Thermodynamic Analysis Of Counter Flow Vortex Tube

Upendra S. Gupta (Asst. Prof. SVITS Indore)

Manoj K. Joshi (SGSITS Indore)

Saurabh Rai

Abstract: The vortex tube also known as Ranque tube, with no moving parts and reliable device it produces hot and cold gas streams simultaneously from the source of the compressed gas, An experiment has been conducted to evaluate the thermodynamic analysis of the vortex tube. During the study the cold mass fraction was varied from 0.2-0.8 for a fix inlet pressure of 4 bar and the inlet pressure was varied between 2 -7 bar for a fix opening of the cone valve. The maximum temperature drop was observed for cold mass fraction of 0.4 and the effective refrigerating effect was observed between the 0.35-0.65 of the cold mass fraction, as the refrigerating effect and heating effect is a function of mass of the cold air and the temperature drop. A thermodynamic model has been used to investigate vortex tube energy separation, the results indicated that as hot tube length or inlet pressure increases the temperature difference increases too,The second law analysis shows that the for air, it is middle tube which generates lowest entropy.

Keywords:-Vortex tube (RHVT), counter-flow type vortex tube, energy separation, temperature reduction, coefficient of performance (COP),Second law analysis .

Introduction: The vortex tube also known as Ranque tube is a remarkably a simple device, reliable (since no moving parts) and produces hot and cold gas streams simultaneously from the source of the compressed gas. It is also light in weight, ozone friendly, free from pollution, low cost and effective solution to a wide variety of industrial cooling problems especially like drilling, turning and welding and heating problems as early start of the boilers and drying chambers etc.

The vortex tube was first discovered by G. J. Ranque (1933) then Hilsch (1947) [1], a German engineer, performed comprehensive experimental and theoretical studies aimed at improving the efficiency. Martynovskii and Alekseev [2] studied experimentally the effect of various design parameters of vortex tubes. Hilsch [1] suggested that ratio L/D should be around 50 for good temperature separation. According to Westley [3] the only requirement is that the tube exceeds 10D. Gulyaev [4,5] determined that the minimum length for cylindrical hot tube was about 10D. If the hot tube is conical, rather than cylindrical, the minimum length must be increased, to about 13D. Lewellen [6] stated that ‘‘as long as the tube wall is insulated the temperature separation in the tube remains unaffected by L/D as long as some minimum length is exceeded. Takahama and Yokosawa [7] suggested a tube length L/D 100 in order to obtain a better performance. Amitani et al. [8] indicated that the shortened vortex tube of six tube diameters length had the same efficiency. Saidi and Yazdi [9] found that increasing tube length increases temperature differences and decreases exergy destruction. For L/D 20 energy separation was quite low. Saidi and Valipour [10] concluded that the optimum value of L/D is in the range of 20–55. Singh et al. [11] concluded that length of the tube has no effect on the performance of the vortex tube in the range of L/D 45–55. Behera et al. [12] presented that increase in the length of tube enhances the temperature separation up to the condition that stagnation point is within the length of tube. The investigations had shown that L/D ratio should be more than 10 for the tube to be effective. Gao [13] investigated three different shapes of hot plug to study the energy separation effect and concluded that the shape of the hot plug is not a critical component in RHVT.

Promvonge and Smith [14] concluded that increasing the number of inlet nozzle increases the temperature drop and the isentropic efficiency. Prabakaran. J et al [15] performed investigation for vortex tube of L/D ratio 10 and three different sizes of orifices i.e. 5mm, 6 mm and 7mm and inlet pressure 4 bar and concluded that the maximum temperature drop is obtained for 5mm and the temperature difference is reduced as the size of the orifice increase.

Volkan kirmaci [16] used Taguchi method to optimize the design of counter flow Ranque-Hilsch vortex tube for the performance. Balmer (1988) who investigated theoretically the temperature separation phenomenon in a vortex tube, used the second law of thermodynamics to show temperature separation effect with a net increase in entropy is possible when incompressible liquids are used in the tube. This was confirmed by experiments with liquid water which showed that temperature separation occurred when an inlet pressure was sufficiently high. Silverman (1982)[17] questioned whether the VT is a violation of the second law of thermodynamics or not. Usage of exergy concepts in evaluating the performance of energy systems are increasing nowadays due to its clear indication of loss at various locations which is more informative than energy analysis. Exergy is the work potential of energy in a given environment. It has been also employed in studying vortex tube performance theoretically. Rosen and Dincer (2004) studied the effect of dead state variation on energy and exergy analysis of thermal system and showed that the variation does not affect the energy and exergy values significantly. In exergy analysis losses are measured in terms of exergy destruction, which provide direct measure of thermodynamic inefficiencies. Saidi (1999)[9] studied the effect of inlet pressure on temperature difference in the vortex tube and discussed the advantage of an exergy analysis. He also listed equation for calculating rates of entropy generation and total irreversibility. Kırmacı (2009)[16] studied the exergy analysis on vortex tube for two different gases (air, oxygen) by using different inlet pressures and different nozzle numbers. Farezaneh-Gord and Kargaran (2010)[23] have carried out an experimental study to investigate natural gas temperature behaviors in a VT. The effects of the VT cold orifice diameter on the VT thermal separation are also studied. Farzaneh-Gord et al. (2012a)[24,25] have studied the effects of hot tube length on the VT thermal separation.

In this work, an attempt was made to fabricate and test the performance of a counter flow vortex tube of total length 15D and a vortex generator which was fixed inside the tube to create four vortices along the inner periphery of the rube. Different parameters were evaluated like temperature reduction in cold tube end, temperature rise in hot tube, refrigerating effect, heating effect, amount of entropy generation and the coefficient of performance was compared for different operating parameter like cold mass fraction and inlet pressure.

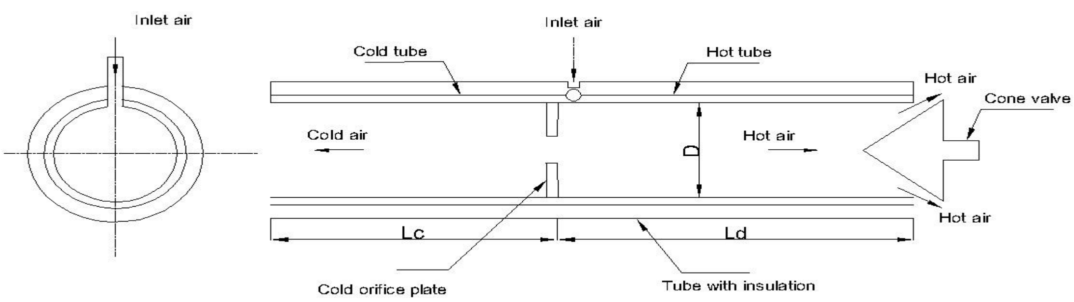

Fig. 1 Schematic sketch of Counter-flow Vortex Tube.

Geometrical parameter of Counter Flow Vortex Tube

Parameter	D(mm)	d_c(mm)	d_i(mm)	d_n	L_h(mm)	L_c(mm)	Φ(degree)
Value	12	6	4	2	120	30	45

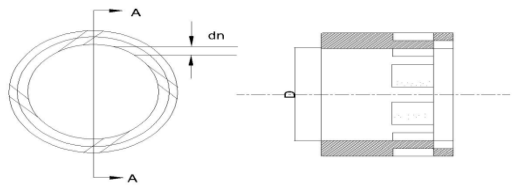

Fig 2. Nozzle Configuration or Swirl generator

EXPERIMENTAL SET UP

The detailed drawing and of the vortex tube is given in Fig. 1 and Fig.2. The schematic sketch of the complete experimental set is as shown in Fig.3. It consists of a two stage compressor and a receiver as a source of compressed air, auto pressure cut-off switch, an air filter, a counter-flow vortex tube. Compressed air from the receiver of compressor is supplied through a hand operated control valve to control the pressure at the inlet to the vortex tube as shown in figure. The pressure at the inlet to the vortex tube is measured with the help of a calibrated pressure gauge indicator. The temperature of the hot air and temperature of the cold air coming out of the vortex tube is measured with the thermocouple located immediately on the downstream of the cone shaped valve, and downstream of the orifice located next to the inlet respectively. The temperature of the air is also measured at the inlet to the vortex tube to calculate the temperature drop or temperature rise of the cold and hot air respectively. The thermocouples along with the digital indicator used in this experiment are calibrated to an accuracy of ±0.1°C.The mass flow rates of the cold air and hot air discharges are measured by calibrated orifice flow meters. The pressure difference across the orifice is measured by an inclined tube manometer connected to the pressure tapping at distance d (orifice diameter) on the upstream side and d/2 on the downstream side of the orifice. The ratio, called a cold mass fraction is changed by regulating the cone-shaped valve opening. The complete vortex tube is further insulated with the insulating material to minimize the heat loss/gain from the surrounding.

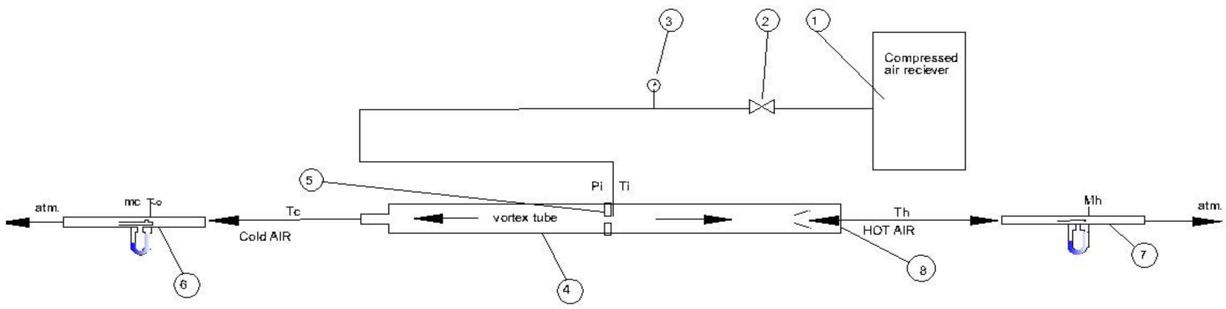

Fig 3. Experimental apparatus, 1) Compressed air Receiver , 2) Hand operated Valve , 3) Pressure gauge , 4)Counter flow Vortex Tube , 5) a set of orifice flow meters,6-7) Orifice Flow meter, 9) Cone-shape valve.

Thermodynamic Analysis

The cold flow mass ratio (cold mass fraction) is the most important parameter used for indicating the vortex tube performance of RHVT. The cold mass fraction is the ratio of mass of cold air that is released through the cold end of the tube to the total mass of the input compressed air. It is represented as follows

$$\varepsilon = \frac{m_c}{m_i} = \frac{T_i - T_h}{T_c - T_h} \qquad (1)$$

Where, m_c represents the mass flow rate of the cold stream released, m_i represents the inlet or total mass flow rate of the pressurized air at the inlet. Therefore, ε varies in the range of 0-1.

Cold air temperature difference or temperature reduction is defined as the difference between inlet flow temperature and cold air temperature:

$$\Delta T_c = T_i - T_c \qquad (2)$$

where T_i is the inlet flow temperature and T_c is the cold air temperature. Similarly, hot air temperature difference is defined as

$$\Delta T_h = T_h - T_i \qquad (3)$$

Had the expansion been isentropic from inlet of the nozzle to the exit pressure and the air to behave like an ideal gas, isentropic efficiency is given by

$$\eta_{is} = \frac{T_i - T_c}{T_i - T_{is}} \qquad (4)$$

For isentropic expansion the exit temperature T_{is}

$$T_{is} = T_i \left(\frac{p_e}{p_i}\right)^{\frac{\gamma-1}{\gamma}} \qquad (5)$$

Where, p_e is the exit pressure of the cold air i.e. atmospheric pressure (p_a) at outlet.

The refrigerating /cooling effect produced by the cold air of vortex tube is give as

$$Q_c = m_c C_p (T_c - T_i) \qquad (6)$$

Since cooling and heating streams are obtained simultaneously the heating effect produced by the vortex tube is give as

$$Q_h = m_h C_p (T_h - T_i) \qquad (7)$$

Since RHVT can be used as a cooler and heater simultaneously hence both the effect that i.e. cooling effect and heating effects are considered. The COP of the system is calculated accordingly. The coefficient of performance of refrigerator is defined as the ratio of refrigerating effect produced by the system to the work done on the system. In the conventional vapor compression refrigeration (VCR) system work input or power is the work of compression or the compressor work. But, the vortex refrigeration systems are used where compressed air or gas is available. Making analogy to the VCR system, the work of compression from the exit/atmos. i.e. from p_e to p_i by a reversible isothermal process, the COP of the system is given as

$$COP_{ref} = \frac{\gamma\varepsilon}{\gamma - 1} \frac{(T_i - T_c)}{T_i ln\left(\frac{p_i}{p_a}\right)} \tag{8}$$

$$COP_{heat\ pump} = \frac{\gamma(1-\varepsilon)}{\gamma - 1} \frac{(T_h - T_i)}{T_i ln\left(\frac{p_i}{p_a}\right)} \tag{9}$$

In the above relation the pressure drop in the supply pipe has been neglected and the pressure at the exit of the cold and the hot air in the vortex tube are assumed to be atmospheric.

The second law analysis

The first law analysis accounts only for heat balance and cooling or heating effect. The second law analysis based on irreversibility approach, considers also the losses due to separation effect as well as the pressure drop losses. As mentioned by Saidi and Yazdi (1999), the separation losses account for a significant portion of vortex-tube irreversibility. Therefore, the vortex-tube optimal design couldn't be accessible following a first law analysis. Assuming steady state condition and neglecting heat transfer, the second law of thermodynamics for the system could be expressed as:

$$S_{gen} = m_h s_h + m_c s_c + m_i s_i \tag{10}$$

Applying the mass conservation and considering the definition of the cold mass fraction, the above equation could be re-expressed as:

$$S_{gen} = m_i\big((1-\varepsilon)s_h + \varepsilon s_c - s_i\big) = m_i\big((1-\varepsilon)(s_h - s_i) + \varepsilon(s_c - s_i)\big) \tag{11}$$

The equation 11 could be non-dimensionalized as:

$$s_{gen} = S_{gen}/m_i\, C_p = 1/C_p\big((1-\varepsilon)(s_h - s_i) + \varepsilon(s_c - s_i)\big) \tag{12}$$

For the case of a prefect gas, by employing the following Tds relation

$$s - s_i = C_p ln\frac{T}{Ti} - \text{R}ln\ \frac{p_a}{(p_i)} \tag{13}$$

The equation 12 could be simplified as

$$s_g = (1-\varepsilon)ln\frac{Th}{Ti} - (1-\varepsilon)\frac{\Upsilon - 1}{(\Upsilon)}ln\frac{ph}{pi} + \varepsilon ln\frac{Tc}{Ti} - \varepsilon\frac{\Upsilon - 1}{(\Upsilon)}ln\frac{pc}{pi} \tag{14}$$

In most cases (as here) , this makes the above equation to be more simplified as: $p_c = p_h$

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$$s_g = ln\left(\frac{T_h{}^{1\text{-}\varepsilon}\, T_c{}^{\varepsilon}}{T_i}\right) - \left(\frac{p_c}{p_i}\right)^{\frac{\gamma-1}{\gamma}} \qquad (15)$$

If the process could be assumed reversible, then:

$$T_{cs} = T_h{}^{1\text{-}\varepsilon}\, T_c{}^{\varepsilon} - T_i \left(\frac{p_c}{p_i}\right)^{\frac{\gamma-1}{\gamma}} \qquad (16)$$

For the case of an irreversible process

$$T_{cs} = T_h{}^{1\text{-}\varepsilon}\, T_c{}^{\varepsilon} - T_i \left(\frac{p_c}{p_i}\right)^{\frac{\gamma-1}{\gamma}} = e^{sg} T_i \left(\frac{p_c}{p_i}\right)^{\frac{\gamma-1}{\gamma}} \qquad (17)$$

The equation 17 and 1 could be combined to obtain an equation for determining cold stream temperature in case of knowing dimensionless entropy generation as:

$$\left(\frac{T_{i-\varepsilon T_c}}{1-\varepsilon}\right)^{1-\varepsilon} T_c{}^{\varepsilon} = e^{sg} T_i\, ln\left(\frac{p_c}{p_i}\right)^{\frac{\gamma-1}{\gamma}} \qquad (18)$$

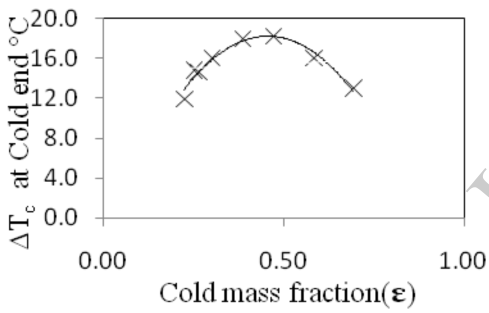

Fig.4 Temperature Drop at cold end as a function of Cold air mass fraction (ε)

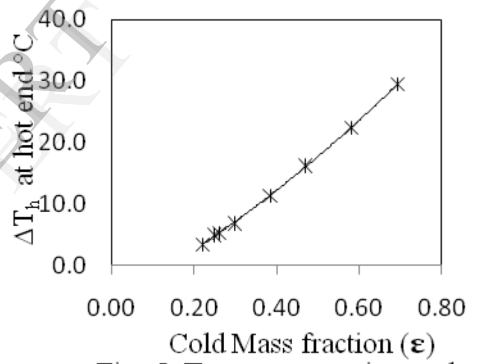

Fig. 5 Temperature rise at the hot end (ΔT_h) as a function of Cold air mass fraction (ε)

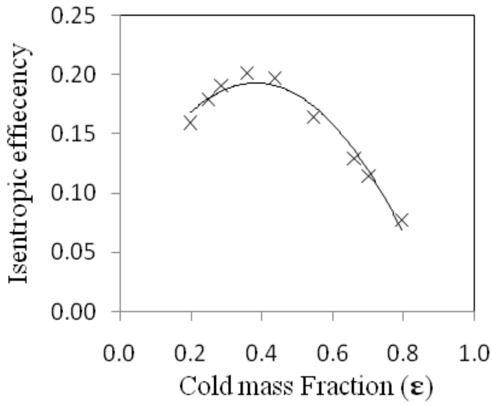

Fig.6 Isentropic effieciency as a function of cold air mass fraction (ε) at inlet pressure 4 bar

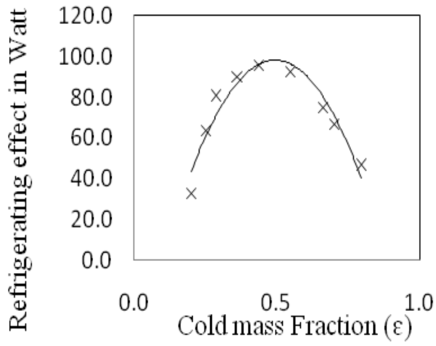

Fig .7 Refrigerating Effect as a function of cold air mass fraction (ε) at inlet pressure 4 bar

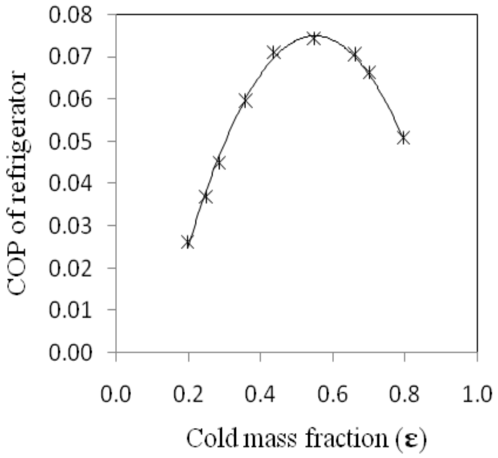

Fig. 8 COP as function of Cold air mass fraction (ε) at inlet pressure 4 bar

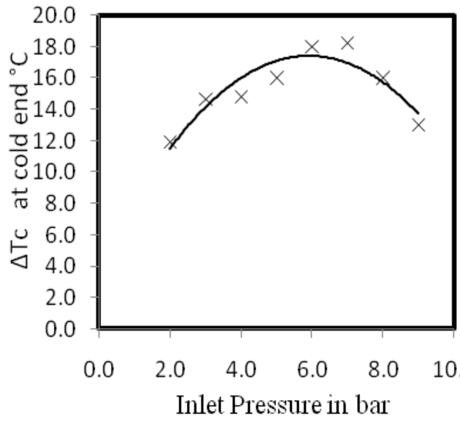

Fig.9 Temperature Drop at Cold end (°C) as a function of Inlet pressure (bar)

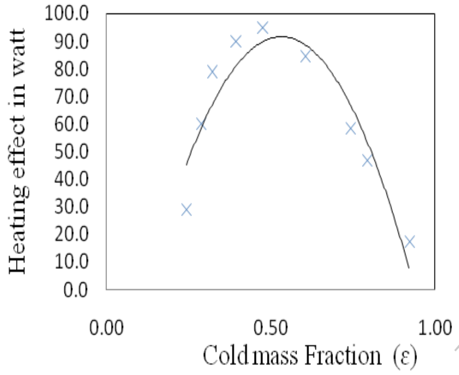

Fig.10 Heating effect of vortex tube is a function of cold mass fraction(ε) at inlet pressure 4 bar

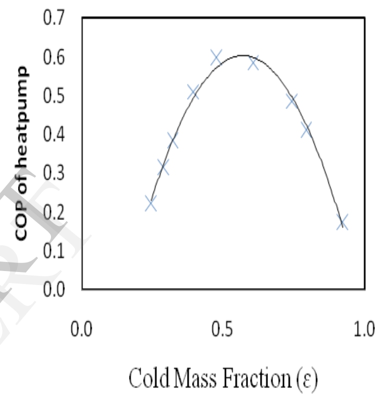

Fig.11 COP of Heat Pump as s a function of cold mass fraction(ε) at inlet pressure 4 bar

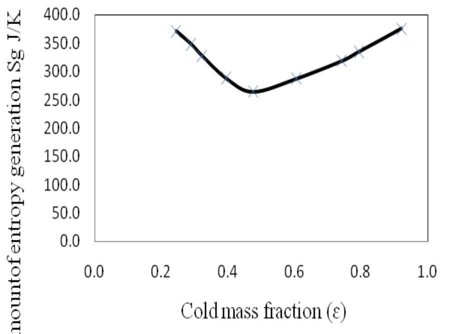

Fig.12 Amount of entropy generation Sg (J/K) as a function of cold mass fraction(ε) at inlet pressure 4 bar

Results and discussions:

The experiment was conducted to investigate the effect of the cold mass fraction on the cold air temperature drop, rise in hot air temperature, isentropic efficiency and the thermodynamic analysis was carried out to evaluate the performance of the vortex tube keeping inlet air pressure to the vortex tube constant at 4 bar.

Fig.4 shows the plot of temperature reduction in cold air to the cold mass fraction for input pressure of 4 bar. It was observed that the temperature reduction (ΔT_c) of the cold air increase with increase in cold mass fraction up to 0.4 and then the temperature reduction decreases. The maximum temperature drop recorded was 16°C.
Fig. 5 Shows the increase in hot air temperature as a function of cold mass fraction, it is observed that the increase/rise in temperature (ΔT_h) increases almost linearly with increasing in the cold mass fraction up to 0.8. and maximum rise in temperature recorded at **ε** of 0.8 is 29.8°C.
Fig. 6 shows the effect of cold air to the mass fraction on the isentropic efficiency of vortex tube. It follows the qualitatively the same trend as that of the drop in cold air temperature as it is the ratio of the actual cold air temperature drop to the isentropic drop in the temperature. The maximum isentropic efficiency varies between 0.08-0.2, and the maximum isentropic efficiency is found 0.2 for cold mass fraction of 0.4.
Fig. 7 shows the effect of the refrigerating effect as a function of cold mass fraction **ε**. It is observed that the refrigerating effect increase initially and then decrease after 0.5. The range in which the refrigerating effect is found to be effective lies between the cold mass fraction of 0.35-0.65 and the corresponding refrigerating effect is found 32.51 (W) to 95.781 (W) for the present experimental set of reading.
Fig. 8 shows the plot of COP of refrigerator as a function of **ε**. It is observe that the maximum COP is found to be 0.08.Here it is assumed that the work of compression required to increase the pressure from the exit (atmosphere) condition to the inlet pressure follows reversal isothermal process.
Fig. 9 shows the plot of drop in temperature of cold air as a function of increase in the inlet pressure keeping the valve opening fix, orifice diameter and tube diameter constant. It is observe the drop in the cold air temperature enhances linearly with the increase in the inlet pressure. It may be due to stronger vortex that is created on the inner periphery of the tube due to increase in the inlet pressure and the temperature separation is a function of the inlet pressure and the exit pressure.
Fig. 10 shows the effect of the Heating effect as a function of cold mass fraction **ε**. It is observed that the Heating effect increase initially and then decrease after 0.5. The range in which the Heating effect is found to be effective lies between the cold mass fraction of 0.32-0.475 and the corresponding Heating effect is found 79.04 (W) to 94.735 (W) for the present experimental set of reading.
Fig. 11 shows the plot of COP of Heat Pump as a function of **ε**. It is observe that the maximum COP is found to be 0.013.
Fig. 12 shows the effect of the Entropy generation as a function of cold mass fraction **ε**. It is observed that the Entropy generation for D =12 mm and air as working fluid .With increasing temperature differences increases and entropy generation decreases.

Conclusion: An experimental study on the temperature separation and thermodynamic analysis in the vortex tube has been carried out and these findings summarized are as follows:

1. The maximum temperature drop and temperature rise is found between 0.3-0.4 and 0.8 cold air mass fractions respectively.

2. The temperature drop increases with increase in inlet pressure.
3. The refrigerating effect is more effective when the cold air mass fraction lies between the 0.35-0.65.
4. The maximum COP on the vortex tube is found to 0.08 for inlet pressure of 4 bar and assuming the work of compression from exit to inlet pressure follows reversible isothermal process and the working medium obeys ideal gas laws.
5. The Heating effect is more effective when the cold air mass fraction lies between the 0.32-0.60.
6. The maximum COP of Heat pump is found to 0.013 for inlet pressure of 4 bar.
7. The results indicated that as hot tube length or inlet pressure increases the temperature difference increases too, The second law analysis shows that the For air, it is middle tube which generates lowest entropy.

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