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Effect of cold orifice diameter and geometry of hot end valves on performance of converging type Ranque Hilsch vortex tube

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Abstract

Refrigeration by vortex tube works on the principle of heat transfer between two layers moving opposite to each other. Various experiments have been performed and has revealed two different approaches one for attaining high cold mass fraction and the other for attaining cold end temperature. The geometry requirement is different for the two approaches. The need is to have a tube to produce higher mass of cold air coming out at low temperature. For this purpose converging type of vortex tube is experimented and the results are promising. The results show increase in cold mass fraction as well as cold end temperature. The overall change in cold end temperature drop was 63% and the COP of the converging tube as compared to straight divergent tube increased by 102%. For conical valve angle of 45^0 air supply pressure of 5 bars and cold orifice diameter as 7 mm the lowest temperature observed was 5^0C producing cold mass fraction of about 0.9.

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*Keywords:*hot stream; cold stream; swirling flow; cold mass fraction; adiabatic efficiency

1. Introduction

Vortex tube is one of the non-conventional refrigeration methods. The vortex tube is a simple tube that separates pressurized air into two streams viz. cold and hot. The vortex tube is referred to as the Ranque Vortex Tube (RVT) being invented by George j. Ranque [1] in 1931, Hilsch Vortex Tube (HVT) [2], Maxwell–Demon Vortex Tube (MDVT) [3] and Ranque–Hilsch Vortex Tube (RHVT). [4] The vortex tube is further known as uniflow and counter

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flow tube based on the cold stream exit direction. In a counter-flow vortex tube the two exits are placed at opposite ends of the tube, and in a uniflow vortex tube the two exits are placed at the same end.

Nomenclature

C	Specific Heat of air	$KjKg^{-1}K^{-1}$
D	system enthalpy change	$KjKg^{-1}$
M	mass of air	Kg
Q	Heat exchange between system and surrounding	Kj
R	Gas constant of air	$KjKg^{-1}K^{-1}$
T	Temperature	K
W	work input of compressor	Kj

Greek Letters

n	Polytropic index of compression process	
θ	conical valve angles	degrees
Φ	divergent cone angle of tube	degrees
η	efficiency of the tube	

Subscripts

a	adiabatic process parameter
c	compressor efficiency
i	supply inlet condition
C	cold stream
H	hot stream
1	inlet
2	outlets

The vortex tube consists of nozzle, diaphragm, hot end valve, hot end, and cold end. [1] The schematic diagram of vortex tube is shown in the Fig.1.

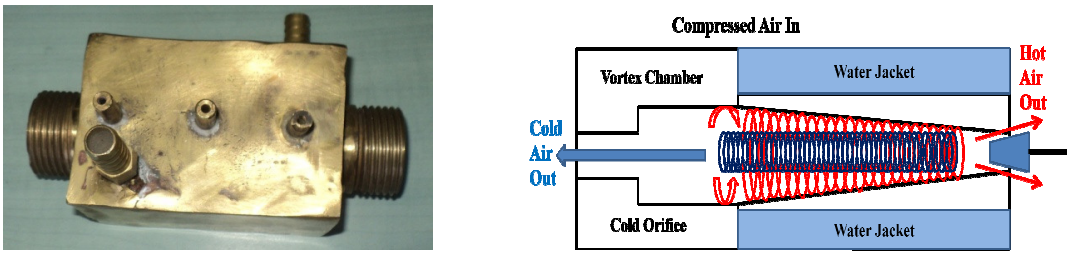

Figure 1 Vortex tube schematic

Compressed air enters through the nozzle as shown in Fig.1. At entry the air expands and acquires high velocity due to particular shape of the nozzle. A vortex flow is created in the chamber and air travels in spiral like motion along the periphery of the hot side at around million rpm's. This swirling flow is restricted by the valve. When the pressure of the air near valve is made more than outside by partly closing the valve, a reversed axial flow through the core of the hot side starts from high pressure region to low-pressure region. During this process, heat transfer takes place between reversed stream and forward stream. Therefore, air stream through the core gets cooled, while air

stream in forward direction gets heated up. The cold stream is escaped through the diaphragm hole into the cold side, while hot stream escapes through the valve opening at hot end. By controlling the opening of the valve, the quantity of the cold air and its temperature can be varied. [1] The cold air thus obtained can be used in applications like air suits for cold mine workers, spot cooling of machining operations, electronic panel and circuit cooling, for setting hot melts, cooling of soldered parts in circuit.

Thus vortex tube can become good alternative for commercial vapor compression and absorption refrigeration system that utilizes refrigerants for producing refrigeration effect. Wide scale commercialization of this concept is still not unveiled because of its low thermal efficiency and inability to produce large mass of cold air. Singh et al. [5] varied L/D ratio of the vortex tube and observed that the effect of nozzle design is more important than the cold orifice design for getting higher temperature drops. Cold mass fraction as well as adiabatic efficiency is more influenced by the size of the cold orifice rather than the size of the nozzle. GAO [6] with various length-diameter ratios found that with the length the temperature drop increases. Varying the shape of hot end plug does not affect the performance of the tube. Arzomandi and xue [7] studied the effect of varying area of hot end plugs where Size of plug determines the cold mass fraction and results in maximum efficiency when the area ratio (area of tube to area of orifice) is between 0.9 and 0.98. Devade and Pise [8] studied the effect of varying conical valve angles on efficiency of vortex tubes and concluded that for short straight divergent tubes of L/D equal to 8 valve angle of 45^0 gives maximum cooling effect and valves of 60^0 angle provide better heating effect , valves are shown in Fig. 2.

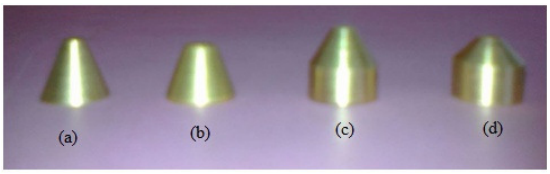

Figure 2 Conical valves used by devade and pise

Markal et al. [9] studied the effect of the valve angle of counter-flow Ranque–Hilsch vortex tubes on thermal energy separation, it is observed that the valve angle has a weak influence on the system performance. Promvonge [10] studied the effect of number of tangential entries, cold orifice diameter and tube insulation the results show that the insulated vortex tube with 4 inlet nozzles and cold orifice diameter of 0.5D yielded the highest temperature reduction (temperature separation) and isentropic efficiency at about 30^0C and 33% respectively. The increase of the number of inlet nozzles led to higher temperature separation in the vortex tube.

Dincer et al. [11] performed experiments with hot plug located at one end, effects of position, diameter (5, 6, 7, 8 mm) and angle (30°–180°) of a mobile plug. The maximum difference in the temperatures of hot and cold streams was obtained for the plug diameter of 5 mm, tip angles of 30° and 60°, 4 nozzles and by keeping the plug location at the far extreme end. Nimbalkar and Muller [12] investigated for the optimum geometry for cold end orifice of the tube, for different inlet pressures and cold fractions. The experimental results indicate that there is an optimum diameter of cold end orifice for achieving maximum energy separation. Dincer et al. [13] performed experiments with various nozzle areas and maximum temperature difference was obtained for 3x3 sq.mm nozzle area. Exergy efficiency strongly depends upon the input pressure, cold mass fraction, and cold mass velocity.

Aydin et al. [14] a new geometry is introduced for the cold end side (i.e. where the swirl flow is introduced into the tube), which is called ‘helical swirl flow generator’. S. Eiamsa-ard [15] experimented with multiple snail entries to vortex tube. The experimental results reveal that the RHVT with the snail entry provides greater cold air temperature reduction and cooling efficiency. The increase in the nozzle number and the supply pressure leads to the rise of the swirl/vortex intensity and thus the energy separation in the tube.

Wu et al. [16] developed three innovative designs to improve the efficiency of the vortex tube a new nozzle with equal gradient of Mach number and a new intake flow passage of nozzles with equal flow velocity were designed and developed to reduce the flow loss. The cooling effect of the vortex tube with diffuser is up to 5^0C lower than that

without diffuser. Wu used new entry geometry. And nozzle along with diffuser at hot end for the experimentation and enhanced the results by almost 5%.

Arzomandi and xue [17] experimented with vortex angle entries the vortex angle plays important role in both the temperature separation and the vortex tube performance. Smaller vortex angle gives larger temperature difference.

The above work has identified that vortex tube for low temperature and high cold mass fraction both are different issues and cannot be combined, but combined high CMF and Low temperature is the need of today, and hence this attempt is made to attain both. Also the conical valve angle has no effect is stated, the same is proposed to study.

2. Test rig and converging vortex tube

The vortex tube to produce high mass of cold air and low temperatures is designed to overcome the disadvantage of straight and divergent section hot tube. The short divergent tube is prone to mixing of cold and hot streams at orifice of the tube and because of this mixing the cold end temperature is observed higher. Hence the tube is designed with length to diameter ratio as 15. The diameter of the tube is maintained 14mm on hot end the converging taper is set to 6 degrees. Different cold end orifices are used with diameters as 5mm, 6mm and 7mm respectively. The purpose is to study the effect of orifice diameter on the performance of the tube. Four different conical valves of angles of 30^0, 45^0,60^0,90^0 are used on hot end. All these designed parameters are summarized in the Table1.

Table 1. Vortex tube geometry

No.	Parameter	Dimensions and number
1.	Tube diameter entry	36mm
2.	Tube diameter exit	14mm
3.	Inlet nozzle diameter	4mm
4.	Cold orifice diameter	5,6,7mm
5.	Length of tube	225mm
6.	Cone angle, φ	6^0
7.	Conical valve angles Θ	30^0,45^0,60^0,90^0
8.	No. of entry nozzles	2

3. Experimental Method

3.1. Experimental Set Up

The test rig for experimentation consists of a compressor operating at 8 bar pressure, FRL unit, rotameter, vortex tube, and thermocouples for temperature measurement. The set up schematic is shown in Fig.3. Various instruments mounted on the vortex tube are well calibrated and give precise readings.

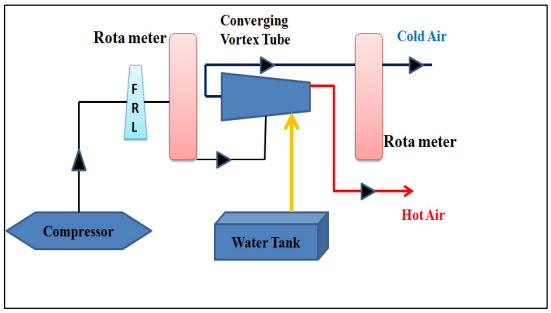

Figure 3 Experimental setup

The details of the instruments are furnished here for the reference. Table 2 shows the details of the instrumentation provided with the test rig.

Table 2. Measuring instrument details

Instrument	Range	Uncertainty
Rota-meters	0 to 300 LPM	± 3%
FRL unit	8 bar	
RTD (PT 100)*	-50 to 250°C	± 0.1°C

For temperature measurement, three RTD sensors and a digital temperature indicator having least count of 0.1°C is used. To measure the flow of air rotameter is used having least count 10 LPM. The FRL (Filter, Regulator and Lubricator) unit is also required to ensure clean air free from dust and moisture. Regulator is used to control the inlet pressure. Lubricator is used to lubricate the air. Out of two rota meters, one is mounted in inlet line to measure Flow rate of inlet air and another one is used to measure the flow of air coming out of cold end of vortex tube.

3.2. Test Procedure

The experimentation is carried out with converging vortex tube by using various orifice diameters with each type of valves and with pressures ranging from 2 to 5 bars. For each valve and orifice diameter and for every entry pressure 3 sets of readings are taken and then the average temperature drop and rise on cold and hot ends respectively is noted.

Compressed air is taken as input fluid and it is made to pass through the FRL unit to regulate the pressure then it passes through 2 tangential nozzles into the vortex chamber and comes out through hot and cold ends. With the help of valve adjustment mechanism the valve opening is adjusted to get the optimum reading for each set. The temperature measurement is done at inlet of the air and then at cold and hot ends respectively using PT 100 thermocouples. The cold air is then made to pass through the flow meters on outlet line. Using flow meters the mass of air supplied and the mass of air coming out as cold stream can be thus measured.

3.3. Data Reduction

An examination of the system at steady state indicates that from the first law of thermodynamics,

$$D_h = Q \quad (1)$$

Assuming that Q is approximately zero even though the cold tube may have frost on it and the hot tube is very

warm, if this is the case then,

$$D_h = Q_h + Q_{hc} = 0 \quad (2)$$

Assuming the air as an ideal gas, the total enthalpy change can be written as [11]

$$D_h = m_h C_p\,(T_h - T_i) + m_c C_p\,(T_i - T_C) = 0 \quad (3)$$

To estimate its coefficient of performance (COP), Focusing on the cooling effect that can be achieved by placing the cold pipe within an enclosure, the COP can be calculated by,

$$COP = \frac{Qhc}{W} \quad (4)$$

Where Qhc is obtained from,

$$Qh_C = M_C.C_P\,(T_I - T_C) \quad (5)$$

And W in the present case is the work done to compress the air from atmospheric pressure and temperature to the inlet conditions of the tube. Assuming reversible compression (isothermal, minimum work), W then obtained from:

$$\dot{W} = P_1.V_1 log_e \frac{P_2}{P_1} \quad (6)$$

$$W = M_i.C_p(T_2 - T_1) \quad (7)$$

The refrigeration effect of the vortex tube is simply equal to Q_c [12],

$$Qh_c = M_c.C_p.(T_i - T_C) \quad (8)$$

The cold mass fraction of the tube can be calculated by

$$CMF = mc/mi \quad (9)$$

If the process had undergone an isentropic expansion from inlet pressure Pi to atmospheric pressure Pa at the cold end then the static temperature drop due to expansion is given by

$$\Delta T'c = Ti - T'c = Ti[1 - (Pa/Pi)(\gamma - 1)/\gamma] \quad (10)$$

The temperature drop occurred in Vortex tube is$\Delta T'c$. The ratio of ΔTc to $\Delta T'c$ is called Relative Temperature drop

$$\Delta Trel = \Delta Tc/\Delta T'c \quad (11)$$

The product of μ and $\Delta Trel$ represents the adiabatic efficiency of the Vortex tube because it is defined as

Similarly theoretical COP for the vortex tube can also be calculated with assuming adiabatic compression with work done as,

$$W_{comp} = mrt * (pi/pa)^{\gamma-1/\gamma} - 1) \quad (12)$$

Theoretical cop of vortex tube is

$$C.O.P = \eta ab.\eta ac.[(Pa/Pi)(\gamma - 1)/\gamma] \quad (13)$$

4. Results and Discussions

After extensive experimentation on converging vortex tube with varying conical valve angles for different inlet pressures and various orifice diameters results are obtained. These are discussed in terms of effect of conical angles on cold end temperatures, temperature difference, cooling effect, COP and adiabatic efficiency.

4.1. Effect of conical valve angles on cold end temperature

Figure 4 shows the effect of conical valve angles and orifice diameter on cold end temperature. For 5, 6 and 7 mm orifice diameters and valves of 30^0, 45^0, 60^0, 90^0 angles. The figures are plotted for 5 mm 6mm and 7mm orifice diameters individually. The results depict that the performance of 45^0 conical valve angles is best for highest supply pressure of 5 bars and with orifice diameter as 7mm; the temperature observed is 5^0C on cold end side. Performance of 90^0 valves is also comparable to that of 45^0 conical valves it also produces the low temperature on cold end. The 30 and 60 degrees conical valve performance is not much promising.

The result show that with increase in valve angles the temperature drops at all supply pressures, Fig.4, Fig.5, Fig.6. With change in valve angles the flow is guided and when valve angle is 45 degrees the reversal of flow is smooth and as the orifice diameter increases chances of secondary circulation are minimized hence there is no

mixing of the hot stream and cold stream. At highest input pressure and adiabatic expansion at the entry the temperature drop increases. The expanding flow as reaches the hot end because of the converging section of the tube attains high velocity. The velocity along the axis increases because of convergent section and provides potential for the heat transfer among the hot stream and cold stream. The results also reveal that the mass of air escaping out from the hot end is less. If the pressure is held constant, changes in valve angles shows change in temperature in the descending sequence of 60, 30, 90 and 45 degrees. Thus valve angle has effect on the energy separation. Almost 15 to 25% changes are observed with increasing valve angle from 30 to 45 degrees.

Maximum cold mass of air is obtained on the cold end side because of the converging section of the tube. For most of the experiments cold mass fraction is 0.8, 0.9 to 1. The cold mass fraction obtained is promising.

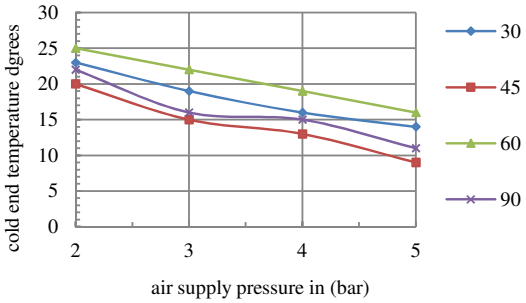

Figure 4 Effect on cold end temperature orifice, dia =5

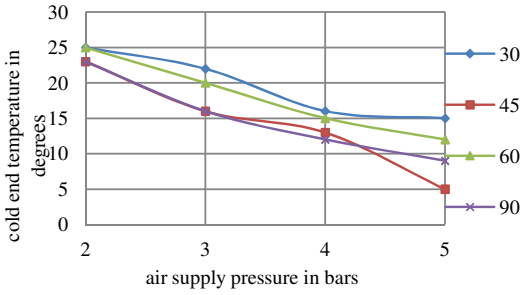

Figure 5 Effect on cold end temperature orifice, dia=6

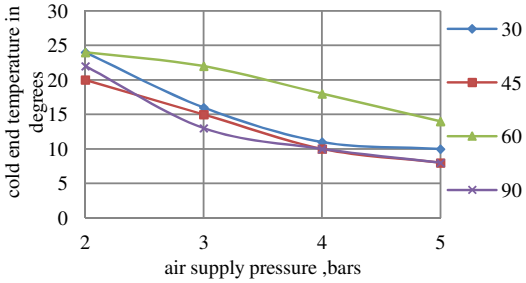

Figure 6 Effect on cold end temperature orifice, dia=7

4.2. Effect of conical valve angles and orifice diameters on COP

The COP of vortex tube is usually very low but with converging type of vortex tube the efficiency is seen to be increased and the same is shown in the following figures. The COP of the vortex tube primarily depends on the cold mass of air and the cold end temperature produced. In case of converging type of tube low both mass flow rate of the cold air and the temperature drop are significant hence for majority of the cases as pressure increases the COP of the tube increases. And for the orifice diameter of 7 mm highest COP is observed for 45 degree valve.

The Figures7, 8, 9 and 10 above show the maximum obtained COP is 0.202 for 45 degree valve and orifice diameter of 7mm for supply pressure of 5 bars, as for this combination the cold mass fraction obtained is 0.9. The temperature drop is more than that of the isentropic temperature drop, because of the optimum dimension ratio of 0.5 between the tube and the orifice, there is no secondary circulation.

As seen in Fig. 9 for all the valves COP increases for converging tube with pressure. The Ranque Hilsch effect is now combined with nozzle effect and thus the COP of the tube increases with supply air pressure.

The complete mass of air undergoes expansion and is returned back to the cold end with lowest temperature as 5^0C. The increase in cold mass of air is because of the converging section of the tube.

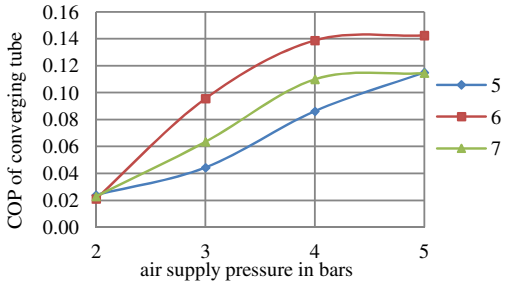

Figure7 Effect on COP for 30 degree valve

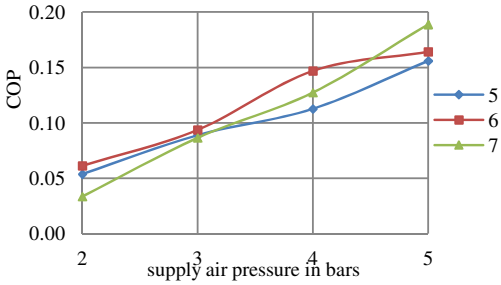

Fig. 8 Effect on COP for 45 degree valve

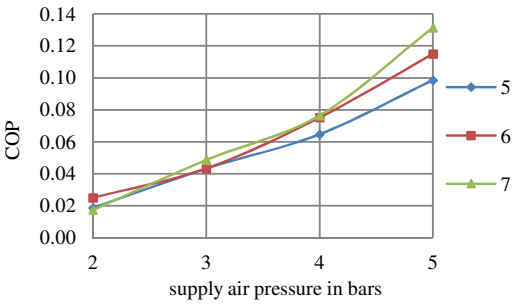

Figure 9 Effect on COP for 60 degree valve angle

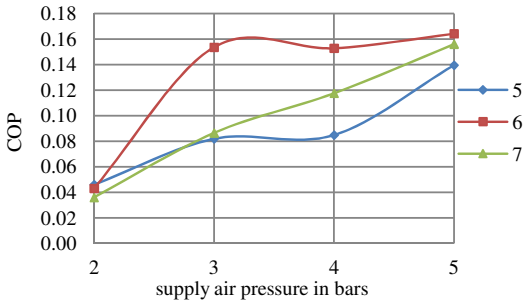

Fig. 10 Effect on COP for 90 degree valve

4.3. Effect on actual and static temperature drop

Static or isentropic temperature drop is based on the isentropic expansion of the air inside the tube and actual temperature drop is practically obtained as difference between the inlet temperature and cold outlet temperature, the following Figures 11, 12, 13 and 14 shows the effect of valve angles and orifice diameters on the static and actual temperature drops. Static temperature drop is also dependent on the supply air temperature. As the temperature and pressure ratio is fixed the static temperature drop is fixed.

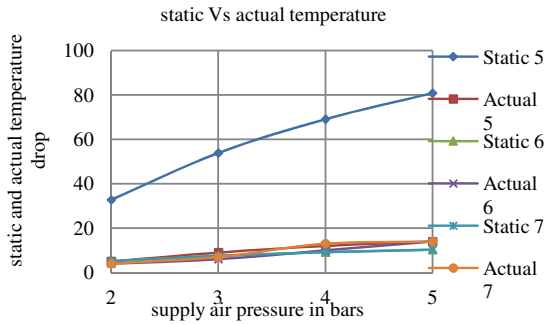

Figure 11 Static and actual temperature drop for 30 degree valve valve

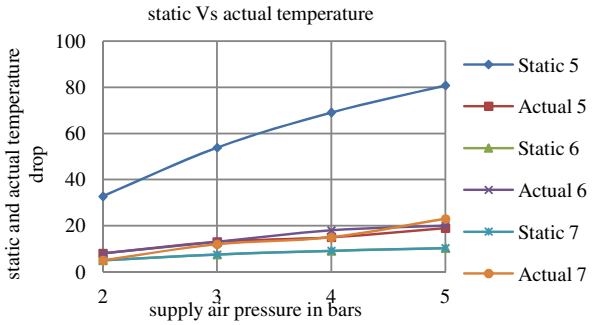

Fig. 12 Static and actual temperature drop for 45 degree

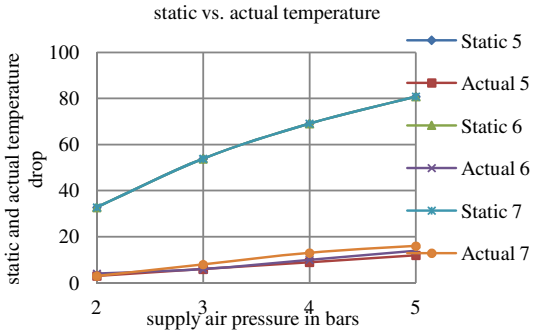

Figure 13 Static and actual temperature drop for 60 degree valve

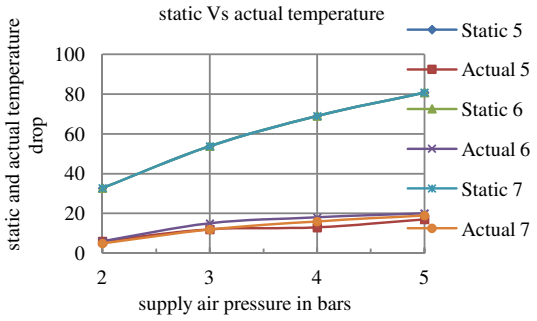

Fig. 14 Static and actual temperature drop for 90 degree valve

4.4. Effect on adiabatic effectiveness for cold mass fraction of 0.9

Adiabatic efficiency of the tube is the ratio of actual drop of temperature at the cold end to the adiabatic temperature drop considering the adiabatic expansion of the air. The following Fig.15 shows the effect on adiabatic efficiency at cold mass fraction of 0.9.

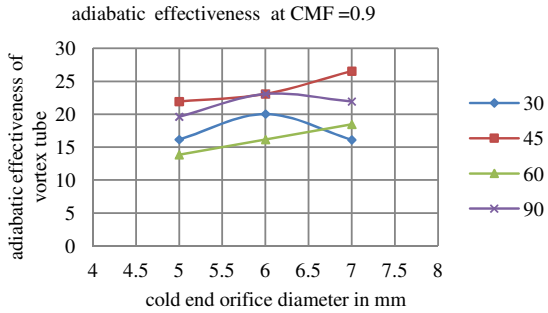

Figure 15 Adiabatic effectiveness of VT

The Fig. 15 shows that the adiabatic efficiency of the tube increases to 208% as the actual temperature drop is more than that of the theoretical temperature drop. This result is for valve angle of 45 degrees and 7 mm orifice diameter and for area ratio of 0.06, for this area ratio highest efficiency is observed.

4.5. Comparison between actual and theoretical COP of the tube

The COP of the vortex tube is the ratio of cooling effect obtained and the compressor work input. The theoretical COP assumes that the compressor work is minimum i.e. the compression is isothermal and actual COP deals with the actual work input required for the compressor. A small deviation is noted between the two but it is quite comparable. Following Fig.16 indicates the same.

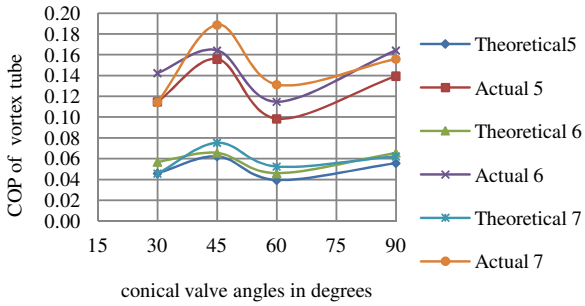

Figure 16 COP of vortex tube

4.6. Effect of area ratio on COP

Following figures Show the effect of ratio of area of orifice to the area of vortex tube on COP of the tube. All the Figures 17, 18, 19 and 20 above show the effect of area ratio on COP of the tube and it can be seen that the COP increases with increase in the air supply pressure and COP also increases with area ratio. Maximum COP is found for 30^0 and 90^0 valve angle at area ratio of 0.05. For all other valves Cop increases with increase in pressure and area ratio.

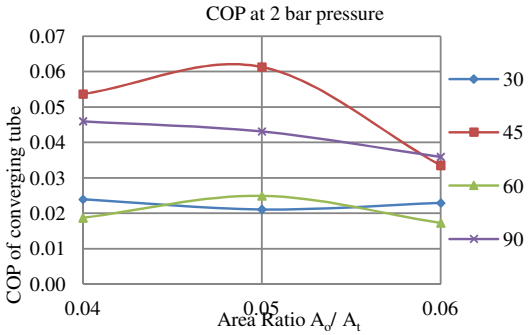

Figure 17 Effect of area ratio on COP

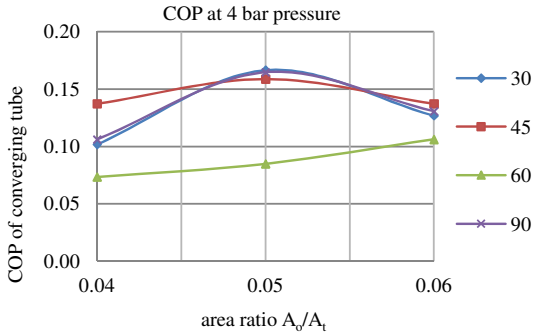

Fig. 18 Effect of area ratio on COP

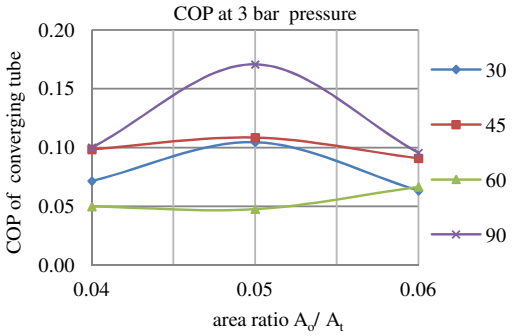

Figure 19 Effect of area ratio on COP

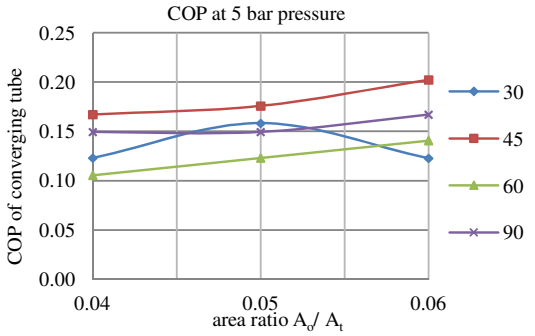

Fig. 20 Effect of area ratio on COP

Similarly at 4 bar Pressure as seen in Fig. 19 the 30 and 90 degree valve shows good performance at area ratio of 0.05. But at higher and lower area ratios COP decreases because at higher valve angle flow directions change

suddenly on hot end side. Therefore, for higher values of the valve angle, flow instability arises, and efficiency decreases.

For 30^0 valve angle the flow experiences turbulence between the tube walls and the valve end, because of the rotation of the flow the mixing of the two streams take place causing COP to drop.

4.7. Effect of Conical valve angles at CMF =0.9

When maximum cold mass of air is desired conical valves at exit have certain effect and the effect varies with orifice diameters. Following figures indicate the effect of valve angles and orifice diameter on cold end temperature difference, effect on cooling effect and effect on COP (Fig.21)

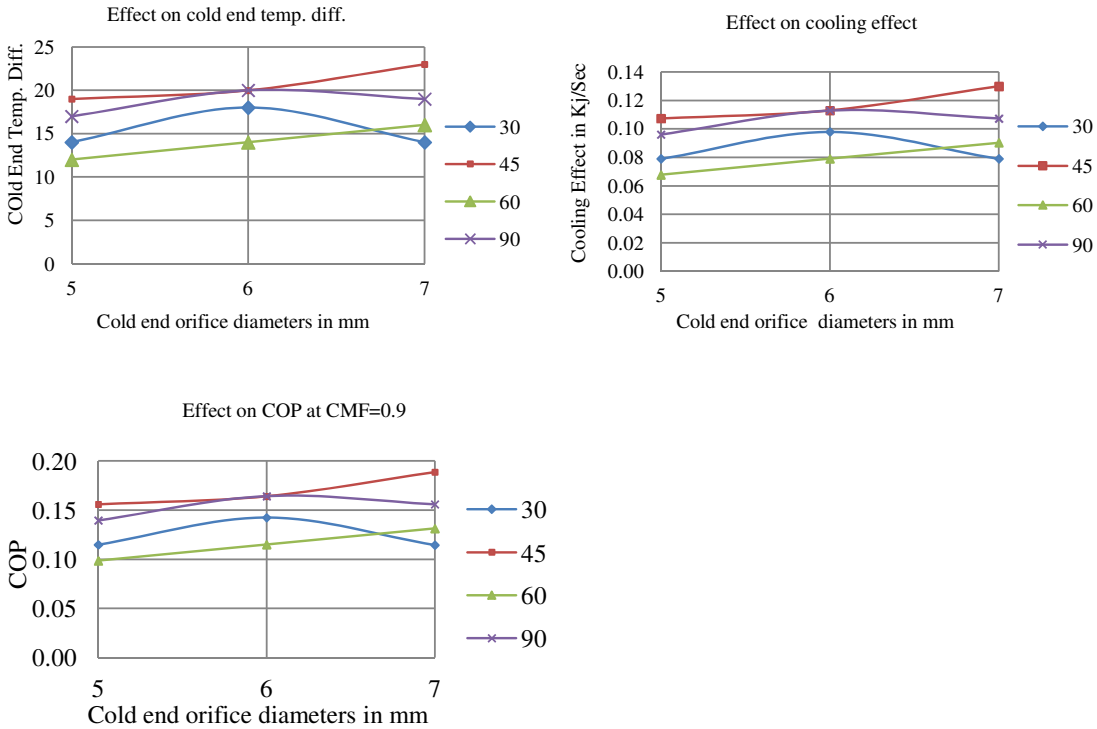

Figure 21 shows the effect of orifice diameter and valve angles on cold end temp. diff., cooling effect and COP

The Figure 21 shows that at CMF of 0.9 cold end temperature difference increases as the orifice diameter increases for all valve angles except 30^0. This indicates that the effects of secondary circulation can be nullified. As cold end temperature difference increases we get more cooling effect and similarly it results in higher COP as shown by the graphs.

5. Conclusion

With reference to the discussed results it can be concluded that the converging type of vortex tube has proved to be promising as far as the optimization of cold mass fraction and lower cold end temperatures are considered. It has satisfactorily produced lower temperature of about 5^0C and cold mass fractions of the order of 0.9 with COP as high as 0.202. The adiabatic efficiency of the tube is on higher side and is 208%. But small deviation of 0.39 is observed

in theoretical and actual COP of the tube. The same can be improved and brought nearer to the theoretical value. Further investigations with changes in valve geometries and using higher input pressures are possible. For getting closer results insulated type of converging vortex tube can also be tested for obtaining close to adiabatic performance. The valve geometry and cold orifice diameter influence the performance of such tubes on larger scale and can become a promising refrigeration device in future.

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