[Kulmitra* *et al.*, 5(7): July, 2016] ISSN: 2277-9655
IC™ Value: 3.00 Impact Factor: 4.116

IJESRT
INTERNATIONAL JOURNAL OF ENGINEERING SCIENCES & RESEARCH TECHNOLOGY

NUMERICAL INVESTIGATION OF CAPILLARY TUBE BY REPLACING THE INSIDE REFRIGERANT AND DIAMETER

Abhinav Kulmitra *, Ajay Kumar Verma
* PG Students, Mechanical Department, Shri Shankarachrya Group of College, Bhilai, INDIA
Associate Professor, Mechanical Department, Shri Shankarachrya Group of College, Bhilai, INDIA

DOI: 10.5281/zenodo.56913

ABSTRACT

The capillary tube used in the mostly in the refrigerant flow control devices. Hence performance of the capillary tube is best for good refrigerant flow. The many researchers had been concluding performance using experimentally, theoretically and analysis based. In this present work analyze the flow analysis of the refrigerant inside a capillary tube for adiabatic flow conditions. The proposed model can predict flow characteristics in adiabatic capillary tubes for a given mass flow rate. In the present work R-22 is replaced by Ammonia refrigerant has been used as a working fluid inside the capillary tube and the capillary tube design is changed straight to coiled capillary, which taken from good literature. The analysis is done in ANSYS CFX 16.2 software. It is observed from the results dryness fraction by using the helical capillary tube (Ammonia refrigerant flow) is better than straight and existing helical capillary tube (R22 refrigerant flow). The best suitable helical coiled design is suggested.

KEYWORDS: Capillary tube- straight and helical, R22 and Ammonia refrigerant, ANSYS.

INTRODUCTION

An expansion device is another basic component of a refrigeration system. The basic functions of an expansion device used in refrigeration systems are to:

1. Reduce pressure from condenser pressure to evaporator pressure, and
2. Regulate the refrigerant flow from the high-pressure liquid line into the evaporator at a rate equal to the evaporation rate in the evaporator

Under ideal conditions, the mass flow rate of refrigerant in the system should be proportional to the cooling load. Sometimes, the product to be cooled is such that a constant evaporator temperature has to be maintained. In other cases, it is desirable that liquid refrigerant should not enter the compressor. In such a case, the mass flow rate has to be controlled in such a manner that only superheated vapour leaves the evaporator. Again, an ideal refrigeration system should have the facility to control it in such a way that the energy requirement is minimum and the required criterion of temperature and cooling load are satisfied. Some additional controls to control the capacity of compressor and the space temperature may be required in addition, so as to minimize the energy consumption.

The expansion devices used in refrigeration systems can be divided into fixed opening type or variable opening type. As the name implies, in fixed opening type the flow area remains fixed, while in variable opening type the flow area changes with changing mass flow rates. There are basically seven types of refrigerant expansion devices. These are:

1. Hand (manual) expansion valves
2. Capillary Tubes
3. Orifice
4. Constant pressure or Automatic Expansion Valve (AEV)
5. Thermostatic Expansion Valve (TEV)
6. Float type Expansion Valve
   a) High Side Float Valve
   b) Low Side Float Valve
7. Electronic Expansion Valve

Of the above seven types, Capillary tube and orifice belong to the fixed opening type, while the rest belong to the variable opening type. Of the above seven types, the hand operated expansion valve is not used when an automatic control is required. The orifice type expansion is used only in some special applications.

1.1 CAPILLARY TUBE

A capillary tube is a long, narrow tube of constant diameter. The word “capillary” is a misnomer since surface tension is not important in refrigeration application of capillary tubes. Typical tube diameters of refrigerant capillary tubes range from 0.5 mm to 3 mm and the length ranges from 1.0 m to 6 m.

The pressure reduction in a capillary tube occurs due to the following two factors:

1. The refrigerant has to overcome the frictional resistance offered by tube walls. This leads to some pressure drop, and
2. The liquid refrigerant flashes (evaporates) into mixture of liquid and vapour as its pressure reduces. The density of vapour is less than that of the liquid. Hence, the average density of refrigerant decreases as it flows in the tube. The mass flow rate and tube diameter (hence area) being constant, the velocity of refrigerant increases since $\dot{m} = \rho VA$. The increase in velocity or acceleration of the refrigerant also requires pressure drop.

Several combinations of length and bore are available for the same mass flow rate and pressure drop. However, once a capillary tube of some diameter and length has been installed in a refrigeration system, the mass flow rate through it will vary in such a manner that the total pressure drop through it matches with the pressure difference between condenser and the evaporator. Its mass flow rate is totally dependent upon the pressure difference across it; it cannot adjust itself to variation of load effectively.

1.1.1 Balance Point of Compressor and Capillary Tube

The compressor and the capillary tube, under steady state must arrive at some suction and discharge pressures, which allows the same mass flow rate through the compressor and the capillary tube. This state is called the balance point. Condenser and evaporator pressures are saturation pressures at corresponding condenser and evaporator temperatures. Figure 24.1 shows the variation of mass flow rate with evaporator pressure through the compressor and the capillary tube for three values of condenser temperatures namely, 30, 40 and 50°C.

The mass flow rate through the compressor decreases if the pressure ratio increases since the volumetric efficiency of the compressor decreases with the increase of pressure ratio. The pressure ratio increases when either the evaporator pressure decreases or the condenser pressure increases. Hence, the mass flow rate through the compressor decreases with increase in condenser pressure and/or with decrease in evaporator pressure.

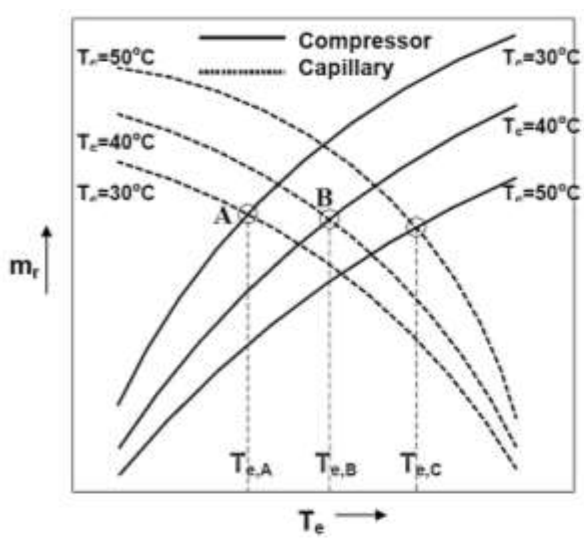


Fig 1.1: Variation of refrigerant mass flow rate through compressor and capillary tube with evaporator and condenser temperature (A, B & C are the balance points)

The pressure difference across the capillary tube is the driving force for the refrigerant to flow through it, hence mass flow rate through the capillary tube increases with increase in pressure difference across it. Thus the mass flow rate through the capillary tube increases as the condenser pressure increases and/or the evaporator pressure decreases. The variation of mass flow rate through capillary tube is shown for three condenser temperatures, namely, 30, 40 and 50℃ in Figure 1.1. This is the opposite of the effect of pressures on the compressor mass flow rate. Hence, for a given value of condenser pressure, there is a definite value of evaporator pressure at which the mass flow rates through the compressor and the evaporator are the same. This pressure is the balance point that the system will acquire in steady state. Hence, for a given condenser temperature, there is a definite value of evaporator temperature at which the balance point will occur. Figure 1.1 shows a set of three balance points A, B and C for the three condenser temperatures. These balance points occur at evaporator temperatures of $T_{e,A}$, $T_{e,B}$, and $T_{e,C}$. It is observed that the evaporator temperature at balance point increases with increase of condenser temperature.

1.2 REFRIGERANT

A refrigerant is a substance or mixture, usually a fluid, used in a heat pump and refrigeration cycle. In most cycles it undergoes phase transitions from a liquid to a gas and back again. Many working fluids have been used for such purposes. Fluorocarbons, especially chlorofluorocarbons, became commonplace in the 20th century, but they are being phased out because of their ozone depletion effects. Other common refrigerants used in various applications are ammonia, sulfur dioxide, and non-halogenated hydrocarbons such as propane.

The ideal refrigerant would have favorable thermodynamic properties, be noncorrosive to mechanical components, and be safe, including free from toxicity and flammability. It would not cause ozone depletion or climate change. Since different fluids have the desired traits in different degree, choice is a matter of trade-off.

The desired thermodynamic properties are a boiling point somewhat below the target temperature, a high heat of vaporization, a moderate density in liquid form, a relatively high density in gaseous form, and a high critical temperature. Since boiling point and gas density are affected by pressure, refrigerants may be made more suitable for a particular application by choice of operating pressures.

1.2.1 Types of refrigerant

The most common types of refrigerants in use nowadays are presented below:

a) halocarbons or freons.
b) azeotropic refrigerants.
c) zeotropic refrigerants.
d) inorganic refrigerants like carbon dioxide, ammonia, water and air.
e) hydrocarbon refrigerants.

LITERATURE REVIEW

The early work of Mikol (1963) [1] showed that the vapor generation in the capillary tube not necessary to occur when the liquid approaches its bubble point. But rather the refrigerant remains in the liquid phase for further length of the capillary tube where the pressure is below the saturation value. The same trend of behavior for this situation was later observed by Li et al. (1990) [2].

The experimental work of Wei et al. (2001) [3] conducted for small refrigeration units has reported the length required for these units. The length was ranged between (0. 4) to (2.5) mm and as a result, the capillary tubes are normally folded so as to reduce the required space.

An extensive data for the adiabatic capillary tubes and reliable diagram are available in the work of Bittle et al. (1998) [4], ASHRAE (1994) [5], Melo et al. (1999) [6], Sami and Maltais (2000) [7]. A qualitative data is also available from the work of Sami and co-workers in [8, 9].

A number of theoretical models, numerical and rating charts for the characteristics of the capillary tube design and selection are published. The early work conducted by Hopkins (1950) [10] and Whitesel (1957) [11] resulted in developing a rating charts for (R-12) and (R-22). ASHRAE (1979) [12] established graphical method representation for the capillary tube rating for specified entering conditions. More detailed rating charts are also included in ASHRAE (1998) [13] 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) [14] was developed for the prediction of the refrigerant mass flow rate through the capillary tube. It is based mainly on the Buckingham-pi theorem incorporating the physical factors and fluid properties that affect the capillary flow rate. Their correlation showed a good agreement with their own and other investigators experimental data for the considered refrigerants.

METHEMATICAL MODELLING

3.1 GOVERNING EQUATIONS

The fundamental equations governing capillary tube flow are the mass, momentum and energy conservation equations. In a 1D mesh, these equations can be integrated and solved simultaneously by an iterative process for each control volume of length Δz like the shown in fig 4.1.

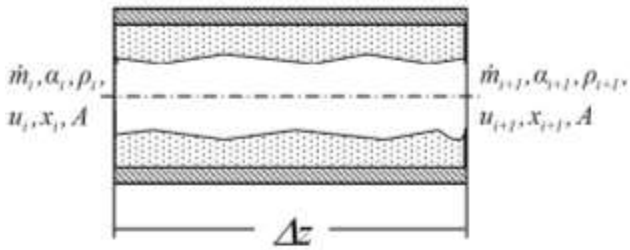


Fig 4.1: Control volume

The process can be repeated along the capillary tube length including single-phase and two-phase cells. For the two phase flow region a separated flow model is assumed.

The model was developed with the following considerations:

- The capillary tube is horizontal (gravity effects are neglected), and of constant cross section.
- The flow in the capillary tube is steady, one-dimensional, and adiabatic.
- When the fluid reaches the saturation pressure, the fluid starts to evaporate.
- The fluid is always in local thermodynamic equilibrium corresponding to its local pressure.
  Pressure losses can be defined by:
- Pressure drop by entrance effects (from the upstream tube to the capillary tube)
- Pressure drop by friction: Single-phase friction from entrance up to the saturation point, and two phase friction from saturation point up to the end.
- There is no recovery of the pressure in the enlargement from the outlet to the downstream tube, where the evaporation pressure is encountered.

3.1.1 Conservation equations

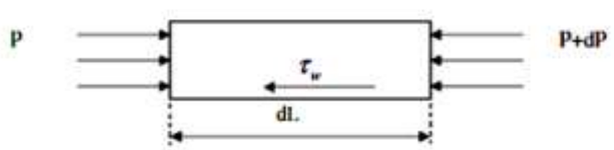


Fig 4.2: Forces acting on the fluid element

Mass Balance: Application of continuity equation results into the following:

$$m = \rho A V \text{ or } G = \frac{m}{A} = \rho V$$

Momentum Balance: On applying the principle of momentum conservation or the Second law of Thermodynamics, the following equation will result:

$$P.A - (P + dP).A - \tau_w(\pi d)dL = mdV$$

$$-dP = \frac{f}{2d}\rho V^2 dL + \rho V dV$$

Taking log both sides of above equation and then differentiating and simplifying

$$-\frac{dV}{V} = \frac{d\rho}{\rho}$$

Hence, $dL = \frac{2d}{f}\left(\frac{\rho dP}{G^2} - \frac{d\rho}{\rho}\right)$

Energy Balance: On applying the steady flow energy equation on the element to get,

$$\delta q - \delta w = dh + VdV + gdZ$$

3.1.2 Single-phase region

In the single phase liquid region, the refrigerant density is almost constant as the liquids are practically incompressible (ρ = constant) and with tube cross sectional area being constant, from mass balance represented by Equation, the velocity is constant. Integrating Equation, the length of single-phase liquid region expressed in equations.

$$L_{sp} = \frac{d}{f_{sp}}\left(\frac{2}{\rho V^2}(\mathrm{p}_2 - \mathrm{p}_3)\right) = \frac{2d\rho(\mathrm{p}_2 - \mathrm{p}_3)}{fG^2}$$

Pressure loss due to entrance effects

$$\mathrm{p}_1 - \mathrm{p}_2 = k\frac{\rho V^2}{2}$$

Where k is the entrance loss coefficient, taken as 1.5 From equations.

$$L_{sp} = \frac{d}{f_{sp}}\left(\frac{2}{G^2}(\mathrm{p}_2 - \mathrm{p}_3) - k\right)$$

Where „f_{sp}‟ is the single phase friction factor.

METHODOLOGY

For setting up any CFX (Central Florida Expressway) problem, the geometry has to be modelled with required details, mesh has to be generated optimally to obtain the results correctly and flow parameters and boundary conditions are to be set up for solving the problem. The discretized domain is solved using solver and results are analyzed in post processor. In the present investigation, Creo parametric (for modelling) and ANSYS-CFX (for analyzing) software is used for the geometry modelling, mesh generation and solutions. ANSYS CFX software issued for defining boundary conditions, solving and post processing.

4.1 COMPUTAIONAL FLUID DYNAMICS (CFD) ANALYSIS

4.1.1 Modelling

The body about which flow is to be analyzed requires modeling. This generally involves modeling the geometry with a CAD software package.

The various geometries are following and properties of capillary tube [18] as shown in table 4.1 and 4.2

Table 4.1 helical coiled capillary tubes models [18]

Models	No. of turns (N)	Pitch of the coil (p) (mm)	Diameter of the coil (D) (mm)	Length of the coil (L) (mm)
1	5	3	48.5	761.98
2	10	3	24.2364	761.99
3	30	3	8.0248	762
4	40	3	5.988	761.98

Diameter of tube d=1.27mm [18]
Dimaeter of tube d= 1.11mm (For present analysis)

Table 4.2 Properties of Experimental and computational for the straight capillary [18]

Properties	Experimental	Computational
Inlet temperature (Tin)	52 °C	51.3 °C
Outlet temperature($Tout$)	8 °C	9.1 °C
Inlet pressure (Pin)	20.328 bar	20.07 bar
Outlet pressure ($Pout$)	6.406 bar	6.629 bar
Inlet mass fraction liquid	1	1
Outlet mass fraction of liquid	0.7205	0.719
Inlet mass fraction of vapour	0	0
Outlet mass fraction of vapour	0.2795	0.2806

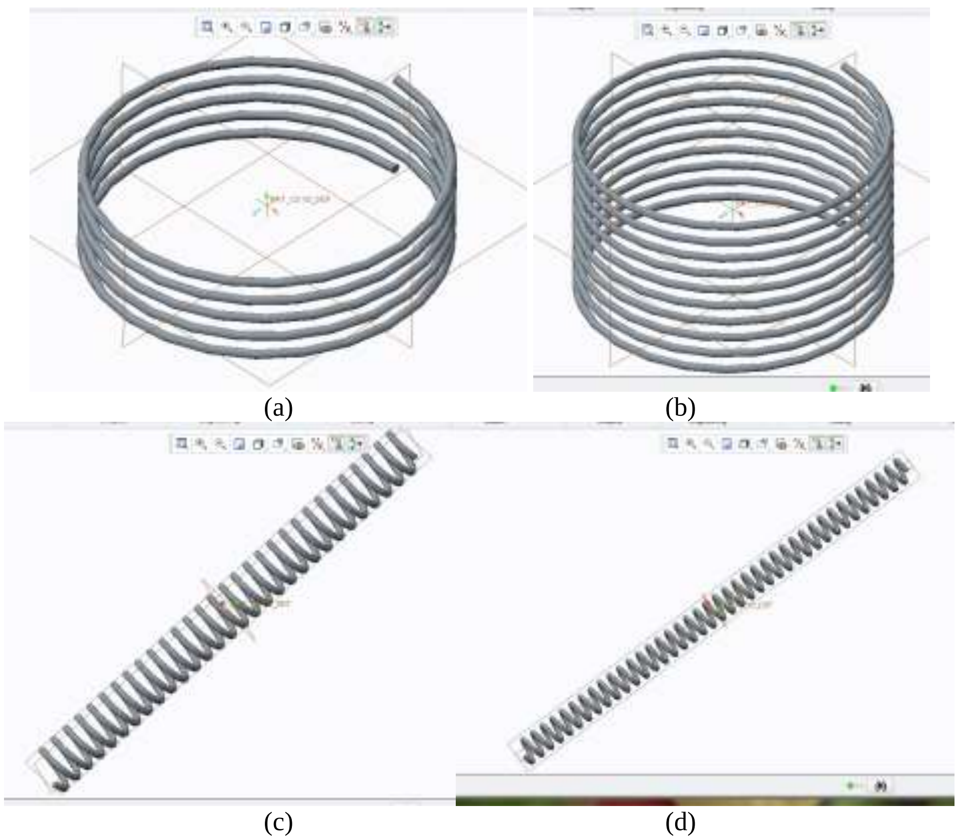
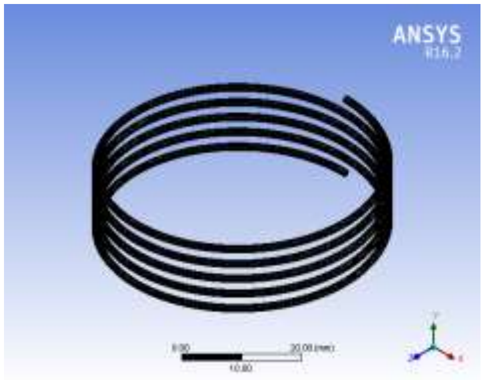
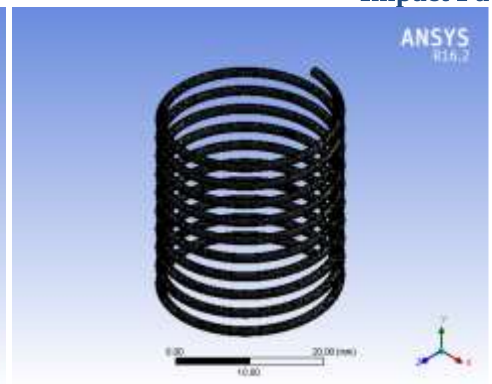


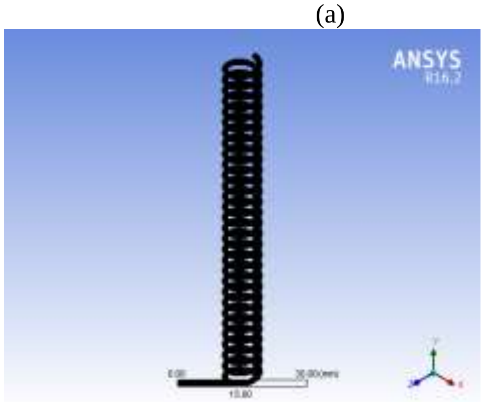
Fig 4.1: (a)(b(c)(d) 3D view of capillary tube (When n=5, 10, 30 and 40 and Dia d=1.17mm)

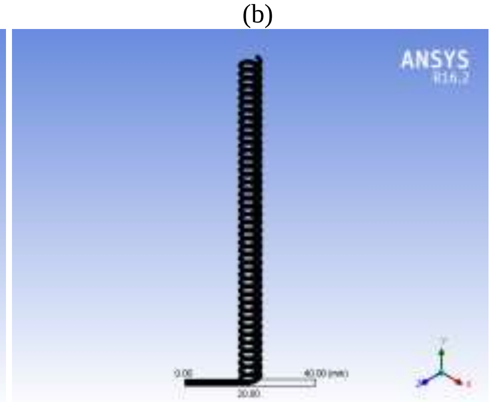
4.3.2 Generate Mesh

After modelling the product in Creo import this file to ANSYS CFX and generate mesh in model. In this process can uses unstructured meshing method as shown in Figure 4.2. ANSYS CFX uses unstructured meshes in order to reduce the amount of time you spend generating meshes, to simplify the geometry modeling and mesh generation process, to allow modeling of more complex geometries than you can handle with conventional, multi-block structured meshes, and to let you adapt the mesh to resolve the flow-field features.



(a)


(b)


(c)


(d)

Fig 4.2: (a)(b(c)(d) Meshing view of capillary tube (When n=5, 10, 30 and 40 resp.)

4.3.3 Boundary Condition

Since a finite flow domain is specified, physical conditions are required on the boundaries of the flow domain. The simulation generally starts from an initial solution and uses an iterative method to reach a final flow field solution. The boundary name is defined like; Inlet, Outlet and Capillary wall (Cap_Wall).

4.3.4 Establish the Simulation Strategy

The strategy for performing the simulation involves determining such things as the use of space-marching or time-marching, the choice of turbulence or chemistry model, and the choice of algorithms.

4.3.5 Perform the Simulation

The simulation is performed with various possible with options for interactive or batch processing and distributed processing.

4.3.6 Simulation result

Post-Processing involves extracting the desired flow properties (thrust, lift, drag, etc...) from the computed flow field.

RESULT AND DISCUSSION

ANSYS CFX is a high-performance computational fluid dynamics (CFD) software tool that gives reliable and accurate solutions and results quickly and robustly across a wide range of CFD/CFX and multi-physics applications.

Table 5.1 From literature [18], Experimental results of straight capillary tube and Computational results of Helical coiled capillary tube (R22 refrigerant)

Properties	Exp.	Case 1	Case 2	Case 3	Case 4
Inlet temp. (T_{in}), °C	52	52	46	49	52
Outlet temp. (T_{out}) °C	8	8	8	7	7. 3

Inlet pre. (P_{in}), bar	20.32	17.6	17.53	19.0	20.28
Outlet pre. (P_{out}), bar	6.406	6.406	6.37	6.181	6.2112
Inlet mass fraction of liquid	1	1	1	1	1
Outlet mass fraction of liquid	0.72	0.713	0.712	0.70	0.708
Inlet mass fraction of vapour	0	0	0	0	0
Outlet mass fraction of vapour	0.279	0.286	0.287	0.29	0.291

Case-1, When n=5, Case-2, When n=10, Case-3, When n=30, Case-4, When n=40

Table 5.2 Simulation results (present) obtained from ANSYS CFX Software (Ammonia refrigerant)

Properties	**Case 1 (n=5)**	**Case 1 (n=10)**	**Case 1 (n=30)**	**Case 1 (n=30)**
Inlet temp. (T_{in}), °C	52	52	52	52
Outlet temp. (T_{out}) °C	7.8	7.7	7.6	7.1
Inlet pre. (P_{in}), bar	20.06	19.98	19.43	20.31
Outlet pre. (P_{out}), bar	6.352	6.368	6.077	6.065
Inlet mass fraction of liquid	1	1	1	1
Outlet mass fraction of liquid	0.7012	0.7080	0.7001	0.6998
Inlet mass fraction of vapour	0	0	0	0
Outlet mass fraction of vapour	0.289	0.296	0.297	0.301

Table 5.3 Validation of Temperature and pressure results (present and literature)

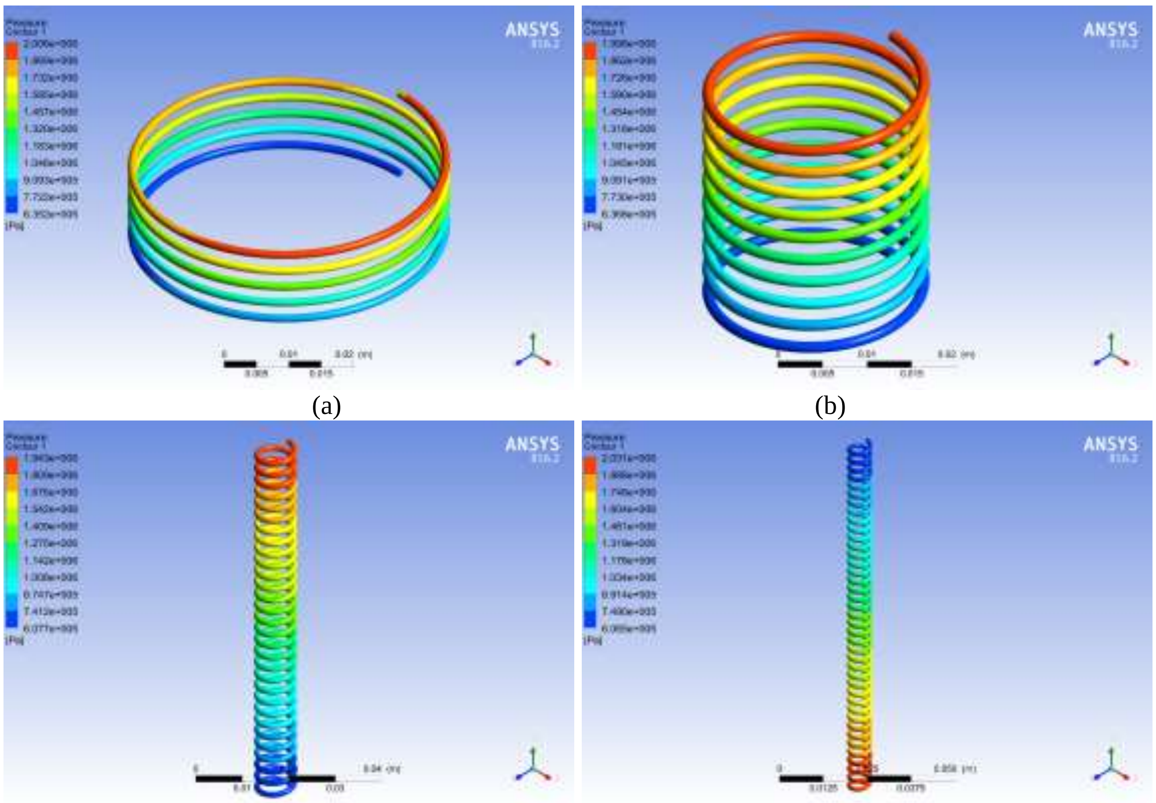
S. No.	**Inlet temp. (T_{in}), °C**	**Outlet temp. (T_{out}) °C**	**Inlet pre. (P_{in}), bar**	**Outlet pre. (P_{out}), bar**	**Pressure Drop (Δp)**
Exp. [18]	52	8	20.32	6.406	13.914
n=5 [18]	52	8	17.6	6.406	11.194
n=10 [18]	46	8	17.33	6.37	10.96
n=30 [18]	49	7	19.0	6.181	12.819
n=40 [18]	52	7.3	20.28	6.2112	14.0688
n= 5 (Present)	52	7.8	20.06	6.352	13.708
n=10 (Present)	52	7.7	19.98	6.368	13.612
n=30 (Present)	52	7.6	19.43	6.077	13.353
n=40 (Present)	52	7.1	20.31	6.065	13.945

Table 5.4 Validation of Mass fraction of liquid and vapour results (present and literature)

S. No.	**Inlet mass fraction of liquid**	**Outlet mass fraction of liquid**	**Inlet mass fraction of vapour**	**Outlet mass fraction of vapour**
Exp. [18]	1	0.72	0	0.279
n=5 [18]	1	0.713	0	0.286
n=10 [18]	1	0.712	0	0.287
n=30 [18]	1	0.70	0	0.29
n=40 [18]	1	0.708	0	0.291
n= 5 (Present)	1	0.7012	0	0.289
n=10 (Present)	1	0.7080	0	0.296

n=30 (Present)	1	0.7001	0	0.297
n=40 (Present)	1	0.6998	0	0.301

The various results obtained in this investigation are

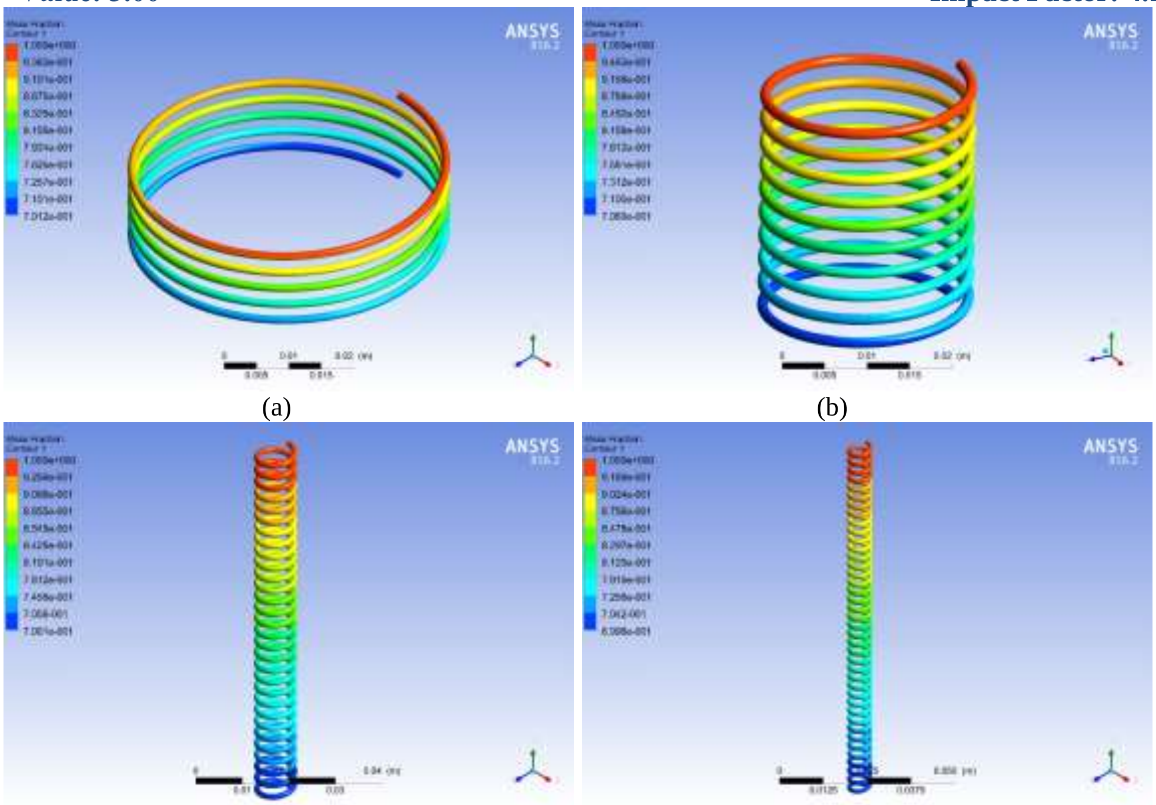


(a) (b) (c) (d)

Fig 5.1: Pressure contours of helical coiled capillary tube.
(a) When n=5, (b) When n=10, (c) When n=30, (d) When n=40

As show in the figure 6.1 a-d, the helical capillary tube by replacing straight capillary tube. The pressure contours are observed form the above figure 6.1 a-d. The main function of the capillary tube is to decrease the pressure of the capillary tube so the pressure has been decreased. In literature, the pressure drop from capillary tube is 11.194, 10.96, 12.819 and 14.0681 bar, when number of turns in 5, 10, 30 and 40 respectively, and the Experimental calculations are from 20.328 bar to be decreased to 6.406 bar (13.914 bar). But in present investigation which concluded that the pressure drop from capillary tube is 13.708, 13.612, 13.353 and 13.945 bar when number of turns in 5, 10, 30 and 40 respectively.

The pressure drop difference between literature and present work is 2.514, 2.652, 0.534 and 0.1231. In the above results, the number of turn 40 is suitable for helical capillary tube. Hence the n=40 is suggested.



(a) (b)

(c) (d)

Fig 5.2: Mass fraction contour of liquid for helical coiled capillary tube.
(a) When n=5, (b) When n=10, (c) When n=30, (d) When n=40

The liquid Mass fraction is calculated from ANSYS CFX as shown figure 6.2 a-d. The main function of capillary tube is to decrease the mass fraction of the capillary tube so the difference between literature [18] and present work of mass fraction of liquid has been obtained as 0.0118, 0.004, 0.0004 and 0.0082. Hence also number of turn 40 (n=40) is suitable for helical capillary tube geometry.

The vapour mass fraction contours are observed form the above table 6.2 the main function of the capillary tube is to Increase the mass fraction of vapour of the capillary tube so the difference between literature [18] and present work of mass fraction of vapour has been calculated as 0.003, 0.009, 0.007 and 0.01. Hence also number of turn 40 (n=40) is suitable for helical capillary tube geometry.

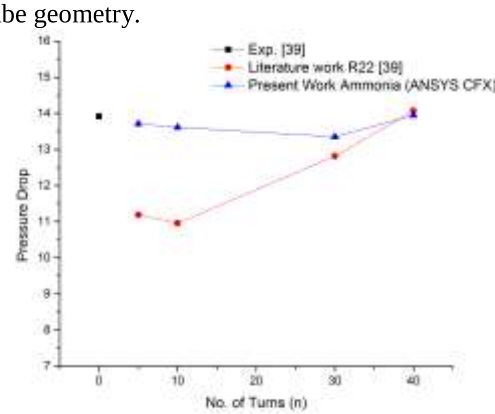


Fig 6.3: Graph plotted b/w pressure drop in tube and no. of turns

The above figure 6.3 has been shows that the pressure drop in capillary tube in literature and present work. In this graph concluded that the replacing of R22 refrigerant to ammonia is best for capillary tube. It has been also concluded that when number of coil turn is 40 the capillary tube performance is increases. In present investigation the pressure drop is 0.88% more efficient as compare to literature work [18].

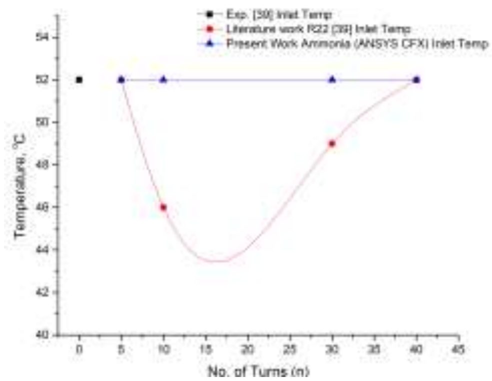


Fig 6.4: Graph plotted b/w Inlet temperature in tube and no. of turns

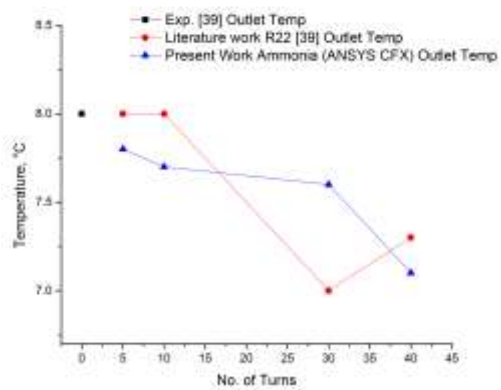


Fig 6.5: Graph plotted b/w Outlet temperature in tube and no. of turns

The above figure 6.5 has been shows that the outlet temperature in capillary tube in literature [18] and present work. In this graph concluded that the replacing of R22 refrigerant to ammonia is best for capillary tube. It has been also concluded that when number of coil turn is 40 the capillary tube performance is increases. In present investigation the temperature is fall 1.15% more as compare to literature work [18].

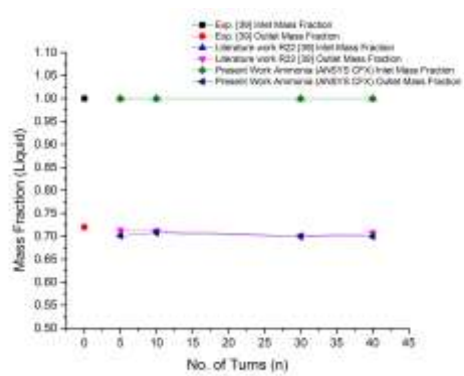


Fig 6.6: Graph potted between Mass fraction of liquid and number of coil turns

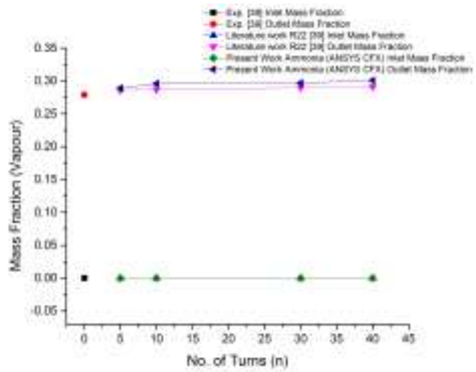


Fig 6.6: Graph potted between Mass fraction of Vapour and number of coil turns

The above figure 6.6 and 6.7 has been shows that the mass fraction of liquid and vapour formed in capillary tube in literature [18] and present work ANSYS CFX. In these investigation we have to concluded the liquid mass fraction is decrease 2.73% as compare to literature work [18] and vapour mass fraction in present work is 3.22% more as compare to literature work [18].

CONCLUSION AND FUTURE SCOPE

The following conclusion and future scope can be drawn.

- Model is created for the better performance with the existing. The helical coiled capillary tube is modeled using literature, literature which also used same design but it was used R22 refrigerant flow. In this present work has been concluded that the better result by replacing R22 to Ammonia refrigerant.
- The literature was investigating when taken diameter of tube as 1.27mm, but in present investigation has been take tube diameter as 1.11mm, because the Ammonia is easily flow when pipe diameter is small or less as compare to R22. Hence present investigation also reduces the size assembly and cost.
- The pressure drop difference between literature and present work is 2.514, 2.652, 0.534 and 0.1231. In the above results, the number of turn 40 is suitable for helical capillary tube. Hence the n=40 is suggested.
- Difference between literature [18] and present work of mass fraction of liquid has been obtained as 0.0118, 0.004, 0.0004 and 0.0082. Hence also number of turn 40 (n=40) is suitable for helical capillary tube geometry.
- Difference between literature [18] and present work of mass fraction of vapour has been calculated as 0.003, 0.009, 0.007 and 0.01. Hence also number of turn 40 (n=40) is suitable for helical capillary tube geometry.
- In present investigation concluded that the replacing of R22 refrigerant to ammonia is best for capillary tube. It has been also concluded that when number of coil turn is 40 the capillary tube performance is increases. In present investigation the pressure drop is 0.88% more efficient as compare to literature work [18].
- In present investigation the temperature is fall 1.15% more as compare to literature work [18].
- Analysis has been done for mass fraction of liquid and vapour formed in capillary tube in literature [18] and present work ANSYS CFX. In these investigation we have to concluded the liquid mass fraction is decrease 2.73% as compare to literature work [18] and vapour mass fraction in present work is 3.22% more as compare to literature work [18].
- In all above conclusion the number of 40 turn helical coil capillary tube by ammonia refrigerant flow is best for more efficient performance.
- Ammonia is the available in lower cost. The price per kg for R22 is about 2.5 times price of ammonia, hence using this type of refrigerant cost is minimizing.
- For a large cold storage, the operation costs are 20-30% lower with ammonia than R22.
- Besides that, only half as much material needs to be purchased to charge a system because the density of ammonia is half of halocarbons.
- Ammonia is also gives up to 10% more efficient (COP) as compare to other refrigerant, hence the electricity consumption is less.

REFERENCES

[1] Mikol, E. P.," Adiabatic single and two-phase flow in small bore tubes", ASHRAE Journal, Vol. 5, no. 11, pp. 75-86, (1963).

[2] Li, R. Y., Lin, Z. Y. Chen, and Z. H. Chen, " Metastable flow of R-12 through capillary tubes", Int. J. of Refrigeration, Vol. 13, no. 3, pp. 181-186, (1990).

[3] Wei, C. Z., Lin, Y. T., Wang, C. C., and Lev, J. S.," An experimental study of the performance of capillary tubes for R-407C refrigerant", ASHRAE Trans., Vol. 27, (2001).

[4] Bittle, R. R., Wolf, D. A., and Pate, M. B.," A generalized performance prediction method for adiabatic capillary tubes", HVAC & Research, Vol.14, No. 1, pp. 27-43, (1998).

[5] ASHRAE. ASHRAE Handbook Refrigeration, Chapter 44, Amer. Soc. Heat. Refrig. Air-cond. Engin., Atlanta, GA, USA., (1994).

[6] Melo, C. et al.," An experimental analysis of adiabatic capillary tubes", Jl. Of Applied Thermal Engineering, Vol. 19, pp. 669-684, (1999).

[7] Sami, S. M., and Maltais, H.," Experimental analysis of capillary tubes behavior with some R-22 alternative refrigerants", Int. Jl. Energy Research, (2000).

[8] Sami, S. M., and Doung, T.," Modeling of capillary tube for refrigerating system", Technical Report MEC/86/5, University of Sherbrooke, Quebec, 40 pages, (1986).

[9] Sami, S. M., and Tribes, C.," Numerical prediction of capillary tube behavior with pure and binary alternative refrigerants", Jl of Applied Thermal Engineering, Vol. 18, no. 6, pp. 491-502, (1998).

[10] Hopkins, N. E.," Rating the restrictor tube", Refrigerating Engineering, Nov., pp. 1087, (1950).

[11] M.M.Tayde, Pranav Datar and Pankaj Kumar, “Optimum choice of refrigerant for miniature vapour compression refrigeration system”, Indian journal of Applied Research, 2013.

[12] Nishant P. Tekade and Dr. U.S.Wankhede, “Selection of spiral capillary tube for refrigeration appliances”, International Journal of Modern Engineering Research, 2012.

[13] Ankush Sharma and Jagdev Singh, “Experimental investigation of refrigerant flow rate with spirally coiled adiabatic capillary tube in vapour compression refrigeration cycle using eco friendly refrigerant”, International Journal of Mechanical and Production Engineering Research and Development, 2013.

[14] Sudharash Bhargava and Jagdev Singh, “Experimental study of azeotropic blend(30% propane, 55% n-butane, 15% iso-butane) refrigerant flow through the serpentine capillary tube in vapour compression refrigeration system”, International Journal of Mechanical and Production Engineering Research and Development, 2013.

[15] Thamir K. Salim, “The effect of the capillary tube coil number on the refrigeration system performance”, Tikrit Journal of Engineering Sciences, 2012.

[16] Akash Deep Singh, "Flow characteristics of refrigerant inside diabatic capillary tube," Thapar University, Patiala, (2009), pp. 1-96.

[17] M.A. Akintunde, Ph.D. "Effect of Coiled Capillary Tube Pitch on Vapor Compression Refrigeration System Performance," The Pacific Journal of Science and Technology Volume 9. Number 2. November (2008), pp. 284-294.

[18] Y Raja Kumar, Dr.P Usha Sri, “CFD Flow Analysis of a Refrigerant inside Adiabatic Capillary Tube”, International Journal of Engineering Research & Technology (IJERT), ISSN: 2278-0181, Vol. 2 Issue 9, September – 2013

[19] https://www.osha.gov/SLTC/etools/ammonia_refrigeration/ammonia/