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Theoretical Design of adiabatic capillary tube of a domestic refrigerator using refrigerant R-600a.

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***Abstract:* -** This paper develops a more accurate theoretical procedure for the design of adiabatic capillary tube of a domestic refrigerator considering a rigorous pressure drop analysis on the refrigerant R-600a while expanding through that tube accompanied with phase change through flash vaporization. Here this eliminates the contradiction of existing concepts on the negative value of the frictional pressure drop after a short distance of expansion due to a large part contribution of the actual pressure drop towards the momentum gain pressure drop. Also this verifies that the momentum gain through phase change is by consumption of internal energy part of the enthalpy and no part of the actual pressure drop energy is used in this respect. So with the concept of nearly total pressure drop being used in overcoming the friction the design of an adiabatic capillary tube of available I mm diameter for 0.1 ton refrigeration capacity has been carried out here. This design procedure causes some increase in the required length of the capillary for a given refrigeration capacity due to the omission of momentum pressure drop concept of different references, but is more accurate with consideration of actual changes involved in the expansion. The procedure is applicable for any other refrigerant of any refrigeration capacity

***Keywords:* -** *Adiabatic capillary tube, finite element iterative calculation, frictional pressure drop, momentum pressure drop, suction line heat exchanger.*

I. INTRODUCTION

A capillary tube is used between the condenser outlet and evaporator inlet of a domestic refrigerator working as simple expansion device in vapour compression refrigeration system (VCRS) to drop the pressure and the corresponding saturation temperature of refrigerant from condenser condition to the evaporator condition. Here as the liquid refrigerant from the condenser flows through the capillary tube, causes the expansion following the flash vaporization due to the drop of internal energy and hence enthalpy, so also with change of other state properties. During this expansion as the refrigerant temperature falls much below the ambient temperature, so the outer heat has the possibility to flow into the refrigerant reducing the heat extraction capacity from the refrigerating space. So the capillary tube is made adiabatic through thermal insulation. Here the suitable diameter & length of the capillary tube for a given pressure drop and maintaining corresponding saturation temperature in the evaporator is essential and hence needs their special designs.

The basic theory of capillary design is the fluid friction Darcy pressure drop relationship according to which greater pressure drop can be achieved with decreased diameter and increased length of the capillary tubes. Since the capillary tubes are available of only specified diameters so their actual design emphasizes on determination of lengths for a given refrigeration effect. Again as the refrigerant flows from the condenser through the capillary tube, pressure drops to the evaporator condition by fluid friction with simultaneous drop of saturation temperature, internal energy and enthalpy, but gain of specific volume, velocity and hence kinetic energy through phase change as explained in my own paper [1]. Here a more realistic concept has been used for the cause of momentum or kinetic energy gain than the pressure energy source as in the existing theory of many references. Also the above paper explains how no part of the total pressure drop is used in momentum gain, but shared in only frictional head drop and gravitational head gain.

So this paper is an extension work of my previous issue [1] for design of an adiabatic capillary tube as a simple expansion device of 0.1 ton domestic refrigerator by neglecting the gravitational head gain and momentum pressure head drop and taking the actual total pressure drop as the frictional head drop. Here the design procedure uses the finite element analysis (FEA) with iterative calculations on refrigerant R-600a properties at different nodes and corresponding mass flow rate giving the above refrigeration effect.

I. DESIGN CONDITION FOR THE CAPILLARY TUBE

This paper aims to design a capillary tube used in a specially arranged vapour compression refrigeration system (VCRS) between a suction line heat exchanger (SLHX) as condensate sub-cooler and evaporator outlet as vapour super heater. The arrangement is shown as in the following figure-1.

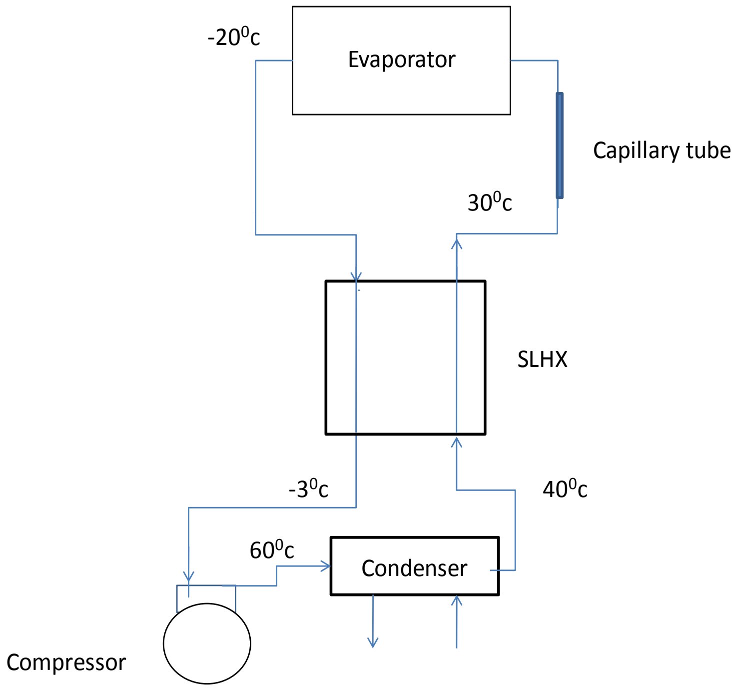

Fig-1. Positions of installation of non adiabatic SLHX & adiabatic capillary tube in VCRS.

Here the SLHX receives the fresh condensate as hot fluid at 40^0c and sub-cools to 30^0c at constant pressure by exchanging heat to the fresh vapour from the evaporator at -20^0c and causes the equivalent super heating. So here the inlet condition of the refrigerant to the capillary tube is sub cooled liquid at 30^0c corresponding to the condenser saturation temperature of 40^0c and outlet condition of -20^0c saturation temperature corresponding to the evaporator pressure. Again as the liquid refrigerant expands during the flow through the capillary tube with fall of temperature below the ambient, that may cause the undesired flow of heat from the ambient to the refrigerant inside the tube and reduces its heat extraction capacity from the refrigerating space and hence reduces the refrigeration effect. So the capillary tube wall is thermally insulated or made adiabatic and so it is assumed in this design. Also here the design of the capillary tube considers for a domestic refrigerator of 0.1 ton refrigeration capacity is based on the use of refrigerant R-600a which is non toxic and has no effect on ozone layer depletion. Again R-600a has additional advantages of low saturation range of pressure between evaporator and condenser than any other refrigerant within which the compression work requirement for same refrigeration capacity becomes lower and hence improves the COP of the refrigerator.

II. EXISTING THERY FOR DESIGN OF ADIABATIC CAPILLARY TUBES

Before the analysis on this paper work an extensive data for the adiabatic capillary tubes and reliable diagrams have been thoroughly studied in the works of Bittle et al. (1998) [2], ASHRAE (1994) [3], Melo et al. (1999) [4], Sami and Maltais (2000) [5].The early works conducted by Hopkins (1950) [6] and Whitesel (1957) [7] resulted in developing rating charts for (R-12) and (R-22). ASHRAE (1979) [8] established graphical method of representation for the capillary tube rating for specified entering conditions. More detailed rating charts are also included in ASHRAE (1998) [9] for pure (R-134a),(R-22), and (R-410A). The object of these curves is to establish the required capillary refrigerant flow rate for specified geometry and entering condition. This condition represents the entering pressure and the sub-cooling available for the refrigerant at inlet to the capillary tube. A generalized prediction equation based on tests with (R-134a), (R-22), and (R-410A), Wolf et al. (1995) [10] was developed for the prediction of the refrigerant mass flow rate through the capillary tube. In the present work a step by step numerical analysis has been developed for the prediction of the performance of the capillary tubes. The results of the present work provide a powerful tool for an estimation of the capillary tube geometry and flow operating conditions.

According to the existing theory in different journal papers and text books as the condensate refrigerant flows through the capillary tube with expansion from high condenser pressure to the low evaporator pressure and corresponding drop of saturation temperature, there occurs simultaneous flash vaporization and increase of specific volume resulting in momentum or kinetic energy gain. So the available pressure drop energy is utilized in overcoming the fluid friction, gain of momentum and potential energies. Again the expansion in the capillary tube is primarily assumed isenthalpic along the vertical line k_s-k-5 on the ph-diagram of fig.-2. But as the actual expansion is assumed adiabatic, so the refrigerant gains momentum using own enthalpy, and the actual expansion follows the curved Fanno line k_s-k-5′ with decreasing enthalpy. Here the decrease of enthalpy at any pressure is equal to the gain of kinetic energy. Again the expansion of the saturated liquid refrigerant from the condenser in the capillary tube follows the expansion through flash vaporization in the liquid- vapour mixture zone on the Ph-chart. So considering all these above changes, the existing relation of the frictional pressure drop share for its use in design calculation of capillary tube, was collected from different references such as a journal by Ali Hussein 2008[13], Ramgopal. M. NPTEL tutorial on refrigeration & air conditioning, lesson 24[14] and other reference books as explained hereunder.

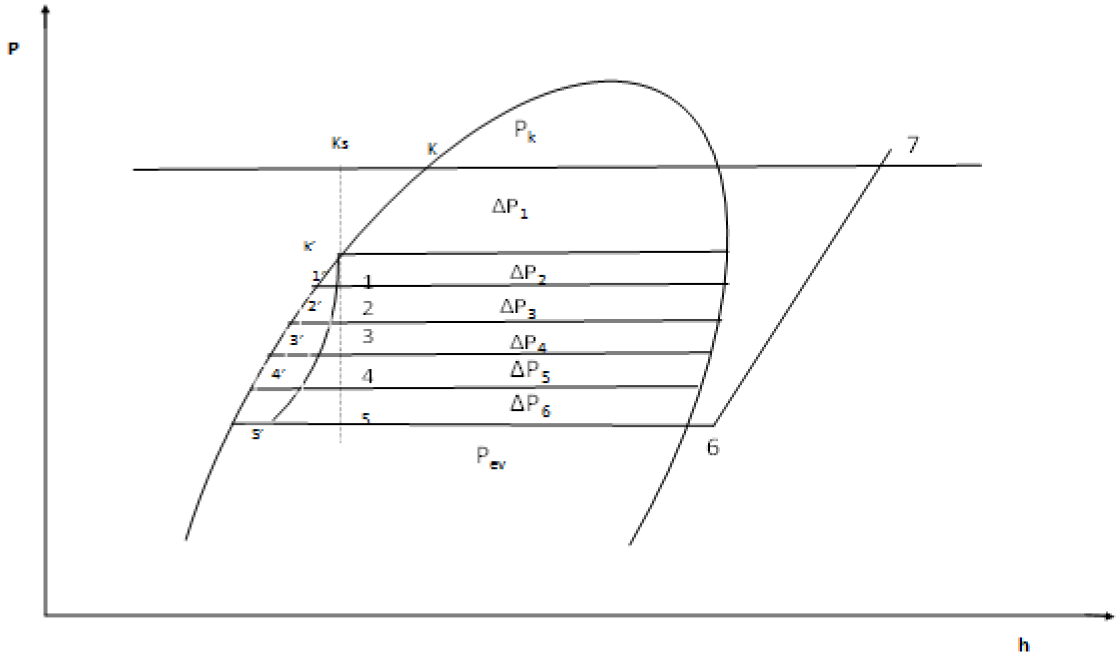

Fig-2. Ph Chart for Pressure & enthalpy changes of elementary lengths of the capillary along Isenthalpic and Fanno line considering sub cooling of condensate by SLHX.

III. EXISTING DESIGN RELATIONS FOR CAPILLARY TUBE

1-First the expansion of the liquid refrigerant is assumed isenthalpic along k_s- k'-1-2-3-4-5 on Ph- chart and the momentum gain is due to the pressure drop of the enthalpy. Here k_s is the point of sub cooling of saturated condensate from the outlet point k of the condenser. So at any i^{th} node enthalpy = $h_i = h_{k'}$

2-At this node quality of the refrigerant $\mathbf{x_i = (h_{k'} - h_{fi})/h_{gf}}$ -(1)

Where h_{fi} = saturation liquid enthalpy, h_{fg} = Enthalpy of vaporization.

3-Specific volume of the refrigerant at this node, $\mathbf{v_i = v_{fi} + x_i (v_{gi} - v_{fi})}$ -(2)

4-Mass flow rate of refrigerant in the capillary, $\mathbf{m = Q_r /(h_{g5} - h_5)}$ - (3)

Where Q_r = refrigeration effect in kW .

5-Mass flow flux : $\mathbf{G = 4\,m/(\pi\ d^2)}$ -(4)

6- Flow velocity of the refrigerant : $\mathbf{u_i = G \times v_i}$ -(5)

7-Since the actual expansion is not isenthalpic, the refrigerant gains momentum or kinetic energy through enthalpy drop along the Fanno line, so the actual enthalpy at any node on the Fanno line is found from:

$\mathbf{h_{i'} = h_k - (u_i^2 - u_k^2)/2000}$ **kj/kg** -
(6)

8- Now the new properties, refrigeration capacity and refrigerant mass flow rate all determined by the above relations will change from isenthalpic condition.

So the new mass flow rate: $\mathbf{m = Q_r /(h_{g5} - h_5')}$ -- (7)

9-This causes the variation of actual mass flow flux and flow velocity at different nodes of the capillary that is determined through many iterations using the same relations as above.

10-The single phase pressure drop analysis in capillary based on the Bernoulli's energy balance equation for incompressible flow through a pipe or tube gives

$\mathbf{P_1/\rho g + u_1^2/2g = P_2/\rho g + u_2^2/2g + h_f}$ -- -(8)

Where h_f = Frictional head loss. Rearranging this equation, we have

$\mathbf{(P_1 - p_2) = \rho (u_1^2 - u_2^2)/2 + h_f \rho g}$ -- (9)

Denoting $(P_1 - p_2) = \Delta P$ = Actual pressure drop, $h_f \rho g = \Delta P_f$ = Frictional pressure drop

$\mathbf{\rho (u_1^2 - u_2^2)/2 = \Delta P_m}$ **= Momentum gain pressure drop.** ---------------------------------- (10)

Now $\mathbf{\Delta P = \Delta P_m + \Delta P_f}$ **,** **So,** $\mathbf{\Delta P_f = \Delta P - \Delta P_m}$ ------------------------------------ (11)

IV. DESIGN PROCEDURE

One important part of the capillary design is the determination of mass flow rate through a suitable available diameter of the capillary tube for some specific refrigeration effect and is determined through several iterations using the equation-7 and other related equations in the above. Here the procedure involves the FEA discritization principle using R-600a as refrigerant in 1 mm diameter capillary tube for 0.1 ton refrigeration capacity and is explained as following. Here the use of R-600a eliminates the toxicity, ozone layer depletion and high power requirement for compression because of the involvement of low saturation pressure between evaporator and condenser. The steps of design are as following.

1. The whole temperature and pressure drops from the capillary tube inlet to the outlet is divided into suitable elementary steps of intermediate nodes, such as 40^0, 30^0, 20^0, 10^0, 0^0,
-10^0 and -20^0c at their corresponding saturation pressures.

2. First the expansion is assumed to be isenthalpic with simultaneous frictional pressure drop, flash vaporisation, change of refrigerant properties and momentum gain without considering the internal mechanism. The expansion is shown by process line k_s-k'-1-2-3-4-5 on Ph- plot (Fig-2) and the saturation pressure, refrigerant properties like liquid enthalpy, phase change enthalpy and specific volume are noted from the property table corresponding to the nodal saturation temperatures as shown in the following Table-1.

Table -1. Shows the properties & flow velocity of refrigerant R-600a at different nodes of 1 mm capillary under isenthalpic expansion for 0.1 ton refrigeration capacity.

Sections (nodes)	t_i ^{0}c	P_i (bar)	h_{fi} (kj/kg)	h_{fg} (kj/kg)	dryness fraction (x_i)	V_i x10^3 (m^3/Kg)	U_i =G xV_i in m
k	40	5.361	272.37	Sub cooled	0	1.826	3.165
k'	30	4.080	272.37	325.59	0	1.826	3.165
1	20	3.043	247.50	336.65	0.0739	11.097	19.235
2	10	2.220	223.42	346.92	0.1411	25.751	44.636
3	0	1.578	200.00	356.59	0.2030	49.402	85.632
4	-10	1.090	177.09	365.87	0.2604	88.113	152.733
5	-20	0.728	154.51	374.95	0.3143	153.590	266.229

3. Here the mass flow flux, dryness fraction, specific volume and velocity at different nodes on the isenthalpic line are first determined using above relations of article IV and are found as in the above table-1.

Here $h_5 = h_{fk}$ = 272.37kj/kg. h_{g5} = 529.46 kj /kg. Q_r = 0.35 kW, d= 0.001m

So, m = 0.35/ (529.46 – 272.37) =0.0014 kg/s, G = (4 x 0.0013)/(π x d^2) =1733kg/m^2s

4. But the actual pressure drop is due to friction whereas the momentum gain is by consumption of own enthalpy. So the corresponding expansion leads to the enthalpy drop along the Fanno line k'-1'-2'-3'-4'-5'. Here the dryness fraction, specific volume, enthalpy and velocity of flow all become lower than isenthalpic values at different nodes.
5. Since the isenthalpic velocity is much higher than the actual Fanno line velocity at any pressure, so the initial Fanno line iteration enthalpies determined by eq. (6) become much lower than the actual values as in Table-2. This leads to much lowering of first iteration values for dryness fraction, specific volume and corresponding new velocity of flow than the actual Fanno line values at different saturation temperature nodes.

Table-2. First iteration refrigerant properties and flow velocity on Fanno line

Sections (nodes)	Fanno line Enthalpy($h_{i'}$)	Dryness fractions(xi)	Sp. Volume v_i x 10^3 m^3/kg	Flow velocity u_i =G x v_i
k	272.37	0	1.826	2.782
k'	272.37	0	1.826	2.782
1	272.19	0.0733	10.891	16.592
2	271.38	0.1382	25.016	38.110
3	268.71	0.1927	46.613	71.012
4	238.46	0.2198	77.173	117.568
5	236.94 2	0.2286	107.544	163.836

Here the mass flow rate = m= 0.35/ (529.46 – 236.94) = 0.001196 kg/s
Mass flow flux = 4 m / (π d ^{2}) =1523 kg/m^2-s

6. The subsequent iteration values of refrigerant properties, mass flow rate and flow velocities at different nodes of the capillary as determined from the same relations were found to oscillate and gradually approach to the actual values. The 9th iteration nearly approaches the actual property values on the Fanno line as in the table-3.

Here the actual mass flow rate of refrigerant = 0.35 / (529.46 –252.240) = 0.00126 kg/s
Mass flow flux = 4 m / (π d^2) = 1608 kg/m^2-s.
This mass flow rate is greater than the first iteration value, but less than the isenthalpic flow rate value.

Table-3. Actual property values and flow velocities on Fanno line nodes.

Sections (nodes)	Enthalpy hi(kj/kg)	Dryness Fraction (xi)	Sp. Volume v_i x 10^3 m^3/kg	u_i = G x v_i m/s
k_s	272.370	0	1.826	2.935
k'	272.370	0	1.826	2.935
1	272.225	0.0734	10.903	17.527
2	271.570	0.1388	25.117	40.376
3	269.475	0.1948	47.102	75.717
4	264.175	0.2380	80.277	129.046
5	252.240	0.2606	127.199	204.474

The following figure-3 shows the plot of quality variation of liquid vapour refrigerant mixture with saturation temperature in isenthalpic and actual Fanno line conditions

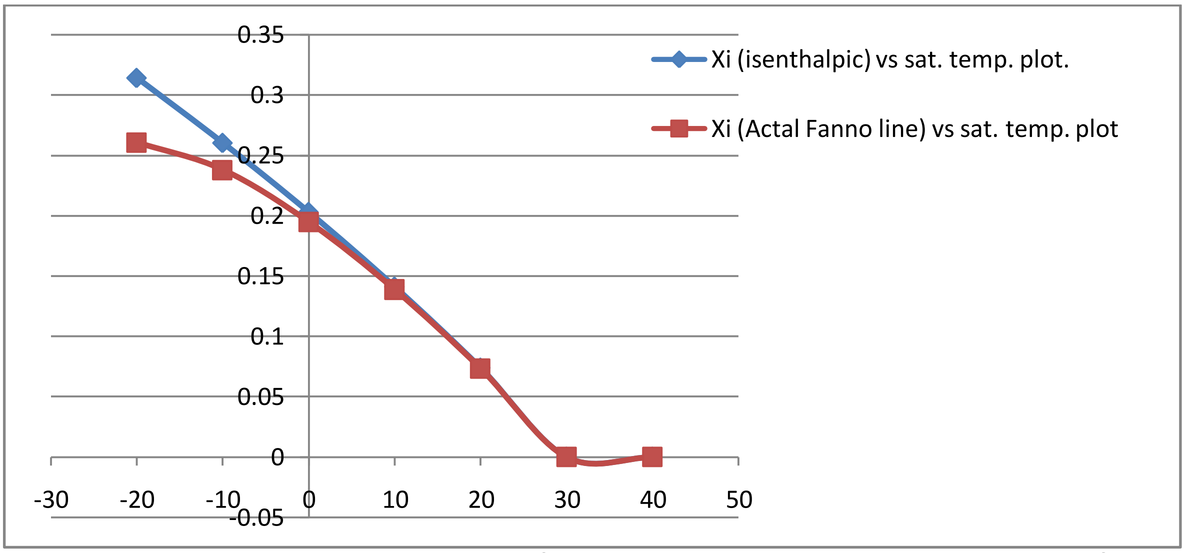

Fig-3. Isenthalpic and actual Fanno line refrigerant quality VS saturation temperature plot for refrigerant in the capillary.

Also the following plot shows the variation of velocity with saturation temperature in isenthalpic and actual Fanno line conditions.

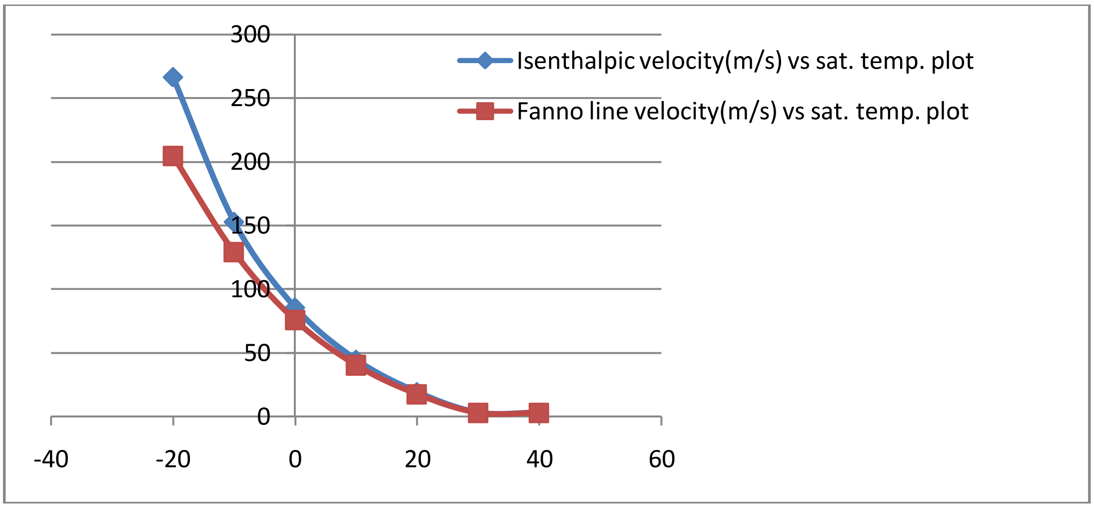

Fig-3. Isenthalpic and actual Fanno line velocity VS saturation temperature plot for refrigerant in the capillary.

7. Next the momentum gain pressure drop at different nodes on the Fanno line, have been determined based on the actual mass flow rate and velocity using equation (10). After few nodes these elementary pressure drops are found to be greater than the actual pressure drops and introduces the controversy against the relation (11) as shown in the following table-4.

Table-4. For comparison of ΔP_i and ΔP_m with properties on Fanno line.

Nodes	Pi (bar)	ΔPi	U_i	ΔUi	ΔP_m=G x ΔUi
k'	4.080		2.935		
1	3.043	1.037	17.527	14.592	0.2346
2	2.220	0.823	40.376	22.849	0.3673
3	1.578	0.642	75.717	35.341	0.5681
4	1.090	0.488	129.046	43.329	0.6965
5	0.728	0.362	204.474	75.428	1.2125

Here ΔP_m the elementary momentum gain pressure drop is much higher than the elementary actual pressure drop ΔP_i at the corresponding nodes after a few distance of flow.

8. This result highlights the much higher values of momentum gain pressure drop after few nodes and confirms the controversy of sharing of actual pressure drop in the momentum gain. This controversy has been clearly explained and eliminated in my own journal paper [1], which concludes that the actual pressure drop is shared in overcoming the fluid friction and potential energy gain but not in momentum gain through phase change due to flash vaporisation. Again neglecting the potential energy gain the actual total pressure drop is nearly equals to the frictional pressure drop.

$$\Delta P_{fi} = \Delta P_i \quad \text{--} \quad (12)$$

From this frictional pressure drop the required design length of the capillary should be calculated.

The following plot shows the variation of actual pressure during expansion along the capillary tube with saturation temperature. Also it shows the variation of elementary frictional pressure drops between successive temperature nodes considering with &without momentum gain. Here it is obvious that the elementary frictional pressure drop considering no share of momentum gain pressure drop gradually decreases towards the lower temperature nodes, but never become negative. However considering momentum gain pressure drop share, the remaining frictional pressure drop share according to the relation (11) decreases rapidly and soon becomes negative, which means no part of the actual pressure drop is available to overcome friction towards the lower temperature nodes and contradicts the possibility of flow that practically does not happen. Again as no momentum gain occurs during liquid phase expansion between first two nodes, the frictional pressure drop in both cases coincide & increase linearly.

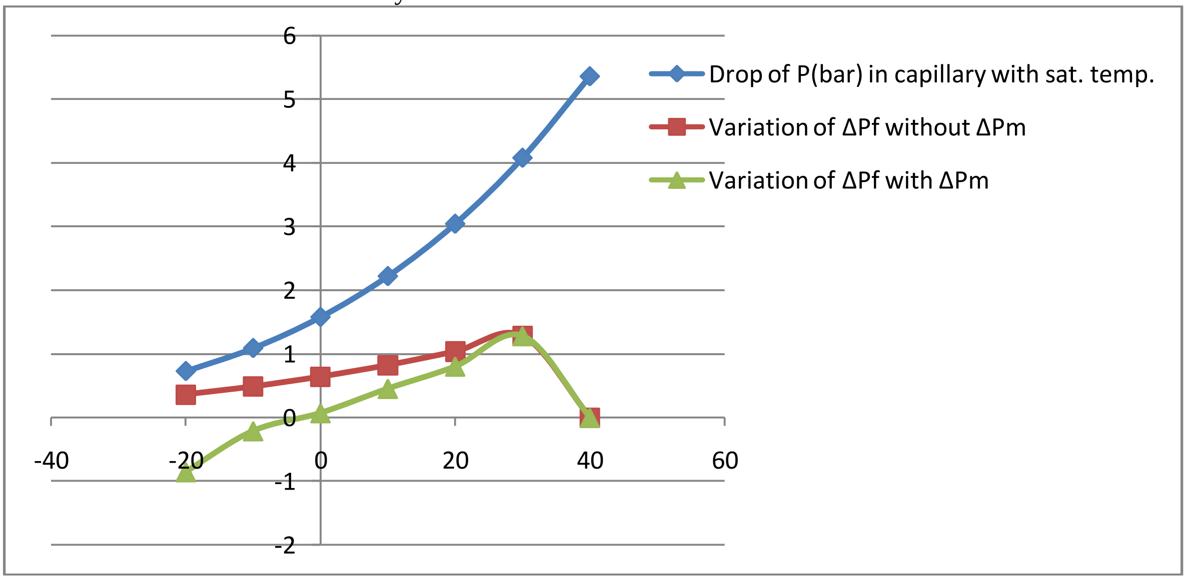

Fig-4. Actual pressure (bar) variation & elementary frictional pressure drop considering with and without momentum gain vs saturation temperature (^{0}c) drop plots.

VI. CALCULATION OF FRICTION FACTOR

The analysis of frictional pressure drop of the refrigerant flowing through the pipe or capillary tube incorporates friction factor "f" that can be determined based on several theoretical and experimental relations. For capillary tube design of refrigerators a standard relation frequently used is the Blasius (1911) [18] relation as following.

$$f = \frac{0.32}{\mathrm{Re}^{0.25}}, \quad where \quad \mathrm{Re} = \frac{\rho dU}{\mu} = \frac{dG}{\mu} \quad \text{------------------------------------} \quad \mathbf{(13)}$$

At different nodes the refrigerant properties vary in the vapour-liquid mixture and the coefficient of dynamic viscosity is determined according to the relation

$$\boldsymbol{\mu_i = \mu_{fi} + x_i (\mu_{gi} - \mu_{fi}).} \quad \text{--} \quad \mathbf{(14)}$$

The values of the friction factor at different nodes for R-600a in 1 mm capillary is determined as in the following table- 5.

Table-5. For calculated values of friction factor at different nodes.

Nodes	Dryness Fraction (xi)	Liquid viscosity μ_{fi} x 10^6 Pa-s	Vapor viscosity μ_{gi} x 10^6 Pa-s	μ_i x 10^6 Pa-s	R_{ei}	f_i
k	0	148.8 (at mean 35 ^{0}c)	Sub cooled	148.8	10806	0.0314
k'	0	158.5	76.12	158.50	10145	0.0319
1	0.0734	165.0	74.90	161.73	9942	0.0320
2	0.1388	178.0	72.42	167.23	9615	0.0323
3	0.1948	202.7	69.92	181.96	8837	0.0330
4	0.2380	227.3	67.39	196.47	8184	0.0336
5	0.2606	252.0	64.85	213.82	7520	0.344

VII. DETERMINATION OF CAPILLARY TUBE LENGTH .

Actually the wall friction of the capillary tube causes the gradual pressure drop of the refrigerant flow which results in the drop of corresponding saturation temperature, internal energy and enthalpy but increase of specific volume and kinetic energy with adiabatic flash vaporization using its own enthalpy. Again the increase of specific volume with equivalent pressure drop keeps the product term (Pv) of the enthalpy nearly constant as explained in [1]. Also the actual pressure drop has no share contribution towards the gain of kinetic energy through phase change and hence the momentum gain pressure drop share (ΔP_m) in the equation (11) is zero. So here the actual total pressure drop (ΔP) is nearly equal to the frictional pressure drop, ΔP_F.

Now taking $\Delta P_F = \Delta P_{i,\ actual}$ for every elementary length of the capillary tube between two adjacent temperature nodes, we have.

$\Delta P_F = (\rho\ f\ \Delta L\ u^2)/2d$

Hence $\mathbf{\Delta L = 2d\ \ \Delta P_F/(\rho\ \ f\ \ u^2)}$ -- **(15)**

Here the determination of the 1mm diameter capillary tube length from the frictional pressure drop (ΔP_F) equal to the actual pressure drop eliminates the controversy of being negative near the lower saturation nodes and introduces greater accuracy with a small increase in tube length and allow the desired low evaporator temperature maintenance. So accordingly the elementary lengths between different adjacent nodes and hence the total length of the capillary tube are determined as in the following table-6.

Table-6. Calculation of elementary lengths at different nodes & hence the total length

Nodes	Pi (bar)	u_i, m/s	fi	ΔP_{Fi}	$u_m = (\frac{u_{i+1}+u_i}{2})$	ΔL in m
k s	5.361	2.935	0.0314			
k'	4.080	2.935	0.0319	1.281	2.935	1.716
1	3.043	17.527	0.0320	1.037	10.231	0.395
2	2.220	40.376	0.0323	0.823	28.952	0.110
3	1.578	75.717	0.0330	0.642	58.047	0.042
4	1.090	129.046	0.0336	0.488	99.382	0 .019
5	0.728	204.474	0.0344	0.362	166.760	0.008
					Total length , L=	**2.290**

The above calculation gives the required capillary tube length for the expansion of the refrigerant R-600a from 30^0c sub-cooled state at condenser saturation temperature of 40^0c to the evaporator saturation temperature of -20^0c with comparatively greater accuracy.

VIII. CONCLUSION

The analysis of this paper summarizes into the following conclusions.

1. The actual Fanno line properties determined from the iterative calculations on the initial isenthalpic properties and the corresponding mass flow rate of the refrigerant for a given refrigeration effect are considered more accurate.
2. This paper determines the elementary momentum gain pressure drops for the flow of a refrigerant through an adiabatic capillary tube of a domestic refrigerator from the actual Fanno line properties based on existing relations of text books and other journal papers and high lights the value to exceed the actual elementary pressure drop values after a small distance of flow from the condenser towards the evaporator.
3. The paper confirms that the actual pressure drop is not shared in momentum gain which is actually due to the consumption of internal energy part of the enthalpy with simultaneous phase change through flash vaporization resulting in increase of specific volume and velocity.
4. Here it claims that the determination of design length of a given diameter capillary tube from the actual pressure drop caused by the frictional drag as more accurate and permits a small increase of length to maintain the desired low temperature in the evaporator.
5. Here also this paper determines the design length of 1mm capillary tube as an example for 0.1 ton refrigeration capacity domestic refrigerator.

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