coefficient and pressure drop of refrigerants R22 and R32/R125 (50/50% by wt) without oil in the refrigerant circuit. The condenser and evaporator test sections consisted of smooth, horizontal copper tubes of 9.53mm outer diameter, 7.75mm inner diameter and 3.05m and 3.66m length respectively. The operating parameters were; the average saturated condensing temperatures were 46.1°C and 51.7°C, the saturated evaporating temperature was 4.4°C, the average inlet and exit qualities for the evaporation tests were 20% and 90% respectively and the mass flux was varied from 160 kg m⁻² s⁻¹ to 561 kg m⁻² s⁻¹. The experimental results showed that at similar mass fluxes the evaporation heat transfer coefficients of R32/R125 were 23% to 63% higher than that of R22.

Wang et al (1996) have presented the two phase heat transfer coefficients and pressure drop characteristics of refrigerants R22 and R407C in a smooth tube of 9.52mm nominal diameter. The experiment was conducted in the operating conditions of 600 kN m⁻² and 680 kN m⁻² evaporating pressures, 100 kg m⁻² s⁻¹ and 300 kg m⁻² s⁻¹ mass fluxes and 6 kW m⁻² and 14 kW m⁻² heat fluxes. The effects of the evaporating pressures, mass fluxes and heat fluxes on the heat transfer coefficients were analyzed. The
experimental results were compared with the existing heat transfer coefficient correlations. It was found that the heat transfer coefficients for R407C were lower than that of R22 by 50% to 70% and the pressure drop of R407C was similar to that of R22 at low mass flux (100 kg m$^{-2}$ s$^{-1}$) and 45% lower than that of R22 at a higher mass flux (300 kg m$^{-2}$ s$^{-1}$).

Shin et al (1997) have measured experimentally the convective boiling heat transfer coefficients of pure refrigerants R22, R32, R134a, R290, and R600a and refrigerant mixtures R32/R134a, R290/R600a, and R32/R125. The experiment was conducted at the exit temperature of 12°C ± 0.5°C for all refrigerants, for heat fluxes of 10 to 30 kW m$^{-2}$ and the mass flux range of 424 to 742 kg m$^{-2}$ s$^{-1}$ for R22, R32, R134a, R32/R134a, and R32/R125; 265-583 kg m$^{-2}$ s$^{-1}$ for R290, R600a, and R290/R600a. It was observed that the heat transfer coefficients depended strongly on heat flux in a low quality region, and become independent as the quality increases. The experimental results were compared with the existing heat transfer coefficient correlations and the Gungor and Winterton correlation (1986 and 1987) for pure substances and the Thome-Shakil modification of the Gungor and Winterton (1986) correlation for refrigerant mixtures over-predicted the heat transfer coefficients measured in that study.

Wang and Chiang (1997) have reported the two-phase friction and heat transfer characteristics of R22 and R407C inside a 6.5 mm smooth tube. The heat transfer tests were carried out for a mass flux of 100 and 400 kg m$^{-2}$ s$^{-1}$ at an evaporation temperature of 2°C and the adiabatic frictional pressure drop was recorded in the mass flux range of 100 to 700 kg m$^{-2}$ s$^{-1}$. It was found that the pressure drop for R407C was lower than that of R22 since the development of the flow pattern for R407C falls behind that of R22. It was found that the major heat transfer mechanism at low mass flux was nucleate boiling, and it virtually became convective evaporation as the mass flux
increased to 400 kg m\(^{-2}\)s\(^{-1}\). It was also perceived that the reduction of the heat transfer coefficients for R407C mixtures was especially profound at low mass flux, and the reduction of the heat transfer coefficient decreases with an increase in the mass flux.

Mathur (1998) presented the tube side heat transfer coefficients for single phase flow, and evaporation using propane (R290), isobutane and a 50/50% mixture (by weight) of propane and isobutane. A horizontal tube with a diameter of 9.53mm and 0.64mm wall thickness was used for experimentation. The effects of the refrigerant properties on the heat transfer coefficient were analyzed for hydrocarbons, R12 and R134a. The evaporation temperature of -6.7°C and 4.4°C and condensation temperature of 37.8°C and 48.9°C were maintained over a wide range of mass flux conditions. It was concluded that the average heat transfer coefficients for hydrocarbons were greater by 194% to 238% in comparison to those of R12 and by 157% to 192% in comparison to those of R134a.

Ebisu and Torikoshi (1998) have carried out an experiment and provided the heat transfer coefficient and pressure drop data for R410A, R407C and R22 flowing inside a smooth horizontal copper tube. For the evaporation and condensation tests, the saturated temperatures were maintained at 5°C and 50°C respectively, the mass fluxes were 150 kg m\(^{-2}\)s\(^{-1}\) and 300 kg m\(^{-2}\)s\(^{-1}\) and a constant heat flux of 7.5 kW m\(^{-2}\). It was observed that the heat transfer coefficient of R410A was 20% higher than that of R22 up to qualities of 0.4, and almost the same up to a quality of 0.6. It was also observed that the heat transfer coefficient of R407C was 30% and 50% lower than those of R22 and R410A, respectively.

Wang et al (1998a) have reported the two-phase flow pattern, friction, and heat transfer characteristics of R22 inside a 6.5 mm smooth tube in the mass flux range of 50 to 700 kg m\(^{-2}\)s\(^{-1}\). It was reported that the two-
phase heat transfer coefficient for the stratified-wavy flow pattern was found to be insensitive to the change of vapor quality, and was approximately proportional to $q^{0.6-0.7}$. The LMTD method was used to calculate the heat transfer coefficient of R22 and the effect of mass flux at a fixed heat flux on the heat transfer coefficient was analyzed. The average heat transfer coefficients were compared with the existing researchers’ results.

Wang et al (1998b) have presented the two phase heat transfer coefficient and pressure drop characteristics of refrigerants R22 and R410A in a smooth horizontal double tube, with a counter flow arrangement and 6.54 mm inside diameter. The experimental results were taken at an evaporation temperature of 2°C, mass flux between 100 kg m$^{-2}$ s$^{-1}$ and 400 kg m$^{-2}$ s$^{-1}$ and heat flux between 2.5 kW m$^{-2}$ and 20 kW m$^{-2}$. The effects of heat flux and mass flux on the heat transfer coefficient were analyzed based on the variation in the flow pattern inside the tube in the considered operating conditions. It was found that the heat transfer coefficient of R410A was 10% to 20% higher than that of R22 at the mass flux of 100 kg m$^{-2}$ s$^{-1}$.

Aprea et al (2000) have measured experimentally the mean heat transfer coefficients of R22 and R407C in the coaxial counter flow evaporator of a refrigerating vapour compression system with a 20 mm inner diameter tube. The experiment was conducted with the heat flux ranging from 1.9 to 9.1 kW m$^{-2}$ and the mass flux varying from 30 to 140 kg m$^{-2}$ s$^{-1}$. The LMTD method was used to calculate the heat transfer coefficient of the refrigerant and the obtained results illustrated that the R22 heat transfer coefficient was always greater than that of R407C but the difference decreased from 54 to 24% with an increase in the refrigerant mass flux. The experimental results were compared with the existing heat transfer coefficient correlations. The comparison relationships showed a strong over prediction for the R407C coefficients.
Boissieux et al (2000) have presented the experimental heat transfer results of Isceon59, R407C and R404A in a double tube horizontal heat exchanger with a counter-flow arrangement of 4 m long. The heat transfer results were compared with those of the Kattan model (1998) for refrigerant mixtures and the more conventional correlations such as the Shah (1982) and Gungor-Winterton (1987). The Kattan et al (1998) model was modified to fit the three new refrigerants. It was reported that the modified Kattan model offers a good prediction of the heat transfer results, with a standard deviation of 6.1%.

Chang et al (2000) have investigated experimentally the performance of a heat pump system using HC refrigerants such as propane, isobutene, butane, propylene, propane/isobutane and propane/butane. Ethyl alcohol was used as the secondary heat transfer fluid in both the condenser and evaporator. The condensation and evaporation heat transfer coefficients of the HC refrigerants were measured based on the overall conductance for each subsection of the heat exchangers. An empirical correlation was evolved, based on the experimental results and the predicted heat transfer coefficients were compared with the experimental results.

Choi et al (2000) have measured the evaporative heat transfer coefficients of R32, R134a, R32/134a, R407C (R32/125/134a: 23/25/52 wt %) and R22 in a smooth horizontal tube of 7.75 inner diameter, and heated length of 5.9m. Experiments were conducted at average temperatures of -12.0°C to 17.0°C, with a mass flux of 240 to 1060 kg m⁻² s⁻¹, and a heat flux of 4.1 to 28.6 kW m⁻². The evaporative heat transfer characteristics of all the considered refrigerants were compared with those of R22. The experimental results were compared with the existing heat transfer coefficient. A new correlation based on a superposition model for pure refrigerants and refrigerant mixtures was presented. A simple form of the
correction factor which represents the heat transfer degradation for zeotropic refrigerant mixtures was incorporated for mixtures in the proposed correlation.

Jabardo and Filho (2000) have performed a convective boiling experiment for refrigerants R22, R134a and R404A in a 12.7 mm internal diameter, 2 m long, horizontal copper tube. Vapor fraction, mass velocity and heat flux varied in the following ranges: 5% to 100%, 50 to 500 kg/m²s and 5 to 20 kW m⁻², respectively. The effects of the evaporating temperatures, vapor fraction, mass flux and heat flux on the heat transfer coefficient were investigated. High quality experiments were also performed up to the point of the tube surface dryout. It was observed that R134a presents the highest heat transfer coefficient whereas the R404A presented the lowest in the convective region. The experimental results were compared with the two existing heat transfer coefficient correlations.

Seo and Kim (2000) have reported the heat transfer coefficients and pressure drop during the evaporation of R22 at low evaporating temperatures that correspond to the heat pump operating conditions in the heating mode. The heat transfer measurements were performed for 3.0 m long smooth and micro-fin tubes with outer diameters of 9.52 and 7.0 mm, respectively. The evaporating temperature varied from -15 to 5°C, mass flux from 70 to 211 kg m⁻² s⁻¹ and heat flux from 5 to 15 kW m⁻². The effects of the evaporating temperature, heat flux, tube diameter and mass flux on the heat transfer coefficient in smooth and micro-fin tubes were investigated. The heat transfer and pressure drop results were compared with those of the existing empirical models.

Somchai and Somjin (2000) have investigated experimentally the two phase heat transfer coefficient of R134a evaporating under forced flow conditions in a counterflow smooth horizontal 1.8 m long double tube with a
9.52mm outer diameter. The tests were carried out at evaporating temperatures ranging between 4 and 25°C, the inlet qualities between 0.1 and 0.25, mass fluxes between 160 and 470 kg m\(^{-2}\) s\(^{-1}\) and heat fluxes between 8 and 55 kW m\(^{-2}\). The effects of heat flux, mass flux, evaporation pressure and lubricating oil on convective heat transfer coefficients were discussed. The experimental results were compared with the existing heat transfer coefficients correlation and a new correlation was proposed, based on the experimental results.

Barbosa and Hewitt (2001) have proposed a model for phase change heat transfer to binary mixtures at high qualities (annular flow regime), based on a Colburn-Drew type formulation. The interfacial parameters such as mass fluxes, compositions and temperature were calculated. The predicted wall temperatures and heat transfer coefficients were compared with the existing literature experimental data.

Lallemand et al. (2001) have investigated experimentally the forced convective boiling of the pure refrigerant R22 and zeotropic refrigerant mixture R407C in both a smooth tube and a microfin tube with a uniform heat flux. The refrigerant mass flux was varied from 100 to 300kg m\(^{-2}\)s\(^{-1}\) and heat flux from 10 to 30 kW m\(^{-2}\). It was perceived that the local heat transfer coefficients depend strongly on heat flux at a low quality and on mass flux at a high quality. The R407C heat transfer coefficients of the smooth and microfin tubes were 15 to 35\% respectively, lower than that of R22. It was concluded that the best heat transfer enhancement was obtained at a low heat flux and mass flow rate.

Yu et al. (2002) have presented the flow pattern and heat transfer results of evaporation in a 10.7 mm diameter, 1.5 m long horizontal smooth and a micro-fin tube. The experiments were carried out for mass flux in the range of 163 and 408 kg m\(^{-2}\) s\(^{-1}\), for heat flux between 2200 and 56,000 W m\(^{-2}\).
and at an evaporation temperature of 6°C. For the measurement of wall temperature at each axial location, three thermocouples were fixed circumferentially, and the average wall temperature was considered for the calculation of the heat transfer coefficients. The experimental data were compared with five existing correlations. The heat transfer coefficients for intermittent and annular flows in both the smooth and the micro-fin tubes were shown to agree well with Gungor and Winterton’s (1986) correlation with modified constants.

Julio et al (2003) have presented the experimental results of the nucleate and convective boiling of R407C, flowing with a mass velocity in the range of 200 to 300 kg m\(^{-2}\) s\(^{-1}\) at 770 kPa, inside horizontal plain and microfin tubes with outside diameters of 7.0 and 12.7 mm. The data are presented for heat fluxes of 10 and 20 kW m\(^{-2}\). For a heat flux of 10 kW m\(^{-2}\) and a mass velocity of 200 kg m\(^{-2}\) s\(^{-1}\), the dominant heat transfer mechanisms for the 7 mm OD tube was the nucleate boiling regime whereas for the 12.7 mm OD tube it was the convective boiling regime. It was concluded that the pressure gradient increased as a function of vapor quality.

Lee et al (2006) have investigated experimentally the evaporating and condensing heat transfer characteristics of R290 (propane), R600a (iso-butane), R1270 (propylene) and R22 in a conventional vapor compression type heat pump system. It was found that the local evaporating heat transfer coefficients of hydrocarbon refrigerants were higher than that of R22 at all quality ranges. It was observed that the average evaporating heat transfer coefficient increased with an increase in the mass velocity. The experimental results of all the refrigerants were compared with a few existing correlations and Kandlikar’s (1990) correlation agreed well with the evaporating heat transfer results.
The reviewed literatures reported that experiments were carried out in a tube-tube heat exchanger to predict the heat transfer coefficients of pure and refrigerant mixtures in different ranges of mass flow rate, heat flux, vapor fraction, pressures, tube diameters and tube lengths. The effects of evaporating pressures, mass fluxes and heat fluxes on the heat transfer coefficients were analyzed. It was perceived that the local heat transfer coefficients depend strongly on heat flux at a low vapor fraction and on mass flux at a high vapor fraction. It was also concluded that the transport properties of the liquid affect heat transfer more than the other properties. In a few of the research works, for the measurement of wall temperature at each axial location, four thermocouples were fixed circumferentially and the average heat transfer coefficient was considered for the calculation of the heat transfer coefficients. A few researchers have developed a new general correlation for forced convection boiling and subcooled boiling with the aid of the large existing literature database.

2.8 HEAT TRANSFER COEFFICIENTS OF REFRIGERANT-OIL MIXTURES

Schlager et al (1988) have studied the evaporation and condensation of the R22 refrigerant- Naphthenic base mineral oil (viscosity of 150 SUS) mixture, both in a smooth and micro-fin tube. The test section was a counter-flow, concentric-tube of 3.66m length with water flowing in the annulus. The oil mass concentration was varied from 0% to 5 % and the mass fluxes were tested over the ranges of 125 to 400 kg m\(^{-2}\) s\(^{-1}\). It was observed that for both smooth and micro-fin tubes, small quantities of oil at approximately 1% to 3% were found to enhance the evaporation heat transfer in comparison with pure refrigerant evaporation.

Schlager et al (1989) have extended the previous evaporation and condensation of refrigerant-oil mixtures test to cover mixtures of a naphthenic
base mineral oil (viscosity of 300 SUS) with R22 in a smooth and micro-fin tube. The test section was a straight tube in tube heat exchanger with an outer diameter of 9.52 mm and a length of 3.67 m. The operating conditions were; the saturated temperature in evaporation was 3°C, mass fluxes were varied from 125 kg m\(^{-2}\)s\(^{-1}\) to 400 kg m\(^{-2}\)s\(^{-1}\), the oil concentration was varied from 0% to 5% by weight and the inlet and exit qualities were varied between 15% and 85%. It was observed that the heat transfer coefficient was degraded to a maximum of about 20% for the smooth tube, at the maximum oil concentration of 5%.

Hambraeus (1991) reported a study of evaporation inside two horizontal smooth tubes of 12 mm inner diameter with a length of 4 and 10 m using the R134a refrigerant and with oil-refrigerant mixtures. The heat flux was varied from 2 to 10 kW m\(^{-2}\) and the oil content was varied from 0 to 2.5 mass percentages (synthetic oil, EXP-0275). It was found that the oil-free R134a had a higher heat transfer coefficient than R22 at the same heat and mass fluxes. It was found that at 2 and 4 kW m\(^{-2}\) the heat transfer coefficient had a maximum value for an oil content of around 0.5 mass percentages and no increase was registered for a heat flux of 6 kW m\(^{-2}\). The heat transfer coefficients for the pure refrigerant were also compared with the two existing correlations.

Zurcher et al (1998) have reported in-tube evaporation for R407C and R407C/oil in a plain copper tube. The tests were conducted at a nominal inlet pressure of 6.45 bar at mass velocities of 100, 200 and 300 kg m\(^{-2}\)s\(^{-1}\) over nearly the entire vapor quality range. The nominal oil mass fractions of 0.0, 0.5, 1.0, 3.0 and 5.0% oil were charged to the refrigeration loop. The experimental results showed that for local vapor qualities from 10 to 70%, the oil had little effect on the local R407C/oil heat transfer coefficients at low mass flux. It was also observed that at vapor quality higher than 70%, the
effect of the oil was very dramatic even with small amounts of oil, and the heat transfer coefficients decreased by 80 to 90%.

Kedzierski (2000) has overviewed the boiling heat transfer research of the recent few years at NIST on refrigerant/lubricant mixtures. The main aim of the work was fundamentally to understand the effects of lubricant properties and bubble parameters on heat transfer performance. The magnitude of the effect of lubricant viscosity, miscibility and concentration on the heat transfer was quantified. The NIST research illustrated that a considerable improvement over pure R134a heat transfer could be obtained for R134a/lubricant mixtures with small lubricant mass fraction, high viscosity and a large critical solution temperature.

Haitao et al (2008) have investigated experimentally the flow pattern and heat transfer characteristics of the R410A-oil mixture inside a straight smooth tube of 7.0mm outside diameter. The experimental conditions included an evaporation temperature of 5°C, a mass flux from 200 to 400 kg m\(^{-2}\) s\(^{-1}\), a heat flux from 7.56 to 15.12 kW m\(^{-2}\), and nominal oil concentration of 0% to 5%. It was observed that the heat transfer coefficient of the R410A-oil mixture increased with mass flux and the range of low and intermediate vapor qualities (< 0.6). The experimental results were compared with the existing heat transfer coefficient correlations for refrigerant-oil mixtures. A new correlation was developed based on the flow patterns and local properties of the refrigerant-oil mixture, and it agreed with 90% of the experiment data within the deviation of ±25%.

Wenjian et al (2007) have investigated experimentally the two-phase flow pattern and heat transfer characteristics of the refrigerant R22-mineral oil NM56 mixture flow boiling inside small tubes. The test condition of nominal oil concentration was from 0% to 5%, mass flux was from 200 to 400 kg m\(^{-2}\) s\(^{-1}\), heat flux was from 3.2 to 14 kW m\(^{-2}\), evaporation
temperature 5°C, inlet quality from 0.1 to 0.8, and quality change from 0.1 to 0.2. It was observed that the oil presence could make annular flow to form early and to retard to diminish in quality direction at nominal oil concentration ≥ 3%. It was found that the two-phase heat transfer multiplier with refrigerant-oil mixture properties increases consistently and monotonically with local oil concentration at different vapor qualities.

Many researchers have investigated experimentally the flow pattern and heat transfer characteristics of a refrigerant-oil mixture inside a smooth tube. All the experiments were carried out in a test section under different diameters, length, mass flux, heat flux, saturation pressure and temperature, and different mass proportion of oil. The oil mass concentration was varied from 0% to 5%. It was observed that the effect of oil in the refrigerant was dependent on the heat flux. It was also concluded that the oil-free refrigerants had a higher heat transfer coefficient than oil-refrigerants at the same heat and mass fluxes.

### 2.9 HEAT TRANSFER COEFFICIENT IN THE U-BEND

Ouazia et al (1994) have presented an experimental study of upward and downward flow of boiling heat transfer characteristics for R22 in the U-bend and it downstream. The test sections had a bend radius of 40 mm, 58 mm and 96 mm with an inner tube diameter of 16 mm, and outside diameter of 20 mm. It was found that the average heat transfer coefficients were higher both in the bends and also the downstream of the bends compared with the flow in an equivalent horizontal straight tube. It was reported that the effects of U bend was remained for a length of about 10-15 U bend diameters and after that the flow pattern was restored to its previous condition. The correlations were proposed to predict the average flow of the boiling heat transfer coefficients of the refrigerant flowing in the U-bends.
Cho and Kim (1998) have investigated experimentally the heat transfer coefficients in the U-bends and the straight sections of smooth and microfin tube evaporators using R22 and R407C. The smooth tube had an outer diameter of 9.52mm, and the radius of the curvature of the U-bend was 2.5 times the outer radius of the U-bend. It was observed that one third of the downstream straight section was affected by the upstream U-bend due to the centrifugal force and mixing of the two phase flow. It was found that the average heat transfer coefficients in the U-bends were higher by 4% to 33% than those in the straight sections.

Cho and Tae (2000) performed an evaporation heat transfer experiment using R407C and R22 mixed with polyol ester and mineral oil in the straight and U-bend sections of a microfin tube. The variations of wall temperature along the U-bend were measured by every 45° for the U-bend section of the tube. The variation of pressure drops and enhancement factors with inlet quality and oil concentration were analyzed for the fixed mass flux and heat flux condition. It was observed that the local heat transfer coefficients in the U-bend had the maximum values at 90° and minimum values at 0° and 180°. It was also observed that the local heat transfer coefficient at the outside curvature of the U-bend was larger than that at the inside curvature of the U-bend.

Wen et al (2007) have reported the flow boiling for refrigerant R600a and R290 mixed with the lubricating oil in serpentine small diameter U-tubes. The tests were conducted at the nominal inlet pressure of 186.2 kN m⁻², vapor qualities in the range of 0 to 0.76, mass flux in the range of 100 to 320 kg m⁻² s⁻¹ and inlet oil concentrations from 0 to 5% mass oil. The effects of mass flux and the U-bend on the average heat transfer coefficients were investigated. It was concluded that the heat transfer coefficients of R600a/R290 with/without oil were affected by the numbers of
the U-bend. It was also observed that the pressure drop during evaporation increased with the addition of a lubricant, mass fluxes and the numbers of the U-turn.

Experiments were carried out in a U-bend with different heat flux, mass flux, vapor quality, bend radius, tube diameter and length. The variations of wall temperature along the U-bend were measured by every 45° for the U-bend section of the tube. It was observed that the local heat transfer coefficients in the U-bend had the maximum values at 90° and minimum values at 0° and 180°. It was also observed that the local heat transfer coefficient at the outside curvature of the U-bend was larger than that at the inside curvature of the U-bend. The average heat transfer coefficients were higher both in the bends and also downstream of the bends compared with the flow in an equivalent horizontal straight tube. Based on the reviewed literatures, in the present study wall temperatures of the fin-and-tube evaporator were measured in the upstream and at 0° location of the U-bend.

2.10 FLOW PATTERN MAP

The flow pattern is the physics that explains how the liquid and vapor phases of the boiling fluid arrange themselves in a heated tube based on the operating conditions. In real appliances at a particular indoor and outdoor condition, which kind of flow pattern occurs, is to be found out, since the reviewed literature revealed that the pressure drop in a parallel flow fin-and-tube is lower at a particular mass flux. This phenomenon occurred because the flow pattern was different from that of other flow configurations. The map can be used to make a fair judgment of the prevailing flow pattern in real appliances under all operating conditions. The flow pattern maps for R22 and M20 are developed by using the Kattan-Thome-Favrat flow pattern map method.
In literatures, many methods such as the Taitel and Dukler (1976) and Steiner (1993) are available to predict the flow pattern in an adiabatic section in a horizontal tube. The main drawback with these flow patterns is that the heat flux has not been considered. Heat flux is the main parameter required to hypothesize the transition from one flow pattern to the other. The Kattan et al (1998) flow pattern map incorporated heat flux and presented the first diabatic flow pattern map. The updated version of the map (Thome, 2005) is used for analysis purposes, and is called as the Kattan-Thome-Favrat flow pattern map.

The map consists of mass flux on the y axis and vapor fraction on the x axis. For a known mass flow rate the mass flux can be calculated based on the flow cross sectional area. The different curves on the map depict the transition lines from one flow pattern to another. The model was constructed using the R123, R402A, R134a, R404A and R502 refrigerants’ experimental results. The various flow patterns, such as Intermittent, Annular, Stratified and Stratified wavy generally prevail in the horizontal tube. The transition lines and the required parameters are calculated using the following equations, 2.1 to 2.20 reported by Kattan et al (1998).

The transition boundary curve between annular and intermittent flows to stratified- wavy flow is calculated as follows,

\[
\cdot \quad m_{wavy} = \left\{ \frac{16A^3_{Gd}g \rho_d \rho_p \rho_g}{\chi^2 \pi^2 \left[ 1 - (2h_{Ld} - 1)^2 \right]^{0.5}} \left[ \frac{\pi^2}{25h^2_{Ld} \left( 1 - \chi \right)^{-F_i(q_l)}} \frac{We^{-F_2(q)}}{Fr} + 1 \right] \right\}^{0.5} + 50
\]

(2.1)
The high quality portion of the flow pattern map curve depends on the ratio of the Froude number (Fr) to the Weber number (We), where Fr is the ratio of the inertia to the surface tension forces and We is the ratio of inertia to the gravity forces. The mass velocity threshold for the transition from *annular to mist* is

\[
\cdot m_{\text{mist}} = \left[ \frac{7680 A_{\text{Gd}}^2 \text{grd}_l \rho_L \rho_G \left( \frac{\text{Fr}}{\text{We}} \right)_L}{\chi^2 \pi^3 \xi_{\text{Ph}}} \right]^{1/3} \tag{2.2}
\]

Evaluating the expression above for the minimum mass velocity of the mist flow transition gives the value of \(\chi_{\text{min}}\), which for \(\chi > \chi_{\text{min}}\) is

\[
\cdot m_{\text{mist}} = m_{\text{min}} \tag{2.3}
\]

The transition between *stratified–wavy flow* and *fully stratified* is given by the following expression

\[
\cdot m_{\text{strat}} = \left\{ \frac{(226.3)^2 A_{\text{Ld}} A_{\text{Gd}}^2 \rho_G (\rho_L - \rho_G) \mu_L \text{gr}}{\chi^2 \pi^3 (1 - \chi)} \right\}^{1/3} \tag{2.4}
\]

The transition threshold into *bubbly flow* is calculated by

\[
\cdot m_{\text{mist}} = \left[ \frac{256 A_{\text{Gd}} A_{\text{Ld}}^2 \text{grd}_l^{1.25} \rho_L (\rho_L - \rho_G)}{0.3164 (1 - \chi)^{1.75} \chi^2 \pi^2 P_{\text{ad}}^{0.25} \mu_L} \right]^{1/1.75} \tag{2.5}
\]

The ratio of the Weber to the Froude number is calculated by

\[
\left( \frac{\text{We}}{\text{Fr}} \right)_L = \frac{\text{grd}_l^2 \rho_L}{\sigma} \tag{2.6}
\]
The friction factor is calculated by the equation (2.7).

\[ \bar{\xi}_{Ph} = \left( 1.138 + 2 \log \frac{\pi}{1.5A_{ld}} \right)^{-2} \]  

(2.7)

The non-dimensional empirical exponents \( F_1(q) \) and \( F_2(q) \) in the equation (2.1) include the effect of heat flux on the onset of the dryout in the annular film; the transition of annular flow into annular flow with partial dryout,

\[ F_1(q) = 646.0 \left( \frac{q}{q_{DNB}} \right)^2 + 64.8 \left( \frac{q}{q_{DNB}} \right) \]  

(2.8)

\[ F_2(q) = 18.8 \left( \frac{q}{q_{DNB}} \right) + 1.023 \]  

(2.9)

The correlation for the heat flux of departure from nucleate boiling, \( q_{DNB} \) is used to normalize the local heat flux:

\[ q_{DNB} = 0.131 \rho_G h_L g \left[ g \left( \rho_L - \rho_G \right) \sigma \right]^{0.25} \]  

(2.10)

The vertical boundary between intermittent flow and annular flow is assumed to occur at a fixed value of the martinelli parameter \( X_{it} \), equal to 0.34,

\[ X_{it} = \left( \frac{1 - \chi}{\chi} \right)^{0.875} \left( \frac{\rho_G}{\rho_L} \right)^{0.5} \left( \frac{\mu_L}{\mu_G} \right)^{0.125} = 0.34 \]  

(2.11)
The threshold line of the intermittent-to-annular flow transition at \( \chi_{IA} \)

\[
\chi_{IA} = \left\{ \left[ 0.2914 \left( \frac{\rho_G}{\rho_L} \right)^{-1/1.75} \left( \frac{\mu_L}{\mu_G} \right)^{-1/7} \right] + 1 \right\}^{-1} \tag{2.12}
\]

Figure 2.1 shows the geometric dimensions of the flow, where \( P_L \) is the wetted perimeter of the tube, \( P_G \) the dry perimeter in contact with only vapor, \( h \) the height of the completely stratified liquid layer and \( P_i \) the length of the phase interface. Similarly \( A_L \) and \( A_G \) are the corresponding cross sectional areas. Normalizing with the tube internal diameter \( d_i \) six dimensionless variables are obtained:

\[
\begin{align*}
 h_{Ld} &= \frac{h}{d_i} & P_{Ld} &= \frac{P_L}{d_i} & P_{Gd} &= \frac{P_G}{d_i} & P_{id} &= \frac{P_i}{d_i} \\
 A_{Ld} &= \frac{A_L}{d_i^2} & A_{Gd} &= \frac{A_G}{d_i^2} \tag{2.13}
\end{align*}
\]

![Figure 2.1 Cross-sectional and peripheral fractions in a circular tube](image-url)
For \( h_{Ld} \leq 0.5 \):

\[
P_{Ld} = \frac{\left\{ 8(h_{Ld})^{0.5} - 2\left[ h_{Ld} (1-h_{Ld}) \right]^{0.5} \right\}}{3} \quad P_{Gd} = \pi - P_{Ld} \tag{2.14}
\]

\[
A_{Ld} = \frac{\left\{ 12\left[ h_{Ld} (1-h_{Ld}) \right]^{0.5} + 8(h_{Ld})^{0.5} \right\} h_{Ld}}{15} \quad A_{Gd} = \frac{\pi}{4} - A_{Ld} \tag{2.15}
\]

For \( h_{Ld} > 0.5 \):

\[
P_{Gd} = \frac{\left\{ 8(1-h_{Ld})^{0.5} - 2\left[ h_{Ld} (1-h_{Ld}) \right]^{0.5} \right\}}{3} \quad P_{Ld} = \pi - P_{Gd} \tag{2.16}
\]

\[
A_{Gd} = \frac{\left\{ 12\left[ h_{Ld} (1-h_{Ld}) \right]^{0.5} + 8(1-h_{Ld})^{0.5} \right\} (1-h_{Ld})}{15} \quad A_{Ld} = \frac{\pi}{4} - A_{Gd} \tag{2.17}
\]

For \( 0 \leq h_{Ld} \leq 1 \):

\[
P_{id} = 2\left[ h_{Ld} (1-h_{Ld}) \right]^{0.5}
\]

Since \( h \) is unknown, an iterative method utilizing the following equation is necessary to calculate the reference liquid level \( h_{Ld} \):

\[
X_n^2 = \left[ \left( \frac{P_{Gd} + P_{id}}{\pi} \right)^{1/4} \frac{\pi^2}{64A_{Gd}} \left( \frac{P_{Gd} + P_{id}}{A_{Gd}} + \frac{P_{id}}{A_{Ld}} \right) \right] \left( \frac{\pi}{\pi^2 P_{Ld}} \right)^{1/4} \left( \frac{64A_{Ld}^3}{\pi^2 P_{Ld}} \right)
\tag{2.18}
\]
Once the reference liquid level \( h_{Ld} \) is known, the dimensionless variables are calculated from equations (2.14) to (2.18) and the transition curves for the new flow pattern map are determined with equations (2.1) to (2.12).

The transition between stratified–wavy flow and fully stratified is given by the following expression of the Zuricher et al. modified equation, since the transition curve \( \dot{m}_{\text{strat}} \) was too low and equation (2.4) was corrected empirically by adding \(+20\chi\) as follows.

\[
\dot{m}_{\text{strat}} = \left( \frac{(226.3)^2 A_{Ld} A_{Gd} \rho_G (\rho_L - \rho_G) \mu_L \varepsilon}{\chi^2 \pi^3 (1 - \chi)} \right)^{1/3} + 20\chi \tag{2.19}
\]

The transition boundary curve between annular and intermittent flows to stratified-wavy flow is calculated by a modified equation (2.20).

\[
\dot{m}_{\text{wavy(new) = wavy}} = \dot{m}_{\text{wavy}} - 75e^{-\left(\frac{x^2-0.97}{x(1-x)}\right)^{1/2}} \tag{2.20}
\]

Using equation 2.1 to 2.20, flow pattern map for R22 and M20 are developed for the present experimental heat flux, evaporator inlet pressure and mass velocity values and are discussed in chapter 5.

It was found that in all the conducted experimental works the performance of the R407C system was lower than that of R22. It was also reported that R290 can be used as an alternative for R22, even though its cooling capacity is lower than that of R22. The main problem concerned with R290 is its flammability nature. It was also observed that R407C/HC blend (M20) is one of the alternatives for R22. It is to be noted that the earlier work reported does not consider BIS and ASHRAE test conditions to maintain the
WBT along with DBT, while experimentation. Moreover, it was found that the capillary tube system was relatively more sensitive to refrigerant charge and outdoor load conditions than other expansion devices. The reviewed literatures reported that experiments were carried out in a tube-tube heat exchanger to predict the heat transfer coefficients of pure and refrigerant mixtures in different ranges of mass flow rate, heat flux, vapor fraction, pressures, tube diameters and tube lengths. Many two phase heat transfer coefficient correlations were evolved based on the experimental data.

Based on all the above observation, an attempt has been made to use the M20 refrigerant mixture in a window airconditioner without changing the condenser (as a retrofit measure). Hence, a detailed study viz, capacity assessment study, heat transfer characteristics study and performance over a range charge quantity study under psychrometric test conditions as per BIS and ASHRAE standards are required and are to be carried out in the present study. Based on the experimental data a separate heat transfer coefficient correlation has to be evolved for M20 for the design of fin-and-tube evaporator.

2.11 OBJECTIVES

The study was envisaged with theoretical and experimental components. The objectives of the study are,

i) Simulation of the evaporation process in the window airconditioner, and compare the refrigerating capacity with that of R22.

ii) To experimentally, assess the performance of a 5.25kW window air conditioner retrofitted with M20 without any major modification but for optimizing the capillary and charge
quantity to maximize COP and compare the same with that of R22.

iii) To experimentally determine the heat transfer characteristics of the fin-and-tube evaporator in the window airconditioner retrofitted with the M20 refrigerant and to compare the same with those of R22.

2.12 SCOPE OF THE PRESENT STUDY

Initially, the capacity assessment study is performed experimentally as per BIS and ASHRAE standards. This is followed by a heat transfer study in the fin-and-tube evaporator with the optimum charge condition evolved in the capacity assessment study. Based on the data base, a correlation has been evolved to predict the heat transfer coefficient of M20 in a typical evaporator of a window airconditioner.

Further, to elucidate the mechanism that governs the actual refrigerating capacity of the system the evaporator is simulated and compared with the present experimental results. Further while retrofitting procedures are carried out, there is a scope for variation in the charge quantity. Hence a study on the influence of undercharged and overcharged condition on the performance of the system has been made.

All the above experimental studies are performed in a Psychrometric test facility to comprehensively understand the retrofitting of window airconditioners with the M20 refrigerant mixture.

Since a similar study was made by Jabaraj et al (2006) a few years ago in the same laboratory, the differences between the two studies are enumerated below in Table 2.1.
Table 2.1  Salient differences between the Jabaraj et al study and the present study

<table>
<thead>
<tr>
<th>Features</th>
<th>Work of Jabaraj et al</th>
<th>Present work</th>
</tr>
</thead>
<tbody>
<tr>
<td>Scope</td>
<td>✓ Studying the performance of the system using R22 and various mixture compositions of HFC407C /HC blend and arrived at M20 with modification in the condenser length.</td>
<td>✓ Capacity assessment study using R22 and M20 without any modification in the condenser.</td>
</tr>
<tr>
<td></td>
<td>✓ Simulation of vapor compression system using R22 and M20 and compare the results with experimental data.</td>
<td>✓ Heat Transfer characteristic study in a fin-and-tube evaporator operating with M20 and evolve a heat transfer coefficient correlation.</td>
</tr>
<tr>
<td></td>
<td>✓ Capacity assessment study using R22 and M20 without any modification in the condenser.</td>
<td>✓ Modeling of fin-and-tube condenser and evaporator with M20 to predict the possible refrigerating capacity.</td>
</tr>
<tr>
<td></td>
<td>✓ Assessing the performance deviation with respect to charge variation in the airconditioner through undercharging / overcharging study using R22 and M20.</td>
<td>✓ Assessing the performance deviation with respect to charge variation in the airconditioner through undercharging / overcharging study using R22 and M20.</td>
</tr>
<tr>
<td>Test facility</td>
<td>Room calorimeter for the indoor side only and with suitable duct arrangement to control condenser inlet air temperature.</td>
<td>Psychrometric test facility with controlled environment chamber for both indoor and outdoor sides that maintain DBT and WBT.</td>
</tr>
<tr>
<td>Test conditions</td>
<td>✓ Evaporator inlet air temperature varied from 21°C to 29°C in steps of 2°C.</td>
<td>Indoor and outdoor air temperatures (both DBT and WBT) were varied in accordance with BIS-1391-1992 and ASHRAE-41.2.1987.</td>
</tr>
<tr>
<td></td>
<td>✓ Condenser inlet air temperature varied from 30°C to 45°C in steps of 5°C.</td>
<td></td>
</tr>
<tr>
<td>Test appliance</td>
<td>0.3 TR window airconditioner</td>
<td>A typical 1.5 TR window airconditioner</td>
</tr>
<tr>
<td>Test procedure</td>
<td>Refrigerating capacity was calculated by the heater load that balances the refrigerating capacity of the unit and maintains the calorimeter room temperature at steady state.</td>
<td>Air enthalpy method was used for the calculation of refrigerating capacity and it was confirmed with the theoretical refrigerating capacity of refrigerant by the measured mass flow rate and evaporator inlet/outlet temperature and pressure. This is performed as per BIS-1391-1992.</td>
</tr>
</tbody>
</table>
Thus the earlier work of Jabaraj et al was to identify a suitable R407C/HC blend mixture, to replace R22 but not as a retrofit measure. The present work expounds the case of retrofitting an R22 window airconditioner with M20 without any change in the system other than changing the capillary.