International Journal of Advanced Mechanical Engineering.
ISSN 2250-3234 Volume 4, Number 1 (2014), pp. 33-54

http://www.ripublication.com/ijame.htm

Modelling of Horizontal Shell and Tube Dry Expansion Refrigerant Evaporator

J.K. Dabas[1], Sudhir Kumar[2], A.K. Dodeja[3] and K.S. Kasana[4]

[1,3]Dairy Engineering Division, National Dairy Research Institute, Karnal-132001, Haryana, India.
[2,4]Department of Mechanical Engineering, National Institute of Technology, Kurukshetra-136119, Haryana, India.

Abstract

A computer program in MS Excel was developed for the performance evaluation of a shell and tube type refrigerant evaporator where refrigerant boils in horizontal smooth tubes and the refrigerated water flows outside the tubes, as the analytical solution for the complicated two phase heat transfer throughout the evaporator is not possible due to varying flow patterns with varying value of vapour quality. Both the fluids were having single pass through the evaporator. The algorithm was based on a numerical method in which the evaporator tube was divided into a grid of large number of small but finite sub-volumes and the energy balance was made for each sub-volume with the available boundary conditions to consider the local parameters based on local values of vapour quality. The appropriate empirical correlations for heat transfer coefficients available in the literature were used in the calculation. Simultaneously an experimental study of the same type of evaporator was also carried out for validation of the program and the predicted and experimental results of performance of evaporator were found in good agreement. The effect of various design parameters on the performance of evaporator was studied in detail by running the computer program. The optimum value of mass velocity can be easily found with the help of this program which decides the optimum value of refrigerant tube diameter. In the second step the length of tube can also be calculated for the required degree of superheat at the evaporator outlet. The program offers an easy and fast solution to the design and analysis problem of such kind of refrigerant evaporator.

Keywords: Evaporator, heat transfer coefficient; computer program; performance prediction.

1. Nomenclature

A	**Surface area of the evaporator tube [m2]**	**Greek symbols**	
C	Specific heat [J kg-1K-1]	α	Heat transfer coefficient [W m-2K-1]
d	Diameter of tube [m]	ρ	Density [kg m-3]
Db	Bubble departure diameter [m]	µ	Coefficient of dynamic viscosity [Pa s]
e	Iteration indicator [integer]	Δ	Difference, division of
f	Friction factor	σ	Surface tension [N m-1]
F	Two phase convection multiplier factor	Φl2	Two-phase frictional multiplier
G	Mass velocity [kg m-2 s-1]		
h	Specific enthalpy [J kg-1]	Subscripts	
j	Serial number of a coil [integer]	ac	acceleration
k	Thermal conductivity [W m-1K-1]	cv	convection
l	Sub-length [m]	e	Evaporator tube
L	Total length of evaporator tube [m]	Eq	Equivalent
La	Laplace constant (characteristic length) [m]	f	Frictional
m	Mass flow rate [kg s-1]	i	Inlet, inner
n	Number of pairs of horizontal tube in the bundle [integer]	l	Saturated liquid refrigerant
N	Number of sub-volumes	nb	Nucleate boiling
P	Pressure [Pa]	o	Outlet, outer
dP	Pressure drop [Pa]	p	Constant pressure
Pr	Prandtl Number	pb	Pool boiling
q	Heat flux [W/ m2]	r	Refrigerant
Q	Total heat transfer rate [W]	s	Saturated
Re	Reynolds Number	sup	Superheated
S	Suppression factor	tp	Two phase
t	Temperature [o C]	v	Saturated vapor refrigerant
T	Absolute temperature [K]	w	water
U	Overall heat transfer coefficient [W m-2K-1]	lv	Liquid-vapour
v	Specific volume [m3kg-1]	lo	Liquid only
x	Vapor quality		
χtt	Martinelli parameter		

2. Introduction

The overall performance of a refrigeration system has a direct and significant dependence on the effectiveness of two phase heat transfer in both of the evaporator and condenser. This effectiveness depends further on the type of refrigerant used, inlet parameters of both the heat exchanging fluids and the typical geometry of heat exchangers. The single phase heat transfer on the non-refrigerant side is easier to predict and manage but the two phase heat transfer on the refrigerant side is more complicated to predict accurately in the design of a new evaporator. A large number of studies of boiling heat transfer are available in the literature and the most of these are for boiling inside the tubes. This is because in most of the evaporator designs, the refrigerant boils and flows through tubes which may be horizontal or curved. Many aspects of boiling heat transfer inside horizontal smooth tubes have been revealed by researchers and a lot of correlations for local heat transfer coefficients based on their experimental data have been suggested. Three different kinds of models for prediction of heat transfer coefficient are available in the literature. In one model, initially at low vapour quality both the nucleate boiling and the convective evaporation coexist and remain independent of vapour quality but as the quality increases, nucleate boiling is suppressed and the direct dependence of convective evaporation on the vapour quality is observed. Here the ratio of two phase heat transfer coefficient to the heat transfer coefficient in liquid only flow is expressed as a function of Martinelli number and Boiling number. The nucleate boiling suppression point is based on the Boiling number, a dimensionless representation of mass flux and heat flux. Jung et al. (1989), Hambraeus (1999), Jung and Radermacher (1993), Hartnett and Minkowycz (2000) suggested these kinds of correlations for refrigerants based on their experimental data. In another model the nucleate boiling dominant region and forced convection dominant region have been expressed separately to predict the local heat transfer coefficient. Lavin and Young (1965), Dembi et al. (1978) and Dhar et al. (1979) worked on this model. The correlations based on these models were purely empirical. In actual practice, the nucleate boiling and forced convection heat transfer coexist all through the boiling tube and both depend separately on the type of two phase flow i.e. annular or stratified flow. Based on this, in the last and most practical and accepted model, the heat transfer coefficient at a point has been expressed as the sum of forced convection coefficient and nucleate boiling coefficient. Takamatsu et al. (1989) suggested the correlations based on this realistic physical model and validated these with their experimental work. The two phase convection coefficient was correlated with liquid only heat transfer coefficient with the help of Martinelli number. The nucleate boiling heat transfer coefficient was related to simple pool boiling coefficient with the help of nucleate boiling heat flux fraction factor and a suppression factor based on effective wall superheat. The suppression factor was depending on the convection coefficient and the heat flux fraction factor was depending on the pool boiling coefficient and also on the suppression factor. In this physical model also some coefficients were determined empirically. Yu et al. (1998) provided an experimental study of surface effect on flow boiling heat transfer in horizontal tubes and reported higher heat transfer coefficient on relatively rough surface. The surface effect was expressed by multiplying the pool boiling coefficient by a correction factor based on the surface roughness measured with SEM and the reduced pressure. In this way a wealth of knowledge is available about every aspect of boiling heat transfer of the commonly used refrigerants like HFC134a, R-22 etc in different types of geometries of the evaporator.

2.1 Need of Computer Simulation

- The various mathematical models and correlations of two phase heat transfer and pressure drop, applicable in different specific conditions of flow and heat transfer can predict only the local or average heat transfer coefficient based on the corresponding local or average values of vapor quality and other variable parameters along the length of evaporator tube. In the two phase flow with heat transfer like in direct expansion evaporator, the vapour quality varies continuously along the length of tube. With this the flow patterns and so the flow and heat transfer characteristics also varies. For accurate prediction of the overall heat transfer performance, the calculation of heat transfer and pressure drop should be done for each local point considering the local conditions and then the output should be integrated for the complete area of the heat exchanger. It becomes a very large calculation which can to be handled only with the help of a computer program.
- In a new design of evaporator, only the inlet conditions of flow, temperature, pressure, and vapour quality etc. of the refrigerant and external fluid are known and a designer is interested in knowing the overall heat transfer performance and output parameters of both the fluids. Without knowing the output parameters, the prediction of performance is not possible analytically and can be predicted only through numerical techniques, which go on calculating for each small sub-volume from the starting to the end of the heat exchanger many times. This kind of techniques can only be used in computer simulation programs.

2.2 Purpose of Present Work

The purpose of the present work was to develop a computer program for prediction of performance and simulation of a shell and tube type evaporator of HFC134a where refrigerant flows through horizontal tube bundle and water flows through the shell, with the help of existing mathematical models and also to validate it by comparing the predicted results with the experimentally found results. For this, an experimental study of the same type of evaporator was also planned. The purpose was also to analyze the effect of varying operating parameters on the overall performance of the evaporator with the help of computer program and to determine the conditions of maximum performance.

3. Materials and Methods

3.1 Experimentation

Figure 1 shows the outline of an experimental model of a shell and tube type evaporator fitted with a simple vapor compression refrigeration system charged with HFC-134a refrigerant. The refrigerant flows through the horizontal tube bundle made of copper from bottom to top in multiple pass and water to be cooled flows through the insulated shell of the evaporator from top to bottom in a single pass. To design this experimental set-up, initially the compressor of Kirloskar-Copeland make and model no. KCE444HAG (1/3 HP), used generally in a water cooler of capacity 40 liters hr^{-1} was selected. The performance characteristics and the specified rating conditions of the compressor as provided by the manufacturer are summed up in table 1 and table 2. The evaporator and condenser designs were selected in the form of insulated rectangular shell and horizontal tube bundle type heat exchangers in which refrigerant flows through tube. Cooling water could flow through the insulated shell of both of the condenser and evaporator. The length of refrigerant tubes (Φ=0.009 m) for both of the condenser and evaporator were calculated on the basis of their heat duties at commercial back pressure. (Refer to table 1) The evaporation and condensation heat transfer coefficients on the refrigerant side were calculated by the correlations proposed by Yu et al. (1999) and Cavallini et al. (2003) for boiling and condensation heat transfer respectively. The water side heat transfer coefficient was calculated by the appropriate correlation chosen from as summed up by Incropera and DeWitt (1996). After designing both the heat exchangers, the fourth component i.e. expansion valve was decided to be a helically coiled double capillary tube of diameter 1.1176 mm. The coil diameter was arbitrarily selected as 50 mm based on its suitability of fixing on the experimental set-up and the pitch was same as the diameter of capillary tube. The length of capillary was found as 1075 mm by using the dimensionless correlation suggested by Mittal et al. (2010).

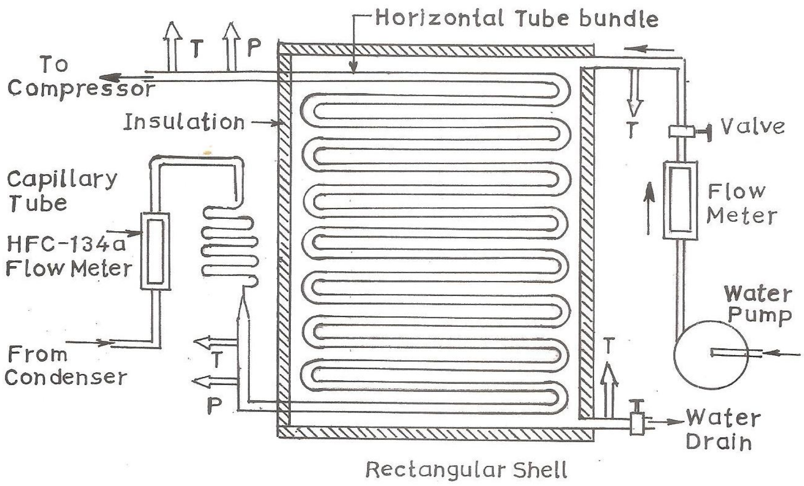


Fig. 1: Experimental set-up of a shell and tube evaporator.

The last design parameter i.e. charge quantity was initially calculated from the simple correlation based only on the internal volume of evaporator and condenser as suggested by Dmitriyev and Pisarenko (1984) and found to be 724 g. In order to vary the conditions of refrigerant at the inlet of evaporator during the experimentation, three different lengths of capillary tube as 750 mm, 1050mm & 1500 mm and three different charge quantities as 500g, 700g and 1000g were chosen to conduct nine different trials of steady state working of the system.

Table 1: Performance rating of compressor.

Horse Power	**Displacement (cc/ rev)**	**Revolutions per minute**	**Cooling Capacity at HBP (W)**	**Cooling Capacity at CBP (W)**	**Input Power at HBP (W)**
0.33	12.05	2900	1253	649	550

Table 2: Rating conditions of compressor.

Ambient Temperature (°C)	**Evaporating Temperature (°C)**	**Condensing Temperature (°C)**	**Sub-cooled Liquid Temperature (°C)**	**Suction Gas Temperature (°C)**	**Suction Pressure (bar)**	**Discharge Pressure (bar)**
High Back Pressure (HBP)						
35.00	7.20	54.40	46.10	35.00	2.75	13.50
Commercial Back Pressure (CBP						
35.00	-6.70	54.40	46.10	35.00	1.25	13.50

The temperatures of refrigerant and the temperatures of water at the inlet and outlet of evaporator were measured with RTDs (Pt 100 Ω at 0^{o}C) strongly insulated along length of tubes by means of polyurethane cellular foam with a maximum uncertainty of ±0.2^{o}C. (Axial heat conduction was hence neglected). The pressures of refrigerant at the same points were also recorded with the help of pressure transducers, with a maximum uncertainty of ± 0.05 bar. Mass flow rate of refrigerant liquid was indicated by a glass tube type rota-meter, fitted in the liquid refrigerant line before capillary tube, with a maximum uncertainty of ± 0.0001kg s^{-1}. Mass flow rate of water to the evaporator shell was also measured by a glass tube type rotameter with a maximum uncertainty of ±0.005 kgs^{-1}.

3.2 Data Reduction

The data of pressure, temperature and flow rate etc. were uploaded and reduced to other performance parameters in Excel worksheets. The thermodynamic properties of HFC-134a like saturation temperature, specific volume and specific enthalpy of saturated liquid and saturated vapor were calculated at the evaporator pressure by using computer subroutines for calculating refrigerant properties as suggested by Cleland

(1986, 1992). The experimental values of performance parameters of evaporator were calculated by the following equations (1) to (3).

3.2.1 Heat transfer rate in the evaporator

$$Q = m_w * C_w * (t_{w,i} - t_{w,o}) \tag{1}$$

3.2.2 Overall heat transfer coefficient over evaporator

$$U = \frac{Q}{A_o * LMTD} \tag{2}$$

$$Where, LMTD = \frac{t_{w,i} - t_{w,o}}{\ln\left(\frac{t_{w,i} - t_s}{t_{w,o} - t_s}\right)} \tag{3}$$

3.3 Uncertainty Analysis

An uncertainty analysis was performed for the above calculated performance parameters using the method described by Moffat, R.J. (1988). The uncertainty in the values of heat transfer rate was calculated in the range of 6.7% to 20% and the uncertainty in the values of heat transfer coefficient was calculated in the range of 9.5% to 26%.

3.4 Correlations used in the Computer Program: (Yu et al., 1998)

3.4.1 Two phase heat transfer coefficient of refrigerant inside the horizontal tube

$$X_{tt} = \left(\frac{1-x}{x}\right)^{0.9} \left(\frac{\rho_v}{\rho_l}\right)^{0.5} \left(\frac{\mu_l}{\mu_v}\right)^{0.1} \tag{4}$$

$$F = 1 + 2 \times X_{tt}^{-0.88} \tag{5}$$

$$\mathrm{Re}_{lo} = \frac{G \times (1-x) \times d_i}{\mu_l} \tag{6}$$

$$\mathrm{Re}_{tp} = F^{1.25} \times \mathrm{Re}_{lo} \tag{7}$$

$$\alpha_{cv} = 0.023 \times \mathrm{Re}_{tp}^{0.8} \times \mathrm{Pr}_l^{0.4} \times \left(\frac{k_l}{d_i}\right) \tag{8}$$

$$La = \sqrt{\frac{2\sigma}{9.81 \times (\rho_l - \rho_v)}} \tag{9}$$

$$D_b = 0.51 \times La \tag{10}$$

$$\alpha_{pb} = 1.25 \times 207 \times \frac{k_l}{D_b} \left(\frac{q_i \times D_b}{k_l \times T_{re}}\right)^{0.745} \left(\frac{\rho_v}{\rho_l}\right)^{0.581} \times \mathrm{Pr}_l^{0.533} \tag{11}$$

$$\xi = 5 \times 10^{-5} \times \left(\frac{\rho_l \times C_{pl} \times T_{re}}{\rho_v \times h_{lv}}\right)^{1.25} \times \frac{\alpha_{cv}}{k_l} \times La \tag{12}$$

$$S = \frac{1}{\xi}\left(1 - \frac{1}{e^{\xi}}\right) \tag{13}$$

$$\eta = \frac{\alpha_{cv}}{S \times \alpha_{pb}} \tag{14}$$

$$K = \frac{1}{1 + 0.875\eta + 0.518\eta^2 - 0.159\eta^3 + 0.7907\eta^4} \quad 15)$$

$$\alpha_{nb} = K \times S \times \alpha_{pb} \tag{16}$$

$$\alpha_{tp} = \alpha_{cv} + \alpha_{nb} \tag{17}$$

3.4.2 Single phase heat transfer coefficient of refrigerant vapor inside the horizontal tube:

$$\alpha_{sup} = 0.023 * Re_v^{\ 0.8} * Pr_v^{0.4} * \left(\frac{k_v}{d_i}\right) \tag{18}$$

$$Where,\ Re_v = \frac{G * d_i}{\mu_v} \quad and \quad Pr_v = \frac{\mu_v * C_{pv}}{k_v}$$

3.4.3 Single phase heat transfer of water flowing across the horizontal tube bundle on shell side:

$$\alpha_w = 0.8 * Re_w^{\ 0.4} * Pr_w^{\ 0.36} * \left(\frac{k_w}{d_o}\right) \tag{19}$$

3.4.4 Overall heat transfer coefficient in the evaporator:

$$U = \left[\frac{1}{\alpha_w} + \frac{d_o . \log_e \frac{d_o}{d_i}}{2 \times k_e} + \frac{1}{\alpha_r}\left(\frac{d_o}{d_i}\right)\right]^{-1} \tag{20}$$

3.4.5 Pressure drop of refrigerant in the horizontal tube:

Frictional pressure drop

$$dP_f = \frac{f \times v \times \Delta l \times G^2}{2 \times d_i} \tag{21}$$

Acceleration pressure drop:

$$dP_{ac} = \frac{G^2 ({v^2}_{tp,o} - v_{tp,i}^2)}{2 v_{tp,i}} \tag{22}$$

Total pressure drop:

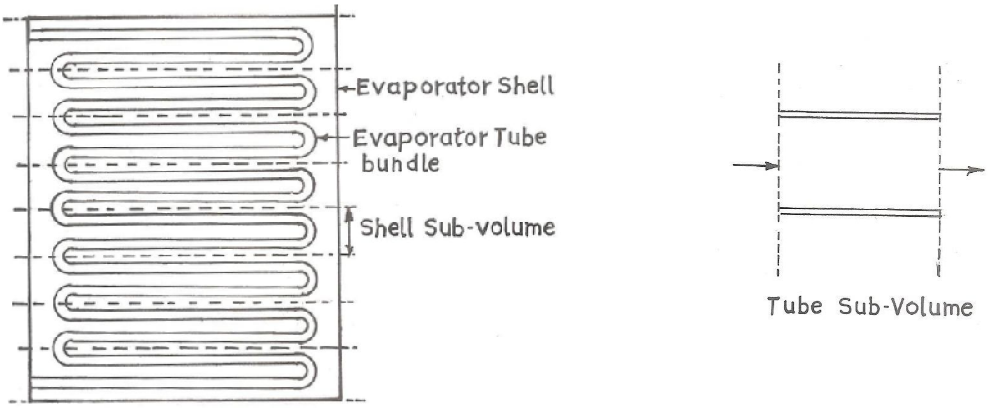
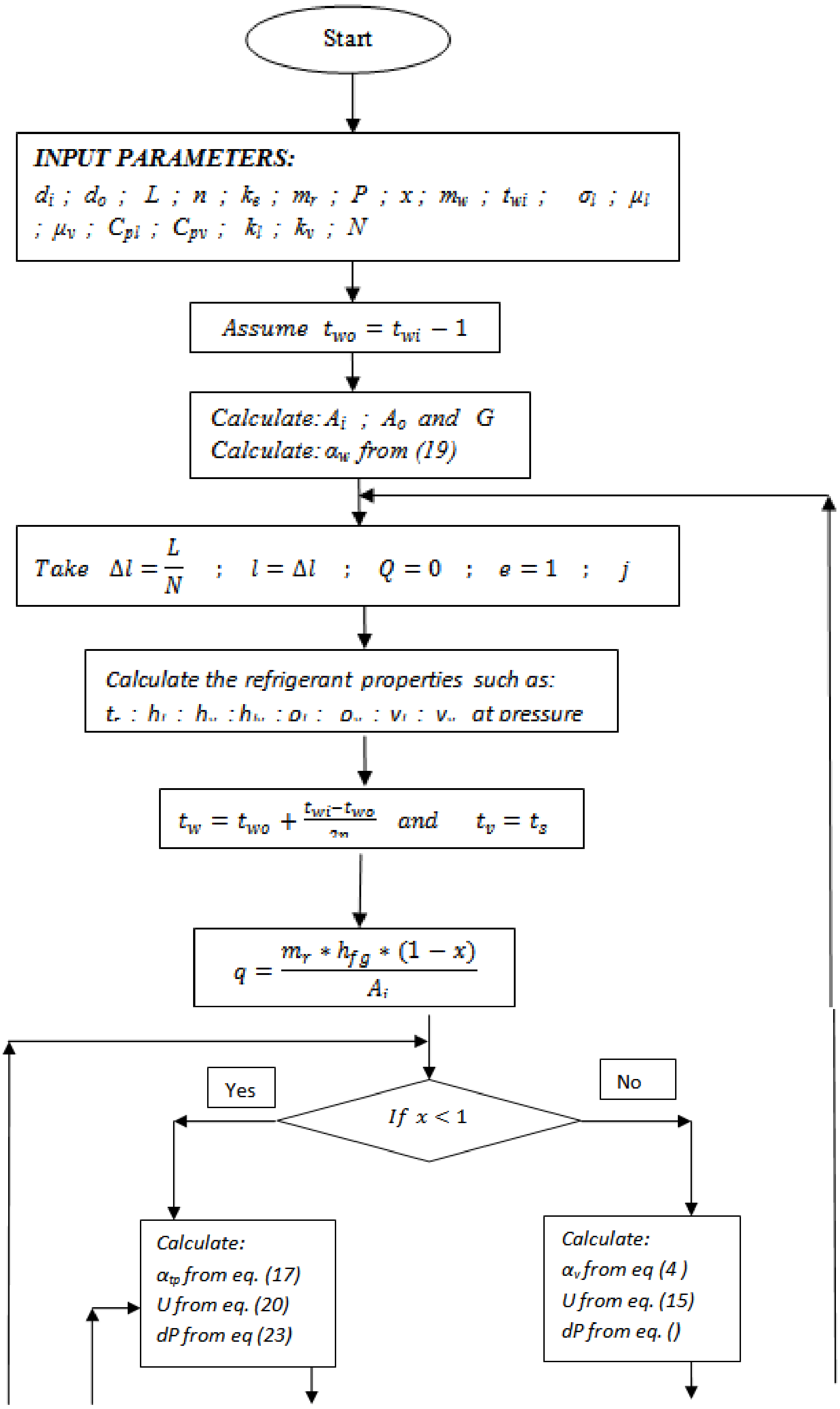


Fig. 2: Division of evaporator shell and tube in the sub-volumes.

(23)

3.5 Computer Program Algorithm

As shown in figure 2a, the cylindrical shell of evaporator was considered to be divided across its height into a number of sub-volumes each covering one pair of horizontal tube of the bundle. The temperature of water flowing over the coil in the shell was assumed to have same value everywhere in one sub-volume but different value from one sub-volume to the other. In this way the temperature of water dropped step by step from one sub-volume to the other and the drop in one sub-volume was calculated by dividing the total temperature drop of water in the shell with the number of tube pairs in the bundle. On the refrigerant side, the total length of the refrigerant tube of the coil was considered to be divided further into a large number of smaller sub-volumes as shown in figure 2b. The properties and other conditions of refrigerant in one such sub-volume were assumed as constant for calculation of heat transfer and pressure drop but changing in the next adjoining sub-volume as a result of heat transfer and pressure drop in the previous sub-volume. The computer program was doing calculations for each of the small sub-volume and summing up the total heat transfer rate in the end. The average value of the overall heat transfer coefficient and refrigerant side heat transfer coefficient could then be calculated. The inlet conditions to the evaporator were applied to the first sub-volume. The conditions in the next sub-volumes were calculated on the basis of heat transfer and pressure drop occurred in their previous sub-volumes. At the end, the conditions calculated for the last sub-volume were taken as outlet conditions. The flow chart of the computer program written in MS Excel is shown in fig.3


Start
INPUT PARAMETERS:
d_i ; d_o ; L ; n ; k_s ; m_r ; P ; x ; m_w ; t_{wi} ; σ_l ; μ_l ; μ_v ; C_{pl} ; C_{pv} ; k_l ; k_v ; N
Assume $t_{wo} = t_{wi} - 1$
Calculate: A_i ; A_o and G
Calculate: α_w from (19)
Take $\Delta l = \frac{L}{N}$; $l = \Delta l$; $Q = 0$; $e = 1$; j
Calculate the refrigerant properties such as:
at pressure
$t_w = t_{wo} +$ and $t_v = t_s$
$q = \frac{m_r * h_{fg} * (1 - x)}{A_i}$
Yes
No
If $x < 1$
Calculate:
α_{tp} from eq. (17)
U from eq. (20)
dP from eq (23)
Calculate:
α_v from eq (4)
U from eq. (15)
dP from eq. ()

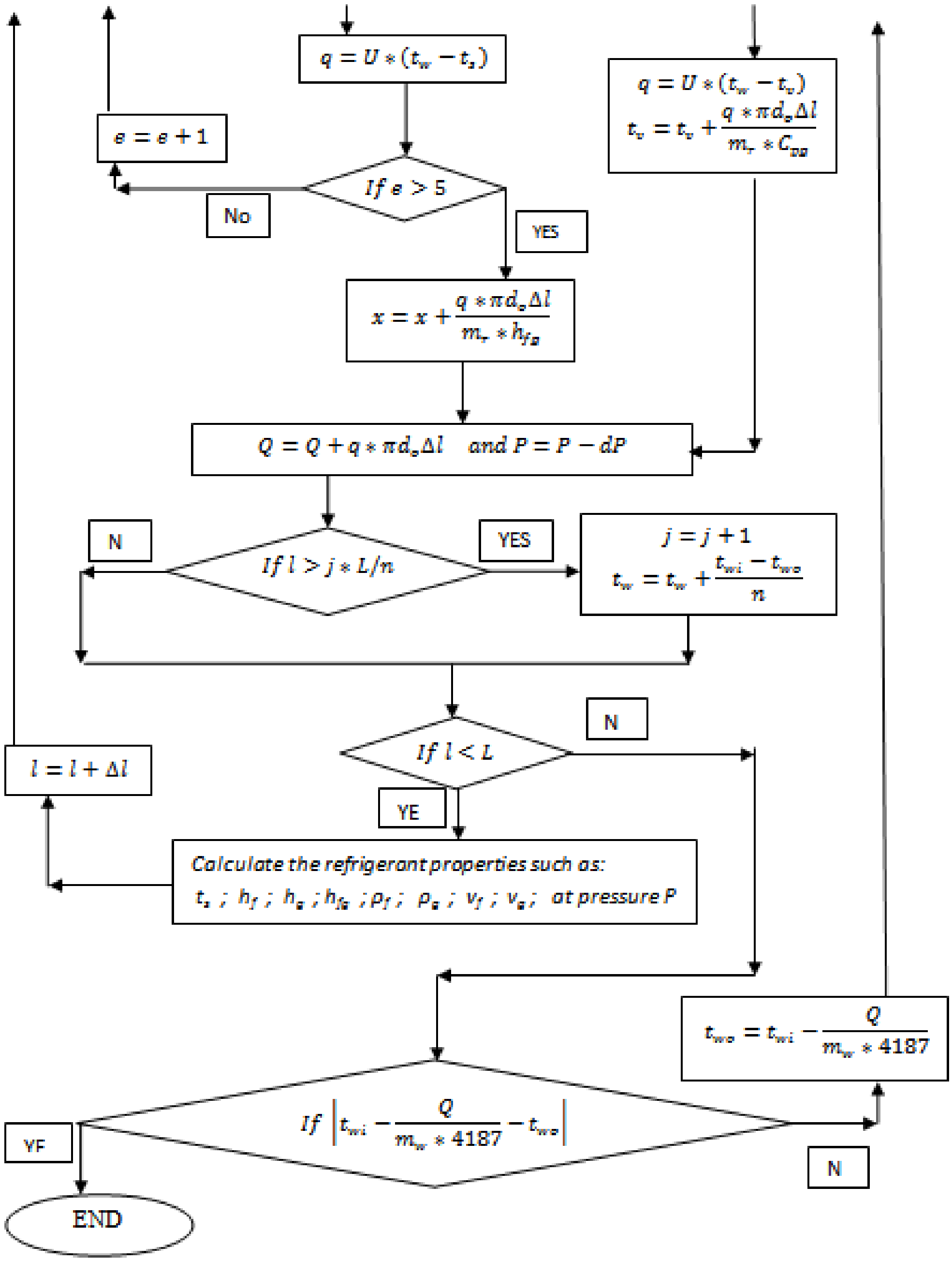


Fig. 3: Algorithm of computer program.

4. Results and Discussion

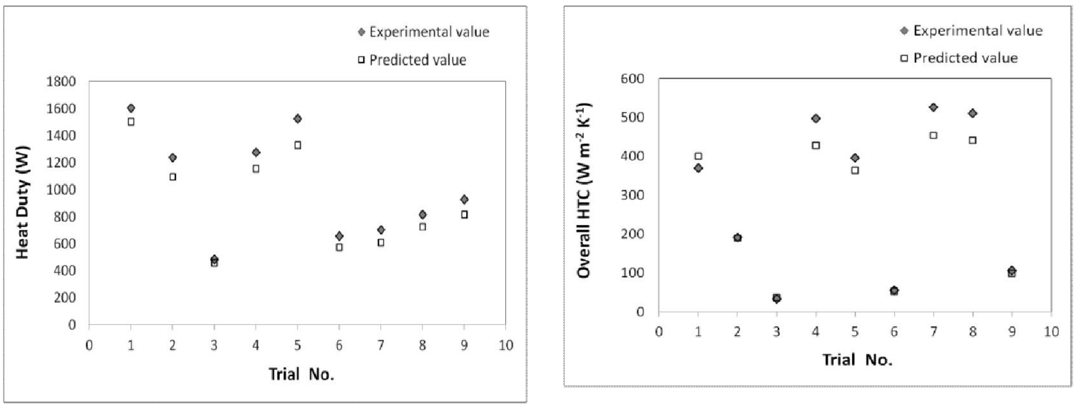
4.1 Comparison of predicted and experimental results:

The input parameters of refrigerant and water to the evaporator were varied in a wide range by conducting trials with three different lengths of capillary tubes and three different quantities of refrigerant charge in the refrigeration system. The values of different sets of input parameters in all the trials have been summed up in table 3.

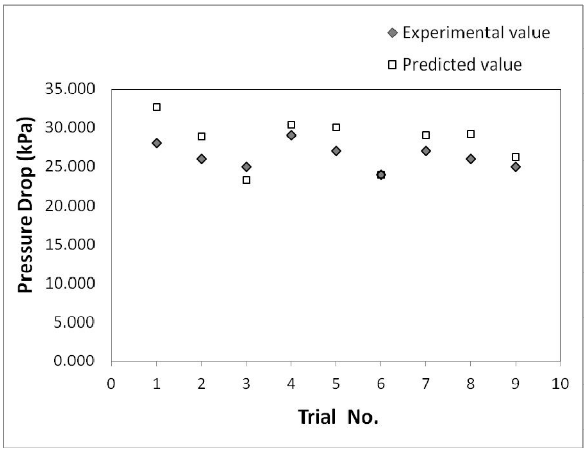
Table 3: Input parameters in the experimental trials on shell and tube evaporator.

Trial No.	Charge quantity	Capillary tube length	Cooling water parameters		Refrigerant Parameters			
			mw	twi	G	Pi	tsi	x
	(g)	(m)	(kg s-1)	(oC)	(kg m-2s-1)	(kPa)	(oC)	
1	500	0.75	0.0910	28.2	176.3	493.965	15.3	0.222
2	500	1.05	0.0950	28.0	118.1	414.655	10.0	0.219
3	500	1.50	0.0920	28.0	44.1	211.207	-8.7	0.216
4	700	0.75	0.0970	29.5	183.4	559.483	19.3	0.234
5	700	1.05	0.0960	28.5	154.3	504.310	16.0	0.224
6	700	1.50	0.0950	28.3	54.7	252.586	-4.0	0.212
7	1000	0.75	0.0987	28.5	211.6	607.759	22.0	0.222
8	1000	1.05	0.0963	28.6	204.5	593.966	21.2	0.217
9	1000	1.50	0.0980	29.6	84.6	345.690	4.7	0.224

These actual input parameters were also fed to the computer program and its predicted output values of heat transfer and pressure drop were compared with the real experimental values. The comparison is shown in fig.4. The predicted values of heat transfer rate were 5% to 13 % less than the experimental values. The predicted overall heat transfer coefficient varied between 7% less to 15% higher than the experimental values. The predicted pressure drop was within 2% of the actual pressure drop. On the basis of this comparison the computer program was validated for predicting the performance of evaporator and thus may be used for the purpose of design and performance analysis of this kind of evaporator.



4 (a) **4(b)**



4(c)

Fig. 4: Comparison of predicted and experimental results.

4.2 Performance Analysis with the help of Computer Program

The analysis of performance of the evaporator was carried out by first establishing the reference values of five different inputs i.e. mass velocity, temperature and vapor quality of the refrigerant and the flow rate and temperature of water entering the evaporator. After that one parameter was varied at a time in a given range while keeping the other four at their reference values. The effects of varying of an individual parameter on the performance of evaporator as predicted by the computer program have been discussed below:

4.2.1 Effect of inlet vapor quality

Fig.5a shows that the total heat transfer rate of evaporator decreases linearly with the increase of vapor quality at the inlet of the evaporator. This is naturally because of lesser liquid refrigerant available for evaporative heat transfer in the evaporator with higher value of vapor quality. Figure 5b shows the same trend for overall heat transfer coefficient with the increase in inlet vapor quality. Figure 5c shows that the pressure drop rises almost linearly with the increasing value of vapor quality. This is because with the lower inlet vapor quality, dry out point is not reached before the exit and the average pressure drop is low due to lower mean velocity throughout the evaporator. Contrary to this at higher inlet vapor quality, dry out point is reached early before the exit and the pressure drop is higher in superheating zone because of higher velocity of vapors.

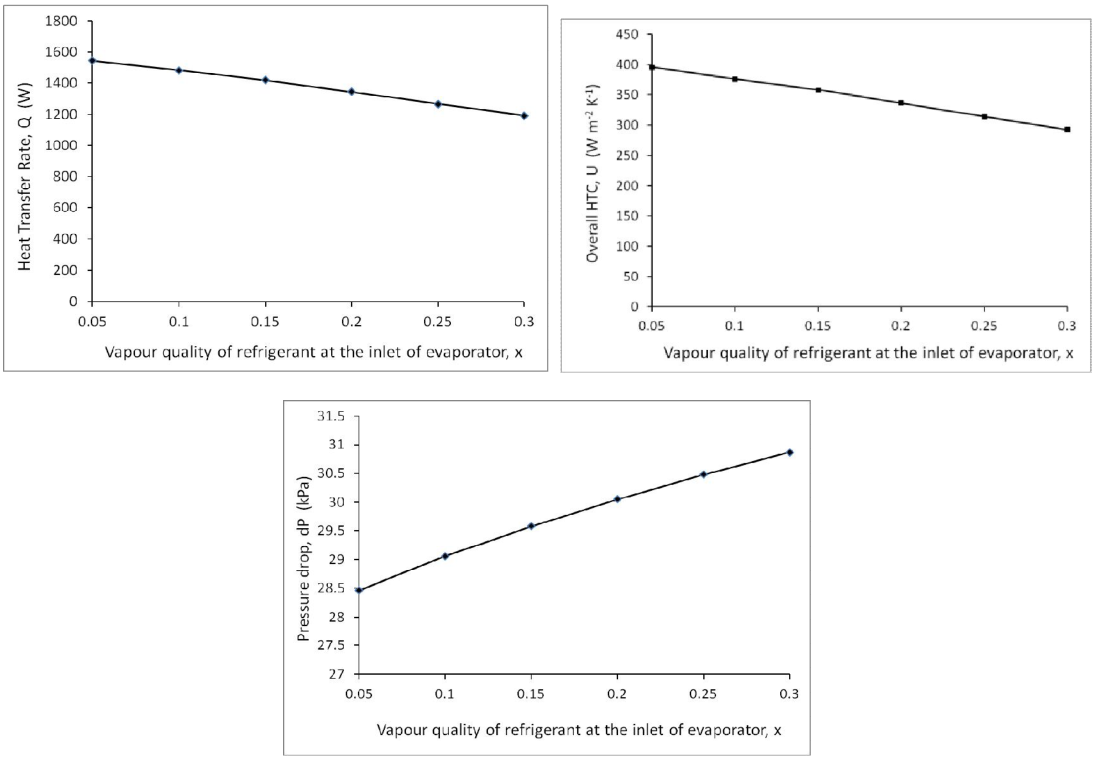


Fig. 5: Effect of inlet vapor quality on the performance of evaporator.

4.2.2 Effect of refrigerant mass velocity

Figure 6a shows that the heat transfer rate generally increases with the increase in refrigerant mass velocity but beyond a limit, there is no further rise in it. This is because with the increase in refrigerant mass velocity, the heat transfer coefficient on the refrigerant side increases thus improving the overall heat transfer coefficient. It primarily increases due to engagement of more and more portion of evaporator in the two phase heat transfer. Also it affects the overall coefficients when it is in the range of waterside heat transfer coefficient. Once the heat transfer resistance on the refrigerant side is made too less as compared to total resistance, further improvement in heat transfer is negligible as shown in fig. 6b unless there is no change in water side heat transfer coefficient. A further rise in mass velocity causes reduction in vapor quality at the exit of the evaporator tube which is undesirable in a refrigeration system. Thus there is an optimum value of mass velocity which ensures maximum heat transfer rate. At the reference values of other parameters considered here it is coming as 200 kg m^{-2} s^{-1}. The pressure drop however increases almost linearly with the increase in mass velocity as shown in figure 6c.

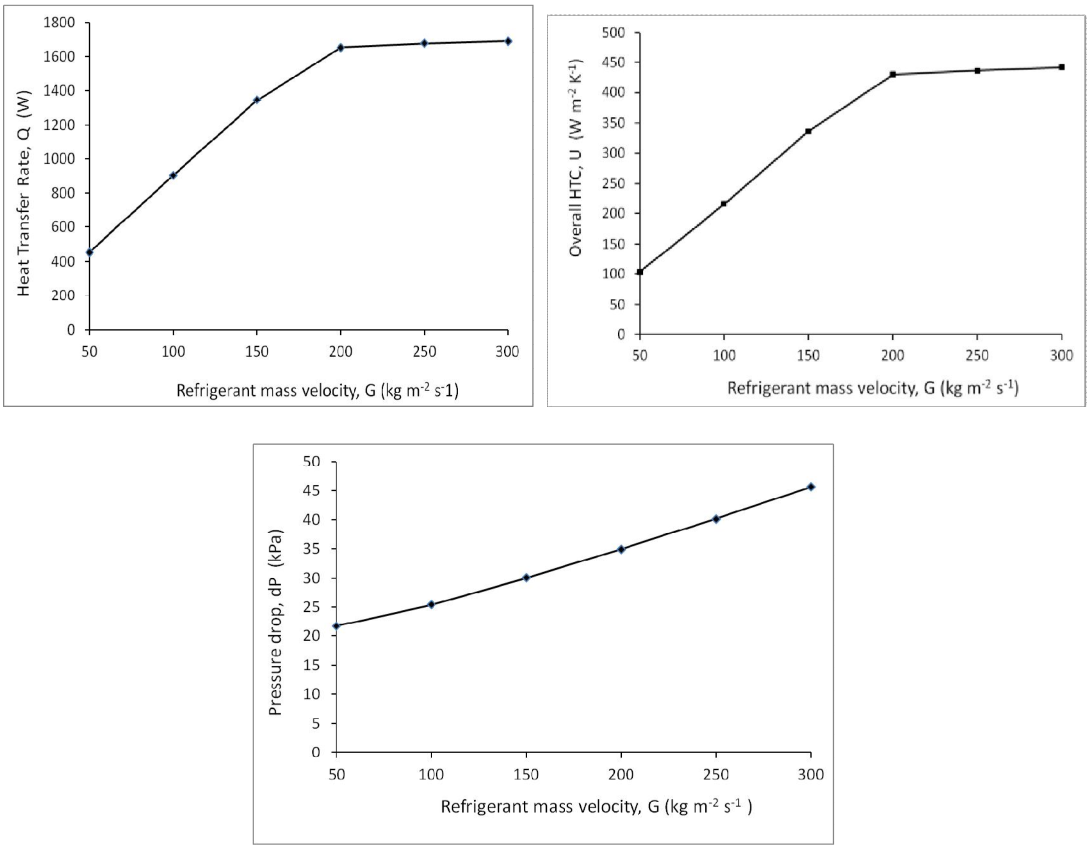


Fig. 6: Effect of refrigerant mass velocity on the performance of evaporator.

4.2.3 Effect of evaporation temperature of refrigerant

Generally, the condition of refrigerant after the expansion valve and at the entrance of evaporator is partially vaporized liquid at saturation temperature corresponding to the evaporator pressure. The highest heat transfer rate comes with the lowest pressure or temperature of refrigerant as shown in figure 7a and reduces with the rise in refrigerant temperature due to lowering of overall temperature difference causing heat transfer between water and refrigerant. However as shown in figure 7b, the overall heat transfer coefficient on the refrigerant side increases at a high rate with the increase in temperature of refrigerant or decrease in net temperature difference between the water and refrigerant. It is highest at 20^0C temperature of refrigerant where the water enters at a temperature of 28^0C. On further higher temperature of refrigerant it again reduces to a lower value. This trend is justified by the fact that at too low temperature of refrigerant, the refrigerant is vaporized earlier due to higher heat transfer rate with higher temperature difference and a larger portion of the evaporator is utilized only in the single phase heat transfer during superheating of vapors. This results in lesser value of the average heat transfer coefficient. With the rise in temperature of refrigerant, the heat transfer rate drops and the refrigerant remains under two phase flow during the larger portion of evaporator tube i.e. dry out point shifts more towards the exit of the evaporator thus improving the average heat transfer coefficient on the refrigerant side. However a very high temperature of refrigerant causes heavy reduction in the heat transfer rate resulting in incomplete evaporation i.e. no dry out point within the evaporator. As the two phase heat transfer coefficient is low at the low vapor quality and increases with its increase, the average value of two phase heat transfer coefficient is lower in incomplete evaporation with lower mean vapor quality than that in complete evaporation with higher mean vapor quality. Within the present data, temperature difference of 10^0C between the water and refrigerant gives maximum value of heat transfer coefficient and so the maximum utilization of evaporator area. Figure 7c shows the falling trend of pressure drop with the rise in saturation temperature of refrigerant which is same as for total heat transfer rate. It is already well known that the pressure drop has a direct relation to the heat transfer rate in a heat exchanger.

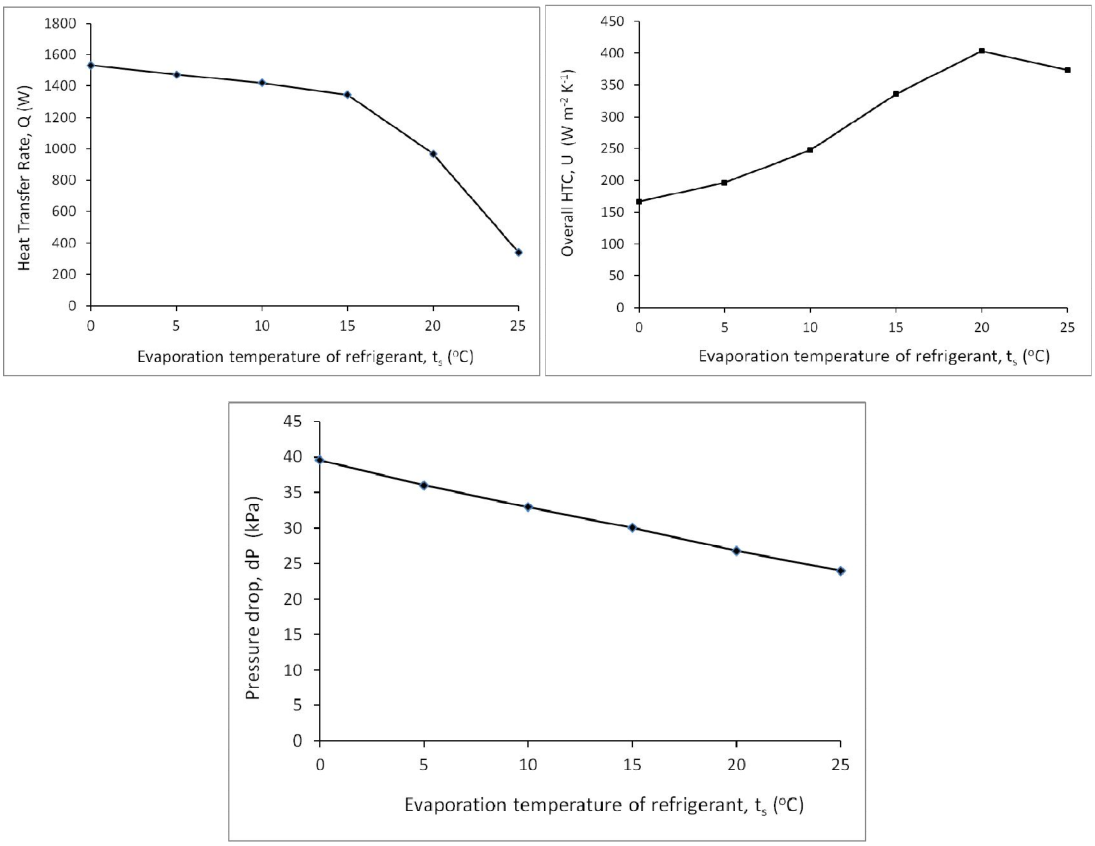


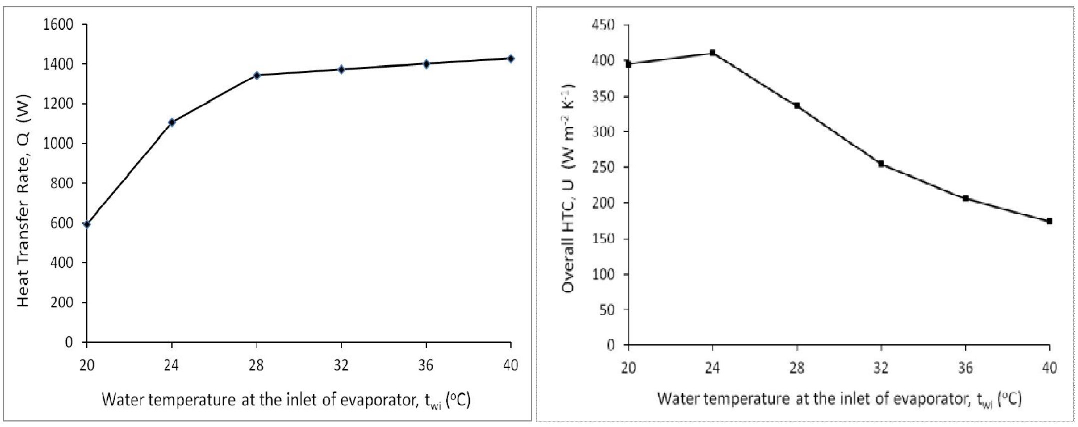
Fig. 7: Effect of evaporation temperature of refrigerant on the performance of evaporator.

4.2.4 Effect of inlet temperature of water

An increase in the temperature of water entering the evaporator shell while keeping the other parameters at their corresponding reference values generally causes an increase in heat transfer rate due to rise in the overall temperature difference between the water and refrigerant. However this increase slows down at a temperature difference more than 10^0C to 12^0C as shown in figure 8a. This is again because of the large reduction in the average heat transfer coefficient on the refrigerant side due to the decreased area under two phase heat transfer and increased area under single phase heat transfer inside the evaporator. Due to this a reduction in the overall heat transfer coefficient is there as shown in fig.8b The explanation for this is also same as given in section 3 above. The best value of heat transfer coefficient and best utilization of the evaporator area comes with 10^0C to 12^0C temperature difference. Regarding pressure drop of refrigerant shown in fig.8c, it slightly increases with increase in water temperature and has resemblance with the heat transfer curve.

4.2.5 Effect of mass flow rate of water

Figure 9a shows that the heat transfer rate rises initially with the increase in mass flow rate of water but starts decreasing after a limit. The rise in heat transfer rate with an increasing mass flow rate is understood by the rise in water side heat transfer coefficient. But simultaneously the refrigerant side average heat transfer coefficient and so the overall heat transfer coefficient lowers down considerably due to lesser areas exposed to two phase heat transfer as a result of increased heat transfer rate as shown in figure 9b. This results in negative effect on further rise in total heat transfer rate with the rise in mass flow rate of water beyond a limit. The trend of the pressure drop of refrigerant with the flow rate of water is also shown in figure 9c. It slightly increases with the increase in flow rate of water because of more area in superheating zone with the reduction in two phase flow area in the evaporator. Actually like the heat transfer rate, the total pressure drop of refrigerant through the evaporator also depends on the relative portion of the evaporator under two-phase flow and single phase flow because both have different values in these two regions. And the relative portions of evaporator under two-phase and single phase heat transfer depend directly on the heat transfer characteristics on both the water side and refrigerant side.



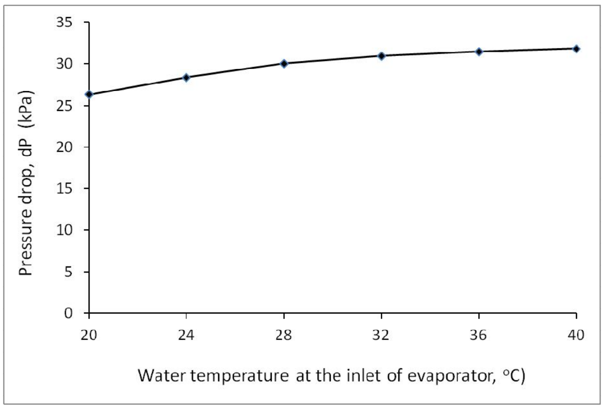


Fig. 8: Effect of inlet temperature of water on the performance of evaporator.

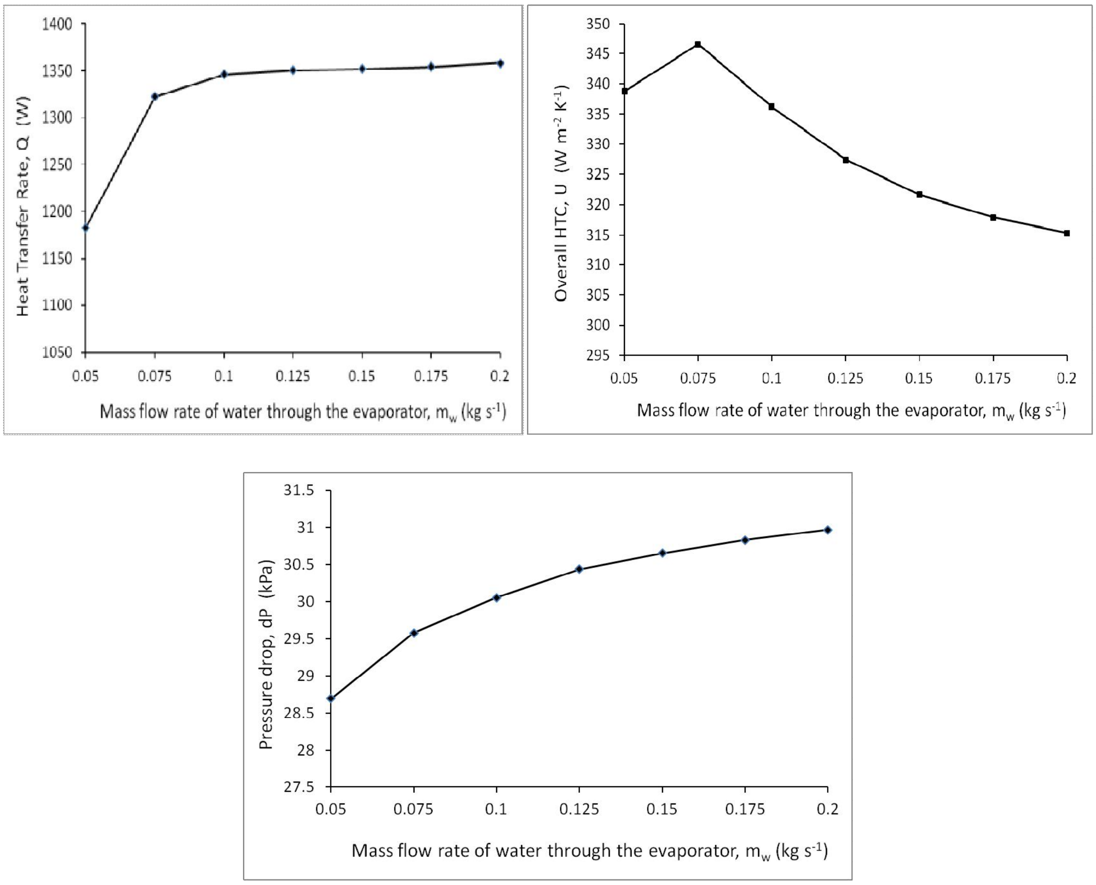


Fig. 9: Effect of mass flow rate of water on the performance of evaporator.

4.2.6 Overall analysis and the applicability of computer program in designing of evaporator:

From the above results it is clear that the most critical parameter in a refrigerant evaporator is refrigerant side average heat transfer coefficient which is affected largely and varies widely with the variation in the input conditions of refrigerant and water.

This is because of the varying ratio of evaporator area under two-phase and single phase flow and the huge difference in two-phase and single phase heat transfer characteristics as shown in fig.10. The separation point between two phase flow and single phase is called dry out point. If the dry out point is near to the exit of the evaporator, the average heat transfer coefficient will be maximized and reduce heavily as the dry out point is shifted away from the exit towards inside of the evaporator. This happens because the heat transfer coefficient in single phase flow after dry out point is less by 5 to 10 times than the heat transfer coefficient in two phase flow before dry out point.

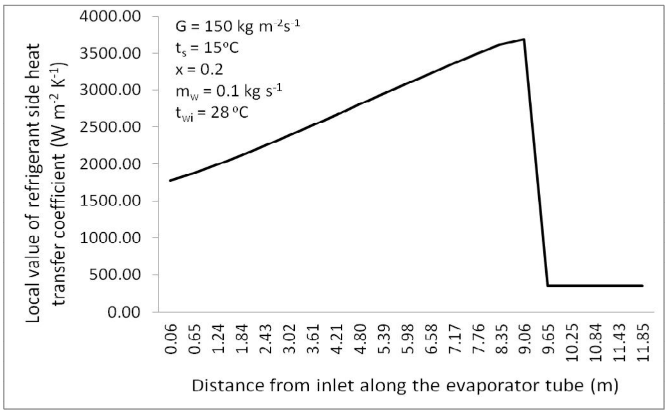


Fig. 10: Variation of heat transfer coefficient on refrigerant side along the length of tube of evaporator.

It was also seen that the improvement of waterside heat transfer conditions simultaneously reduces the average heat transfer coefficient on the refrigerant side due to shifting of dry out point away from the exit and this improvement will not be much beneficial until the input parameters on the refrigerant side are also changed simultaneously to improve heat transfer conditions on the refrigerant side. Thus the optimization of parameters is cumbersome but the computer program developed here is a very easy and fast tool to a designer in checking of performance of evaporator instantly on making alterations in any of the heat exchanger design parameters and the input parameters of both of the fluids. Some simple steps of designing a new evaporator with the help of computer program are as given below:

- Find the optimum mass velocity at the given values of input parameters like pressures, temperatures and mass flow rate of the refrigerant and water and the vapor quality of refrigerant at the inlet of evaporator.
- Calculate the optimum value of cross sectional area of refrigerant tube by dividing the given mass flow rate with the optimum mass velocity. From that the tube diameter is found.
- The tube length can be calculated for the desired degree of superheating of refrigerant at the outlet by running the program at various lengths increased in increments.

5. Conclusion

After detailed study of two phase heat transfer correlations applicable to boiling of HFC-134a in horizontal tube, a computer program in MS Excel was developed for accurate prediction of the performance of a shell and tube evaporator taking into consideration all the variable local parameters like temperature, pressure and vapor quality and also the varying properties of refrigerant with these parameters. An experimental study of the same type of evaporator was also carried out. The predicted and experimental results of performance of evaporator were found in good agreement which validated the computer program for easy and fast use in the design and analysis work.

From the predicted results of program, it was found that the average value of heat transfer coefficient on the refrigerant side is more sensitive to the varying input conditions of both the fluids i.e. refrigerant and water and it varies in a wide range affecting the overall performance of the evaporator. It has a peak value when the dry out point is just near to the evaporator exit and reduces heavily on shifting of dry out point towards the inside of the evaporator. It was concluded that for the highest performance and best utilization of evaporator area, the geometrical parameters and input conditions of refrigerant and water should be kept at such a value which give two phase heat transfer conditions in the maximum area possible on the refrigerant side along with a comparative value of the water side heat transfer coefficient. The computer program was found very useful in the optimization of design parameters to maximize the heat transfer rate of evaporator.

There are some limitations of the computer program due to the assumptions of the same value of parameters within a sub-volume over both the shell side and tube side. On tube side however the size of sub-volume was taken very small thus minimizing the error but on shell side, it was not so as each sub-volume was covering one pair of horizontal tubes and the water temperature for one pair was taken same which may cause slight error in the results. Also the error if any in the chosen correlation for properties of refrigerant and other performance parameters would cause some deviations in the final results. Further work may be carried out towards removing of these limitations.

References

[1] A C Cleland (1986), Computer subroutines for rapid evaluation of refrigerant thermodynamic properties, *International Journal of Refrigeration,* 9, pp. 346-349.

[2] A C Cleland (1992), Polynomial curve-fits for refrigerant thermodynamic properties: extension to include R134a, *International Journal of Refrigeration, 17(*4), pp. 245-249.

[3] D S Jung, M McLinden, R Radermacher and D Didron (1989), Horizontal flow boiling heat transfer experiments with a mixture of R22/ R114, *International Journal of Heat and Mass Transfer,* 32, pp. 131-145.

[4] D S Jung and R Radermacher (1993), Prediction of evaporation heat transfer coefficient and pressure drop of refrigerant mixtures in horizontal tubes, *International Journal of Refrigeration, 16(3)*, pp. 201-209.

[5] F P Incropera and D P DeWitt (1996), Internal Flow, Fundamentals of heat and mass transfer, 4th Ed., *John Wile*

[6] H Takamatsu, S Momoki and T Fuzii (1989), A correlation for forced convective boiling heat transfer of pure refrigerants in a horizontal smooth tube, *International Journal of Heat and Mass Transfer,36(13),* pp. 131-145.

[7] J G Lavin and E H Young (1965), Heat transfer to evaporating refrigerants in two-phase flow, *American Institute of Chemical Engineers Journal, vol.11*, pp. 1124-1132.

[8] J P Hartnett and W J Minkowycz (2000), Two phase evaporative heat transfer coefficients of refrigerant HFC-134a under forced flow conditions in a small horizontal tube, *International Communications in Heat and Mass Transfer, 2,7* pp. 35-48.

[9] J Yu, S Momoki and S Koyama (1999), Experimental study of surface effect on flow boiling heat transfer in horizontal smooth tubes, *International Journal of Heat and Mass Transfer, 42,* pp. 1909-1918.

[10] K Hambraeus (1991), Heat transfer coefficient during two-phase flow boiling of HFC-134a, *International Journal of Refrigeration, 14,* pp. 357-362.

[11] M K Mittal, R Kumar and A Gupta (2010), An experimental study of the flow of R-407C in an adiabatic helical capillary tube, *Int. J. Refrig. 33*, pp. 840-847.

[12] N J Dembi, P L Dhar and C P Arora (1978), Statistical analysis of heat transfer data for convective boiling of refrigerants in a horizontal tube, *Letters in Heat and Mass Transfer, vol. 5,* pp. 287-296.

[13] P L Dhar, N J Dembi and C P Arora (1979), A new correlation of heat transfer data obtained with nucleate boiling of refrigerants in a horizontal tube, *Letters in Heat and Mass Transfer, vol. 6,* pp. 171-177.

[14] R J Moffat (1988), Describing the Uncertainties in Experimental Results, *Experimental Thermal and Fluid Science 1,* pp. 3-17.

[15] V I Dmitriyev and V E Pisarenco (1984), Determination of optimum refrigerant charge for domestic refrigerator units, *Int. J. Refrig. 7-3*, pp.178-180.