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A REVIEW OF FLOW CHARACTERISTICS OF REFRIGERANT FLOWING THROUGH NON ADIABATIC CAPILLARY TUBE

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ABSTRACT

This paper gives summary of research on the characteristics of different refrigerant considering Non Adiabatic flow inside the capillary tube expansion device for various applications. This paper gives the summary of range of geometric and operating parameters like capillary tube length, diameter, suction line diameter, suction line length, heat exchanger length, surface roughness of capillary tube, suction line inlet temperature, coil pitch, inlet sub cooling, condenser pressure, and evaporator pressure etc. The outcome of different research is summarized in tabular form. This paper also gives type of approach, correlation proposed and some special information about investigation. It is found that the literature in Non Adiabatic capillary tube is scare so more investigation is required.

Key words: Expansion device, straight & coiled capillary tube, correlations, parameter

INTRODUCTION

A capillary tube is a constant restriction type expansion device used in small vapor compression refrigeration system (below 10KW) like domestic refrigerator, window air conditioning system. Since friction resistance inside capillary tube is directly proportional to length and inversely proportional to inside diameter so that length of capillary tube is kept large (2 to 5m) and diameter is kept small (0.00033m to 0.0015m). The pressure drop through the capillary tube is due to friction and acceleration. Capillary tube maintains flow rate of fluid (refrigerant) flowing through it at desired level so capillary tube works as an automatic flow rate controller for the refrigerant when load is varying. Capillary tube is most commonly used as expansion device because of this compressor can start at low torque. Also capillary tube is simple, less costly and there is no moving part in this. In most of the literature capillary tube is Adiabatic means no heat transfer from the capillary tube to the surrounding. A few researches are found in non Adiabatic capillary tube which works as counter flow heat exchanger in simple vapor compression refrigeration system. So this paper gives the information about refrigerant used, correlation proposed, and range of parameters, and other important information related to Non Adiabatic capillary tube.

Mainly two types of capillary tube are used in simple vapor Compression refrigeration system:

Adiabatic capillary tube: The Adiabatic capillary tube arrangement is shown in fig1. In vapor compression refrigeration system the sub cooled liquid refrigerant from condenser enters in the capillary tube and flows without transferring heat to surrounding. When sub cooled liquid flows through Adiabatic capillary tube the pressure is reduces linearly and a point is reached where the liquid flashes into vapor this point is known as flash point at which first particle of vapor is formed. The temperature of refrigerant remains constant until flash point and reducing after flash point.

Non Adiabatic capillary tube : In non Adiabatic type of arrangement the capillary tube is bonded with suction line by soldering or brazing as shown in fig2 So that arrangement is works as counter flow heat exchanger. By making this arrangement there are two advantages first there is delay in vaporization and second superheated vapor is going to compressor so compressor damage can be avoided.

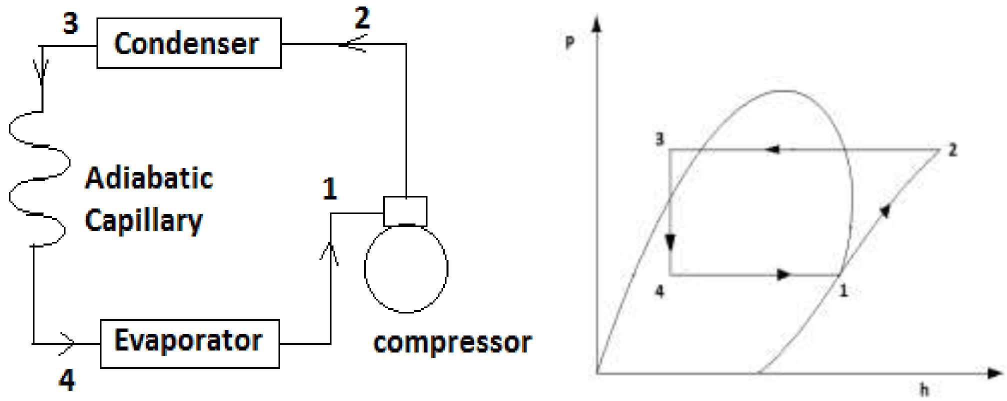

Fig1: Simple vapor compression system with Adiabatic capillary tube: a) block diagram b) p-h diagram

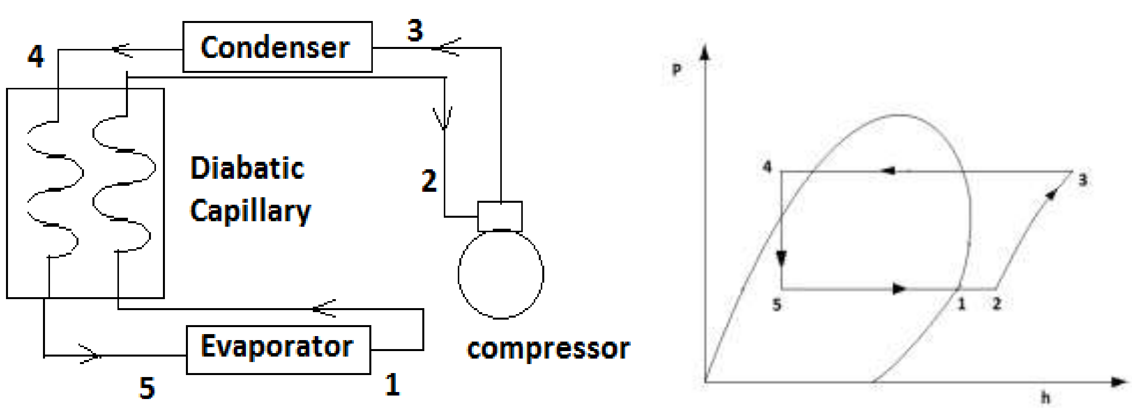

Fig2: Simple vapor compression system with Non Adiabatic capillary tube: a) block diagram b) p-h diagram

In vapor compression refrigeration system Non Adiabatic capillary tube is used in two configuration first in lateral type and second in concentric type.

In **lateral type** of configurations as shown in fig3(b) the capillary tube is directly brazed or soldered with the suction line going to the compressor so that forming a counter flow heat exchanger in which the heat is transferred from capillary tube to suction line which is going to compressor. Some initial length of capillary tube is kept Adiabatic and some last length of capillary tube is also kept Adiabatic and remaining is bonded with suction line.

In **concentric type** of configuration as shown in fig3 (a) capillary tube is kept inside the suction line and heat is transferred from tube to suction line. The hot refrigerant is flowing through the capillary tube and the cold refrigerant in the annulus between the capillary tube outer surface and suction line inner surface.

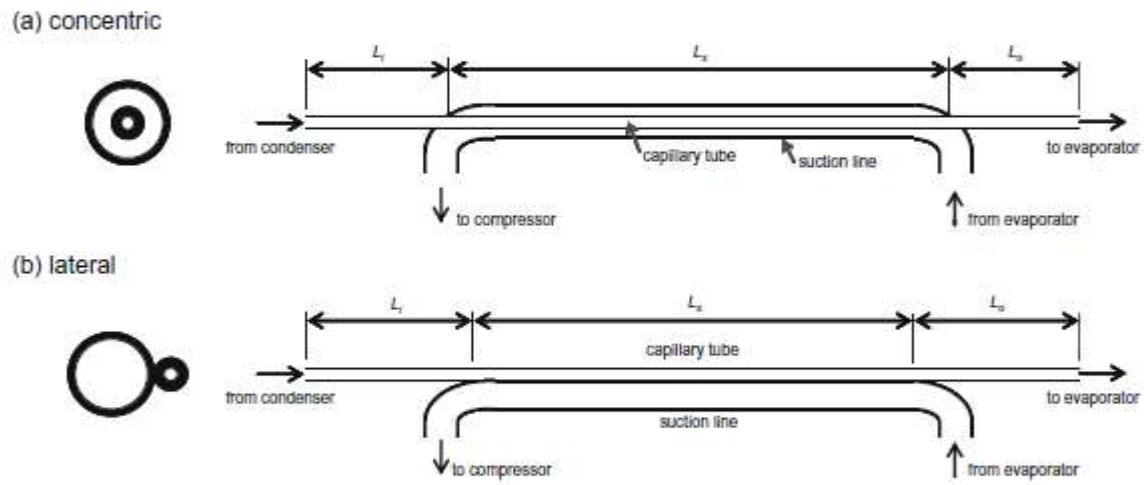

Fig3: a) concentric type b) lateral type arrangement {Source Ref.[23]}

In both lateral and concentric type of arrangement capillary tube can be straight, spiral, or helical. The helically coiled Non Adiabatic capillary tube is most widely used in domestic refrigerator. if the capillary tube is used in straight manner it requires more space as compare to helical or spiral but if capillary tube is used in vertical position the space requirement is reduced because there is sufficient space in back side of the refrigerator.

LITERATURE REVIEW

Use of Non Adiabatic capillary tube is started in 1948 by Staeblar[1]. Staeblar determined the length of capillary tube by using the capacity balance method. 1.22 m length is bonded with suction line. Bolstad and Jordan [2] have find that evaporator pressure has a minor effect on mass flow rate of R-12. Temperature falls rapidly in heat exchanger region. Pate and Tree [3] considering counter flow heat exchanger in which air flows in suction line and refrigerant in capillary tube. They found that due to increase in sub cooling, delay in flash point and thus shorter two phase region. Again Pate and Tree [4] shows the effect of capillary tube length on pressure, quality and temperature. Escanes[5] developed a numerical simulation model by control volume formulation. Both critical and non critical flow has been considered. Bansal et al.[6] proposed a empirical model for refrigerant R-134a. Model is found to agree with earlier studies within ±8% for both the Adiabatic and Non Adiabatic capillary tubes. Bansal has proposed a correlation for length of capillary tube. Dekang chen et al.;[7] experimentally investigate the flow of R-134a and found that under pressure of vaporization decreases with an increase of the heat transfer between the capillary tube and suction line. O. Garc-Valladares[8] made a numerical simulation using R-134a and found Good degree of Correlation between numerical and experimental results for the case of a Concentric Non Adiabatic capillary. B. Xu, et al.[9] also proposed a numerical model for R-134a. The model predictions agreed with available experimental and analytical data to within ±20%. Jiraporn Sinpiboon et al[10] gives a mathematical model for R-12, R-134a and R-152a and measured mass flow rate of R134a has discrepancy of 14.3% and 9.50% for different model. Claudio Melo et al. [11] make Experimental investigation for R-600a.He found that absolute mean deviation error for the refrigerant mass flow rate and suction line outlet temperature was 0.07 kg/h (5.1%) and 0.6°c , respectively. P.K. Bansal et al[12] gives numerical model for R-134a. It is found that the Non Adiabatic capillary tube flow characteristic is discontinuous in some situations. Yasar Islamoglu et al[13] uses artificial neural network (ANN) approach. The results showed that the ANN approach could be considered as an alternative and practical technique to evaluate the refrigerant suction line outlet temperature and mass flow rate. Y. Chen, et al. [14] gives numerical model for $Co_{2.}$ The increase of inner diameter, cooling pressure and outside heat transfer coefficient will lead to longer capillary length also increasing evaporating temperature and cooling capacity lead to a shorter length.

C. Yang et al.[15] gives numerical model. It was found that R-134a performs better in terms of heat transfer rate and evaporator capacity than R-600a. O. Garcı´a-Valladares et al.[16] make Numerical simulation for R-12 R-134a, R22, R-152a and R410A. A detailed numerical model of concentric and lateral capillary tube suction line heat exchangers have been developed. O. Garcı´a-Valladares et al[17] gives Numerical simulation for R410A R-134a R-600a and R152a. Of the 196 data points evaluated for mass flow rate 96.4% are within an error of ±15%, 81.1% are within ± 10% with a mean deviation of ±6.3%. Neeraj Agrawal et al [18] give Homogeneous flow model for CO_2. The increased mass flow through larger tubes causes increase in heat transfer rate by about 68%, for a tube diameter increase of 50%. Again Neeraj Agrawal et al.[19] gives Homogeneous flow model for CO_2 . As the capillary becomes larger, decrease in throttling effect increases the mass flow rate, hence heat transfer rate increases. Christian J.L. Hermes et al.[20] uses Computational model for HFC-134a HC- 600a. It was found that the model predicts 91.5% of the measured refrigerant mass flow rate for Adiabatic and 79.3% for Non Adiabatic flows within an error band of ±10%. A.L. Seixlack et al.[21] investigate R-134a. Mass flow rate can be predicted within error band of 3.6% and 5% for two fluid model and homogenous model respectively. Mohd. Kaleem Khana et al [22] experimentally investigate R-134a. It has been found the proposed correlation predicts the refrigerant mass flow rate in the error band of ±7% of the measured experimental mass flow rate. Christian J.L. Hermes et al.,[23] Analytically investigate flow of R-134a and HC 600a. Comparisons between the model predictions and the experimental data revealed that more than 90% and nearly 100% of all data can be predicted within ±10% and ±15% error bands, respectively.

AUTHOR	REFR	APPROACH	RANGE OF PARAMETERS	CORRELATIONS	CONCLUSION	REMARK
Staeblar[1]refrigerating engg.(1948)	R-12 R-22	Numerical model	d=0.787-1.397mm, T_o=-28.9to-1.1° c, T_K=30° c and 42.2° c		Length of Non Adiabatic capillary tube is determined by using R-12 and R-22	Capacity balance correlation is used
Bolstad et al.[2] refrigerating engg.(1949)	R-12	Numerical model	D=0.66-mm, L=1.83-5.49m, P_k=827.4-1103.3kpa, P_o=103.4kpa		Temperature and pressure profiles are drawn	Friction factor relations has been developed
M.B.Pate et al.[3]ASHRAE Trans.90(1984)	R-12	Mathematical model	D=0.71mm, D_s=6.35mm, L_{hx}=2.086m, L=2.953m		Temperature and pressure profiles are drawn	Considered metastable flow for Adiabatic and not for Non Adiabatic
M.B.Pate et al.[4]ASHRAE Trans.90(1984)	R-12	Linear quality model	d=0.71mm, D_s=6.35mm, P_k=965.2kpa, ΔT_{sub} =7.3, 9.9° c,		Effect of capillary tube length on pressure temperature and Quality	Implicit finite difference method is used
Escanes et al. Int. J.refrigeration [5] (1995)	R-12 R-22	Numerical simulation	d=1.0mm, D_s=6.35mm, P_k=800kpa, L=4m, T_i=25° c		Data Validation by using data of Bolstad (1948) and MiKol (1963)	Control volume method used
Bansal et al.[6] Int.J.refrigeration(96)	R-134a	Empirical model	Δp= 1220.5kPa, ε = 6.06 x 10^{-4}, m = 4kgh^{-1} ΔT_{hx} = 30 K, ϕ = 0.8, ΔT_{sub} = 5.5 K d=0.66mm	$L= k1.\delta p.d.(\delta T_{sub}+k_2).\left(\frac{k_3-\varepsilon}{G^2}\right)+k_4\delta T_{hx}.\phi$	Predictions of the proposed empirical model are found to agree with earlier studies within ±8% for both the Adiabatic and non Adiabatic capillary tubes.	Both Adiabatic and Non Adiabatic capillary tube has been investigated.
Dekang chen et al[7] Int. J.refrigeration(01)	R-134a	Experimental investigation	T_i=28.2-41.1°c, P_i=9.13-13.4 bar, ΔT_{sub} =1.83-16.8°c , G_c=2550-5800 kg/s.m^2, G_s=0-1200 kg/s.m^2	$(p_{sat}-p_v)\sqrt{k}T_{cv} = 0.679\left(\frac{\rho_l}{\rho_l-\rho_g}\right)Re_c^{\;0.914}\left(\frac{\Delta Tsub}{T_{cri}}\right)^{-0.208}\left(\frac{D_c}{D}\right)^{-3.18}$	It was shown that under pressure of vaporization decreases with an increase of the heat transfer between the capillary tube and suction line until it disappears.	Metastable flow through Non Adiabatic capillary tube, under pressure of vaporization.
O. Garc-Valladares[8] Applied Thermal Engineering(02)	R-134a	Numerical simulation	**Case1**- ΔT_{sc} = 5°c, ΔT_{sh} = 11.7°c, P_i = 14bar, P_o=1.1694bar, **Case2**- ΔT_{sc} = 5.2°c, ΔT_{sh} =12.1 °c, P_i =9bar P_o=1.1586bar		Good degree of Correlation between numerical and experimental results have been observed for the case of a Concentric Non Adiabatic capillary	Results presented metastable flow and non-metastable flow modeling,
B. Xu, et al.[9] Applied Thermal Engineering (2002)	R-134a	Numerical model	L_{ent} = 0.7m, L_{hx}= 1.0m, D_c= 0.66mm, $D_{c,o}$= 2mm, D_s= 6.6mm, $D_{s,o}$= 10mm, , ε_c= 0.00046mm, ε_s= 0.00046mm, m= 0.001kg/s, T_{evp}= 258.15K, T_{con}= 321.2K, ΔT_{sub} = 1K, ΔT_{sup} = 2K		The model predictions agreed with available experimental and analytical data to within ±20%.	the present model is capable of handling alternative refrigerant mixtures, .

Jiraporn Sinpiboon et al[10]. Applied Thermal Engineering (2002)	R-12 R-134a R-152a	Mathematical model	T_{con}= 311-327K, $T_{s,I}$ = 258-269.3K, ΔT_{sup} = 2.8-9.9K, D_c= 0.66-0.83mm, D_s= 5.11-9.525mm, L_{ent} = 0.152-0.533m, L_{hx}= 0.760-1.780m, P_{con}= 9-14bar, L_f= 0.356-1.8m		for two-phase flow region measured mass flow rate of R134a has discrepancy of 14.3% and 9.50% for different model	Three cases are considered 1) single phase flow region 2) end of single phase 3) Two phase flow.
Claudio Melo et al. [11]Applied Thermal Engineering (2002)	R-600a	Experimental investigation	P_i = 5.01-6.53bar, , P_o= 0.58-0.90bar, ΔT_{sub} =5-10.2°c, D_c= 0.553-0.766mm, T_i= -20.07to -11.9°c, L_{tc}= 1.0-2.2m, L= 3.0-4.0m, D_s= 6.30-7.86mm, L_e= 0.2-0.6m, T_{exp}= 8.3-29.6°c, m_{exp} =0.74-3.0kg/h, m_{emp}= 0.85-2.89kg/h,	m_{emp}= - 7.1650 + 0.1755Pin +0.8454L + 12.7375D + 0.0276Sub + 0.0960Ltc-0.0005PinTin -0.0150SubLe -1.6512DL + 0.0024LtcDs Temp = 10.0861 + 2.3625Pin + 2.4964Sub + 5.3390D + 11.4987Ltc-3.1265Ds- 0.1446PinDs -4.4467SubD + 0.2263TinLtc -0.0728LLe	Absolute mean deviation error for the refrigerant mass flow rate and suction line outlet temperature was 0.07 kg/h (5.1%) and 0.6°c , respectively.	It is anticipated that the proposed correlations will become a powerful tool for designers modeling HC-600a refrigeration systems
P.K. Bansal et al[12]. Applied Thermal Engineering (2003)	R-134a	Numerical model	L_{ent} = 0.7m, L_{hx}= 1.0m, D_c= 0.66mm, $D_{c,o}$= 2mm, D_s= 6.6mm, $D_{s,o}$= 10mm, , ε_c=0.00046mm, ε_s= 0.00046mm, m= 0.001kg/s, T_{evp}= 258.15K, T_{con}= 321.2K, ΔT_{sub} = 1K, ΔT_{sup} = 2K	$\frac{dp_c}{dz} = f_c \frac{G_c{}^2 v_c}{2D_c} + G_c{}^2 . \frac{dv_c}{dz} + \frac{g\sin\theta}{v_c}$ $dh_c = -\frac{dq}{m} - G_c{}^2 . \frac{dv_c{}^2}{2} -$	It is found that the non Adiabatic capillary tube flow characteristic is discontinuous in some situations and the discontinuity is caused by the re-condensation of the refrigerant within the heat exchanger due to strong heat exchange with the suction line.	
Yasar Islamoglu et al[13]. Energy Conversion and Management (2005)	Not Speci-fied	artificial neural network (ANN) model	P_i = 5.01-6.53bar, ΔT_{sub} = 5-10.2°c, D_c= 0.553-0.766mm, T_i= -20.07to -11.9°c, L_{tc}= 1.0-2.2m, L=3.0-4.0m, D_s= 6.30-7.86mm, L_{ent} =0.2- 0.6m, T=8.3-29.9°c,m=0.74-3.02kg/h	m_{emp}= - 7.1650 + 0.1755Pin +0.8454L + 12.7375D + 0.0276Sub + 0.0960Ltc - 0.0005PinTin – 0.0150SubLe – 1.6512DL + 0.0024LtcDs Temp = 10.0861 + 2.3625Pin + 2.4964Sub + 5.3390D + 11.4987Ltc – 3.1265Ds -0.1446PinDs – 4.4467SubD +	The results showed that the ANN approach could be considered as an alternative and practical technique to evaluate the refrigerant suction line outlet temperature and mass flow rate	Although this approach is generally time consuming but feasible due to its ability to learn and generalize the complex data set with a wide range of Experimental conditions.

				0.2263TinLtc -0.0728LLe		
Y. Chen, et al.[14] Applied Thermal Engineering (2005)	Co_2	Numerical model	Q_0 = 7 kW, $t_{i,c}$ = 45°c, t_0 = 0°c, P_{con} = 1.2 × 10^4 kPa, d_i = 1.85 mm, , ε = 0.0015 mm, and α = 400 W/(m^2 K).		The increase of inner diameter , cooling pressure P_k, and outside heat transfer coefficient a will lead to longer capillary length L ; while increasing evaporating temperature t_0, and cooling capacity Q_0 lead to a shorter L.	The present model can be used for both system design and performance evaluation, it is also very helpful in understanding the supercritical flow behavior inside capillary
C. Yang et al.[15] Applied Thermal Engineering (2005)	R-134a R-600a	Numerical model	L_{ent} = 0.5m, L_{hx}= 1.5m, L_c= 4.5m, D_c= 0.86mm, D_s= 6.6mm, T_c= 315.15K, T_e= 260.15K, ΔT_{sub} = 1K, ΔT_{sup} = 2K	$\frac{dp_c}{dz} = f_c \frac{{G_c}^2 v_c}{2D_c} + {G_c}^2 . \frac{dv_c}{dz} + \frac{g sin\theta}{v_c}$ $dh_c = -\frac{dq}{m} - {G_c}^2 . \frac{d{v_c}^2}{2} -$	It was also found that R-134a performs better in terms of heat transfer rate and evaporator Capacity than R-600a.	The simulation model also postulates the situation where heat may be transferred from capillary tube to the ambient air, before entering the CT-S LHX.
O. Garcı´a-Valladares et al.[16] Int. J. Refrigeration (2007)	R-12 R-134a, R22, R-152a R410A	Numerical simulation	1.44× 10^3 kgm^{-2} s^{-1}< G< 5.09× 10^3 kgm^{-2} s^{-1}, 0 K< ΔT_{sc}< 17 K, 0.66× 10^{-3} m< D1< 1.17× 10^{-3} m.		A detailed numerical model of the thermal and fluid dynamic behavior of concentric and lateral capillary tube suction line heat exchangers have been developed	The simulation has been implemented on the basis of an implicit step-by-step numerical scheme for the fluid flow, and an implicit central difference numerical scheme in the solids.

O. Garcı´a-Valladares et al[17] International Journal of Refrigeration (2007)	R410A R-134a R-600a R152a	Numerical simulation	$D_{c,i}$= 0.553-0.991mm, $D_{c,o}$= 1.5-2.06mm, $D_{s,i}$= 4.80-8.10mm, L_c= 2.06-5.50m, L_{ent}= 0.15-3.4mm, L_{hx}=0.51-2.54mm, , ε =0.45-1.50mm, P_{con}= 0.41-2.89mm, ΔT_{sub} = 1.2-19.22°c		Of the 196 data points evaluated for mass flow rate 96.4% are within an error of ±15%, 81.1% are within ± 10% with a mean deviation of ±6.3%.	
Neeraj Agrawal et al [18] Int. J.Thermal Sciences (2008)	CO_2	Homogeneous flow model	$D_{c,i}$= 1mm, , $D_{c,o}$= 3.17mm, , ε = 0.0015mm, $D_{s,i}$= 9.35mm, $D_{s,o}$= 12mm, L_{hx}= 0.3m, L_{ent}= 0.9m, L_f= 0.1m,		The increased mass flow through larger tubes causes increase in heat transfer rate by about 68% for a tube diameter increase of 50%.	Lowering the evaporator temperature is more effective for heat transfer from capillary compared to the gas cooler Temperature.
Neeraj Agrawal et al.[19] Energy Conversion and Management (2008)	CO_2	Homogeneous flow model	, L_c= 1.3m, D_c= 3.17mm, , ε = 1.5µm, T_{evp}= 276-288K, L_{ent}= 0.9m, D_s= 3/8inch, T_{con}= 310-318K, L_{hx}= 0.3m.		As the capillary becomes larger decrease in throttling effect increase the mass flow rate,	Throttling effect decreases rapidly as internal tube diameter becomes larger
Christian J.L. Hermes et al.[20] international journal of refrigeration (2008)	HFC-134a HC-600a,	Computational model	D_c= 0.553-0.830mm, L_c= 3.00-4.00m, D_s= 4.800-7.860mm, L_{hx}= 0.998-2.670m, L_{ent}= 0.190-1.070m, p_{con}= 500-1411kpa, , ΔT_{sub} = 2.9-20.4°c, , p_{evp}= 59-159kpa, T_{is}= -20.7-5.9°c, m= 0.74-8.74kg/h.		It was found that the model predicts 79.3% for non Adiabatic flows within an error band of ±10%.	The model solves non-Adiabatic flows as fast as Adiabatic ones.
A.L. Seixlack et al.[21], Applied Thermal Engineering(09)	R-134a	Numerical model	D_s= 4.8-7.86mm, L_{hx}= 1.597-2.670m, L_f= 0.533-2.316m, ε = 0.58-2.13 µm, D_c= 0.61-0.83mm, L_c= 4.0-4.10m.		Mass flow rate can be predicted within error band of 3.6% and 5% for two fluid model and homogenous model respectively	Some computational results referring to the quality, void fraction and velocities of each phase are also presented

Mohd. Kaleem Khana et al [22] international journal of refrigerator (2009)	R-134a	Experimental investigation	D_c= 1.12-1.63mm, , L_c= 2.4-6.4m, L_{hx}= 1.6-5.6m, pitch= 20-60mm, ΔT_{sub}= 0-25°c, P_{in}=740kpa,	$\frac{m}{d\mu_f} = c\left(\frac{d^2\rho_f p_{in}}{\mu_f{}^2}\right)^{0.6547} \left(\frac{d^2\rho_f p_{s,in}}{\mu_f{}^2}\right)^{-0.0018} \left(\frac{l}{d}\right)^{-0.3985} \left(\frac{L_{hx}}{d}\right)^{0.1004} \left(\frac{d^2\rho_f{}^2 c_{pf}}{\mu_f{}^2}\Delta t_{sub}\right)^{0.1013} \left(\frac{d^2\rho_f{}^2 c_{pf}}{\mu_f{}^2}\Delta t_{sub}\right)^{-0.0762} \times$	It has been found the proposed correlation predicts the refrigerant mass flow rate in the error band of ±7% of the measured experimental mass flow rate.	The present experimental investigation has been carried out to investigate the effects of various geometric parameters on the mass flow rate of R-134a through Non Adiabatic spiral Capillary tube.
Christian J.L. Hermes et al.,[23] Applied Thermal Engineering (2010)	R-134a HC 600a	Analytical model	D_c= 0.553-0.83mm, L_c= 3.00-4.00m, D_s= 6.300-7.860mm, $L_{hx\ 0.998-2.200m,}$, L_{ent}= 0.190-1.070m, P_{con}= 500-1411kpa, ΔT_{sub}= 2.9-20.4°c, p_{evp}= 59-143kpa.	$\left(\frac{w}{w}\right) = 1.11\left(\frac{L_s}{L_c}\right)^{0.183} \left(\frac{D_s}{D_c}\right)^{-0.335} \varepsilon^{-0.366} \left(\frac{v_f\ \eta_f}{v_v\ \eta_v}\right)^{-0.281}$	Comparisons between the model predictions and the experimental data revealed that more than 90% and nearly 100% of all data can be predicted within ±10% and ±15% error bands, respectively.	The heat and fluid flow were treated as independent phenomena,

CONCLUSION

This paper presents the summary of literature on Non Adiabatic capillary tube with various refrigerants, range of parameter selected, and type of approach. In most of the literature the effect of various geometric and operating parameters on mass flow rate has given. In spiral type of Non Adiabatic capillary rube the mass flow rate is less as compare to straight Non Adiabatic capillary tube. It is found in literature that the length for Non Adiabatic capillary tube is more as compare to adiabatic capillary tube. The literature available for Non Adiabatic capillary tube is scare so more investigation is required**.**

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