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Effect of End Obstruction for optimum performance of Modified Vortex Tube with Dual Forced Vortex Flow

G.Maruthi Prasad Yadav[1], P.Mallikarjuna Reddy[2] and B. Durga Prasad[3]
[1](Associate Prof, Mech Engg Dept, St Johns College of Engg & Technology, Yemmiganur-518360, AP, India)
[2](Prof & Principal, Mech Engg Dept, AVR & SVR College of Engg & Technology, Nandyal-518503, AP, India)
[3](Prof & Head, Mech Engg Dept, Jawaharlal Nehru Technological University, Anantapur 515002, AP, India)

Abstract: *In the present work the cone-shaped valve at the hot end of existing model is replaced with a hollow cone-shaped valve gives a provision for passage through the inner core. A conical valve is introduced at the orifice, guide the forced vortex yet again to strike back evolve one more forced vortex flow, which escapes through the central core of the hollow cone-shaped valve. Thus the revised vortex tube with two forced vortex flows is named as dual forced flow vortex tube (DFFVT). The free vortex end zone, influence the convergent of air and govern the forced vortex flow model. Providing an obstacle at the hollow cone end boosts up the desired flow and improves the energy separation. In the present work, an effort is made to examine the consequence of end obstruction on the performance of modified vortex tube. A series of tubes with distinct level of obstacle were tested. The results imply that the temperature drop is effective with 0.305 end obstruction to tube area. Also the cold orifice diameter is varied for optimal performance of the tube.*

Keywords: *Vortex flow, COP, Temperature drop, End obstruction*.

I. Introduction

In vortex tube compressed air enters tangentially through nozzle acquire radial flow actuate towards the pipe end where it is converged and reversed by cone-shaped valve travels towards nozzle end and escapes through orifice. During this progress energy transfer takes place between the free and forced vortexes, results in hot air at periphery escapes through the valve end and cold air at core escapes through orifice. The schematic diagram presenting the flow pattern of the vortex tube is shown in fig

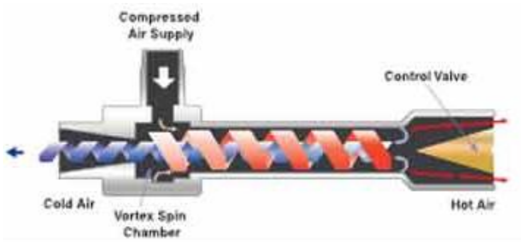


Fig.1: Schematic diagram flow pattern in a vortex tube

The vortex tube was find entirely by accident in 1933. George Ranque [1], a French physics scholar, was experimenting with a vortex-type pump he had developed when he noticed warm air exhausting from one end, and cold air from the other. Ranque soon forgot about his pump and begin a small firm to take advantage of the commercial possibility for this astonishing invention that generate hot and cold air with no moving parts. However, it quickly failed and the vortex tube slipped into obscurity until 1947 when Rudolph Hilsch [2], a German physicist, reveal a extensively read scientific paper on the device.

Ahlbornet al. [3] elaborate the temperature separation in a low pressure vortex tube and they stated the existence of secondary flow. Behera et al., [4] extend out a simulation of vortex tube using CFD. Aljuwayhel et al. studied the mechanism of stream and energy separation inside the vortex tube using renormalization group (RNG) k-e and standard k-e models. They used a two-dimensional axisymmetric model along with the outcome of the rotational velocity and re ported that the RNG k-e model predicts better the performance of the vortex tube [5]. Eiamsa-ard et al. experimentally studied the outcome of the nozzle numbers on the performance of a vortex tube [6]. Pinar et al. (7) examine the consequence of entrance pressure, nozzle number and fluid type factors on the tube vortex performance by means of Taguchi process.

Promvonge and Eiamsaard. (8) describe the outcome of the multitude of entrance tangential nozzles, the cold orifice diameter and the pipe insulations on the temperature diminution and isentropic effectiveness of the vortex tube. Skye et al. (2006) [9] procure the inlet and outlet temperatures in experimental and numeral form and compared them with each other. He used a standard two dimensional turbulence $k-\varepsilon$ model for simulating. [10-12] deliberate numerically the outcome of the length to diameter ratio (L/D) and stagnation point occurrence significance in flow model. Orhan and Muzaffer [18] have conduct out a sequence of experiments and disclose that the higher the entrance pressure, the major the temperature difference of the outlet flow. It was also shown that the cold fraction is an significant feature control the achievement of the energy separation in the vortex tube. Optimum values for the angle of the guide valve, the length of the tube and the diameter of the entrance nozzle were gain.

Eiamsa-ard and Promvonge [19] deliver a entire overview of the past investigations of the mean flow and temperature behaviours in a turbulent vortex tube in order to know the character of the temperature separation or Ranque–Hilsch outcome. They have intend optimal values for the cold orifice to the VT entrance diameter (d/D) of 0.5, the angle of the conical guide valve of 50 degrees, the length of the vortex tube to the VT entrance diameter (L/D) of 20 and the diameter of the inlet nozzle to the VT inlet diameters (δ/D) of 0.33 for air as the operation fluid.

Aydın and Baki [20]and Hamdan et al. [21] shown that inlet pressure and cold mass fraction were the most essential operant parameters. Behera et al. [22] describe numerically that the secondary circulation lower the performance of the vortex tube. Mohammad O. Hamdan [23, 24] imply an experimental study on the achievement of the vortex tube and found that pressure and cold fraction are influential parameters that effect the performance and also found that tangential entry nozzle gives highest energy separation.

Sarkar [25] analyzed vortex tube expansion transcritical CO_2 refrigeration cycle with two cycle layouts based on simple thermodynamic model and reveal moderate COP improvements. Xie et al. [26] and Liu an Jin [27] analyzed CO_2 trans-critical two level compression refrigeration cycle in vortex tube expanse and reported 2.4% to16.3% growth over the cycle with expansion value.

Literature revision disclose that there is no hypothesis so wholly, which gives the sufficient agreement of the vortex tube phenomenon as demonstrate by several researchers. Thus, much of the design and elaboration of vortex tubes has been supported on empirical correlations leaving much scope for optimization of judicious parameters.

In the existent vortex tube model, stagnation point arises in forced vortex flow that originate to growth of secondary flow again back towards the hot end, which arise in multi circulation closely the hot end. Thus providing a way to that secondary stream at the core, further improves the temperature separation. In this prospect, in the present study an innovator plan modification is accomplished by which the forced vortex flow at cold end is made to strike back again to form one more forced vortex flow. Thus, the modified vortex tube is called as dual forced flow vortex tube (DFFVT) comprise of three outlets: one hot outlet and two cold outlets (cold end-I and cold end-II). The schematic diagram of the modified vortex tube is shown in fig 2.

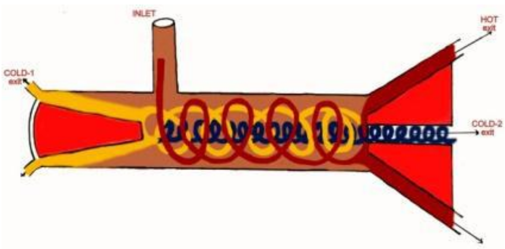


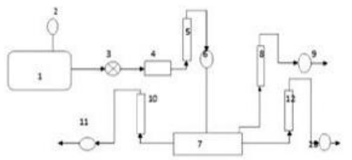
Fig 2: Flow pattern of the modified vortex tube

The objective of the present work is to introduce and study the effect of the hot exit opening provided with obstruction area at different levels, on the performance of a modified vortex tube with dual forced vortex flow.

II. Experimentation

The investigations unit consists of subsequent components: (a) inlet nozzle, (b) vortex chamber, (c) a tube, (d) hollow cone shaped valve for a way out at hot flow and cold flow-II (hot end and cold end-II) and (e) tapered valve at the cold end-I.

In the vortex tube (DFFVT) the air penetrate tangentially through the nozzle obtain screw-shaped flow on the passage to one end, choked-up and reversed by hollow cone-shaped valve, guide the pressure in the system. The reversed axial stream is forced to flow by forward vortex flow, moves towards the conical valve at the other end which is again convergent to the central core and traverse back as forced flow through the interior core of the hollow conical valve. Thus the modified vortex tube consists of dual forced vortex flow. Figures 2, 3 show a schematic plot of DFFVT and setup.



1. Compressor 2. Pressure gauge 3. Control valve 4. Pressure regulator.
5, 8, 10, 12. Rota meter. 6,9,11,13 Thermocouples. 7. Vortex Tube

Fig 3: Experimental setup

Initially compressor is operated for a certain time to attain steady state and then pressurized air at room temperature is made to enter the vortex tube tangentially passing through a pressure gauge and glass flow Rotameter to measure inlet pressure and flow rate. The Rotameter with uncertainty of 0.5LPM is used at the inlet. In addition, a k type thermocouple with 0.5^0C uncertainty is provided to record the inlet temperature. Due to tangential entry, the air attains swirling motion and travels towards partially opened hollow conical valve, where part of the air escapes through it as hot air and the remaining air converges and forced back in the opposite direction where a conical valve is arranged. Again, a part of air escapes through the opening of the cone valve as cold air and the still remaining air is converging to the core, travels back, and escapes through the inner core of a hollow conical valve as still colder air. Thus, the tube posses' three exits say; hot exit, cold exit-I and cold exit-II. At the hot exit and cold exit-I, the air is made to pass through Rotameter either with an uncertainty of 0.5LPM or 0.05LPM to measure the volumetric flow rate. When the flow is low, higher accuracy 0.05LPM Rotameter is used and when the flow is high, lower accuracy 0.5LPM Rotameter is used. Thermocouples with 0.5^0C uncertainty are arranged at all the three exits to measure the temperatures.

In the present work, DFFVT with a tube length to diameter (L/D =11) is tested. Though the optimum L/D = 9.3 [13] an extra length is given due to the arrangement of one more conical valve at the nozzle end. Throughout the study, the 3mm diameter nozzle is used. A provision is prepared at the other end to measure the hot gas temperature. The cold end-I outlet at the nozzle end has provided with a cylindrical piece to measure the temperature. Cold end-II orifice has threaded to the other end where the cold gas temperature is measured.

The test is repeated using a series of pipes without and with end obstruction at hot exit at the end of free vortex say end obstructer-I = 0.159 obstruction to tube area, end obstructer-II= 0.305 obstruction to tube area and end obstructer-III= 0.437 obstruction to tube area, different mass flows by varying the openings of conical plugs. Also the orifice size at cold end-II opening is varied with 3, 4 and 5mm diameter to test the performance of the tube.

III. Mathematical Analysis

The key governing parameters of the function of dual forced flow vortex tube are expressed as follows

Cold Fraction:

In this case we have two cold fractions (i) Cold fraction-I, ε_{c1} (ii) Cold Fraction-II, ε_{c2}

Cold Fraction-I is the ratio of air through cold end-I to the inlet air mass flow rates, whereas cold fraction-II is the ratio of air through the cold end-II to the inlet air mass flow rates.

$\varepsilon_{c1}=m_{c1}/m_i$

$\varepsilon_{c2}=m_{c2}/m_i$

Hot gas fraction, $\varepsilon_h = 1-\varepsilon_{c1}-\varepsilon_{c2}$

The cold gas temperature drop of the tube is expressed as:

$\Delta T_{c1} = T_i - T_{c1}$

ΔT_{c1} – Temperature drop at exit-I in ^{0}C

T_i - Temperature of inlet air in ^{0}C

T_{c1} – Temperature of cold outlet air at cold end-I in ^{0}C

$\Delta T_{c2} = T_i - T_{c2}$

ΔT_{c2} – Temperature drop at exit-I in ^{0}C

T_{c2} – Temperature of cold outlet air at cold end-II in ^{0}C

The temperature rise of the hot air tube is defined as:

$\Delta T_h = T_h - T_i$

T_h – Temperature of hot outlet air in ^{0}C

Analyzing the vortex tube process as isentropic expansion, Temperature at the exit is

$P_a = \text{Ambientpressure}$

$P_i = \text{Inlet Pressure}$

$\gamma = \text{Specific heat ratio of air}$

The efficiency of a refrigerator is expressed in terms of the coefficient of performance (COP) which is expressed as follow, [15]:

$$\mathrm{COP} = \frac{\text{Desired output}}{\text{Required Input}}$$

When vortex tube is used as a cooling device (cold stream is used), the device is called refrigerator and the COP is calculated by dividing the desired output (cooling load) on required input (compression energy). The compression energy is calculated for isothermal process (at constant temperature) which represents the ideal minimum compression work. While the cooling load is calculated for an ideal gas as showed below for COP [14] at end-I:

$$\mathrm{COP_{R1}} = \frac{\text{Cooling Load}}{\text{Isothermal Compression Energy}}$$

$$= \frac{m_c C_p (T_i - T_c)}{m_i R\, T_i \ln(p_i/p_a)}$$

$$\mathrm{COP_{R1}} = \frac{1}{r}\frac{\varepsilon_{c1}(T_i - T_{c1})}{T_i \ln \frac{p_i}{p_a}}$$

COP [14] of cooling process at the end-II is expressed as follows

$$\mathrm{COP_{R2}} = \frac{1}{r}\frac{\varepsilon_{c2}(T_i - T_{c2})}{T_i \ln \frac{p_i}{p_a}}$$

IV. Results And Discussions

4.1 Effect of Cold Fraction on temperature variation

4.1.1Temperature drop through cold end-II

Fig 4 shows the influence of cold fraction-II and hot fraction on the temperature drop at the end-II. Results show that the temperature drop-II increases with increase of cold fraction-II, attains a peak and decreases again thereafter. Max temperature drop in cold end-II is attained at higher cold fraction-II for hot fraction up to 0.51 whereas the same is obtained at lower cold fraction-II for hot fraction > 0.51. Results reveal that max temperature drop-II is attained at cold fraction-II ranges from 0.19 to 0.40 at all considered hot fractions.

The highest temperature drop is 18^0 at 0.6 hot fraction, gain at 0.24 cold fraction-II. But observe the measure and nature together 0.45 hot fraction with 0.34 cold fraction-II is the choose combination which gives a greatest temperature drop of 16^0. Though the temperature drop is higher at 0.6 hot fraction, it happen at lower cold fraction-II (ε_{c2}=0.2 to 0.24).

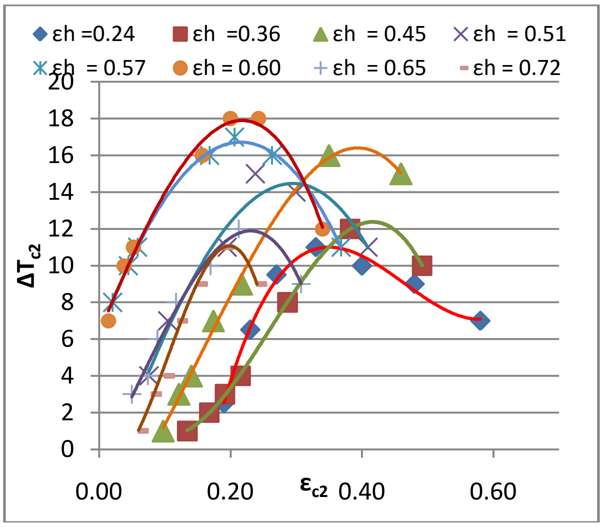


Fig 4: Effect of cold fraction-II on temperature drop through cold end-II for different hot fractions

Lower hot fraction, indicate the way through hot exit is restricted highly. Therefore superfluity air particles at the hot end sector get converged through hollow cone valve. But the converged air is in excess, cannot be directed fully in opposite flow at the intermediate core of the tube. Thus, part of the converged air penetrates in to the core and mixed up with the stream through the end-II that leads to increment in temperature of cold air. At higher hot fraction, only a small part of air is available for formation of forced vortex flow towards the nozzle end and thereby still less air is available at the core to form. There by the flow through cold end-II cannot be sustained that results in a decrease in temperature drop.

4.1.2 Temperature drop through cold end-I

Fig 5 shows the effect cold fraction-I on the temperature drop at the end-I for different hot fractions. The temperature drop increases with increase of cold fraction-I. Peak value of 10^0 temperature drop is obtained at a hot fraction of 0.36 with 0.49 cold fraction-I. Whereas at 0.45 hot fraction (corresponding to optimum performance through cold end-II), maximum temperature drop is 9^0 and obtained at 0.44 cold fraction-I (or 0.11 cold fraction-II). Hence optimal cold fraction for active achievement at the end-I is distinct from that of the optimal cold fraction for active accomplishment at the end-II. At a lower cold fraction-I efficiency of end-I is less because ample air is obtainable at the core towards the cold end-II ensue in higher transpose of energy from the second forced vortex stream at the core towards cold end-II to first forced vortex stream at midst core towards cold end-I, proceed in increase of temperature of air through the end-I.

Moderate hot fraction ranging from 0.35 to 0.5 fetters an active temperature drop through cold end-I. At lower hot fractions, fine amount of air is present at its boundary to take up the energy from the air at the core towards cold end-I. Whereas as at higher hot fraction the air towards cold end-I is small that gets disturbed by pressurized inlet air at the nozzle end, which also declines the performance of the tube through cold end-I.

Together at both ends-I and -II, 0.55 is the cold fraction respective to optimum performance, (0.34 ε_{c2} with a temperature drop of 16^0 at the end-II and 0.21 $\varepsilon c1$ with a temperature drop of 5.5^0 at the end-I), whereas it was only around 0.3 to 0.35 in earlier studies [22]. Hence cold fraction for the optimal temperature drop is better and higher to earlier reports. Still higher drops at the end-I is attained with too lower cold fraction-II, but subsequently it is not active at the end-II, which denote it performance almost all like equal to a vortex tube. Even then, it is better than a formal type vortex tube since in the vortex tube at higher cold fraction, the secondary transmission system which mix with hot air at the other end, whereas in this modified vortex tube, the same air escapes generously through the slot at the end-II. Comparing to stipulated vortex tube, DFFVT works with higher cold fraction together at both end-I and end-II.

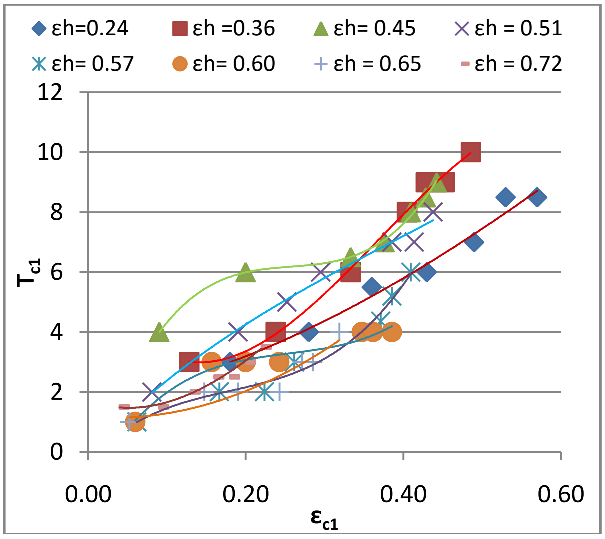


Fig 5: Effect of cold fraction-I on temperature drop through end-I for different hot fractions

4.1.3 Temperature rise through hot exit

Fig 6 shows the effect of ε_{c2} on temperature rise at different hot fractions. The temperature rise at hot end enhances with advance of cold fraction-II and the same tendency is observed at all hot fractions. For a firm hot fraction, higher cold fraction-I indicate lower cold fraction-II and vice versa. At a lower cold fraction-I, cooling achievement at the end-II is higher and at higher cold fraction-I cooling accomplishment at the end-I is higher-up. Therefore, independent of portion of air through the end-I and end-II the energy transpose to the boundary is higher and thereby temperature rise at the hot end is higher. But too higher hot fraction proceeds to the decay of temperature rise due to unavailability of abundant air at the core, which has to transfer the energy to the circumference.

At a lower cold fraction through end-II arise in a lower temperature rise, since most air passes through the end-I and thereby energy transfer principally take place only between the cold air through the end-I and forward running air towards hot outgoing. In addition, gentle rise is observed due to transpose of temperature of air through the end-II to the circumference. Therefore, at all hot fractions higher cold fraction-II fetters better results in a temperature rise. Higher cold fraction-II denote higher stream through the secondary flow that drive the air towards the wall that also increase the temperature rise. Also, it is observed a little bit decay in the temperature rise for too higher cold fraction-II (> 40%) since particles through end-II does not have free flight and it choose its way towards the hot end sector which diminish the temperature of hot air.

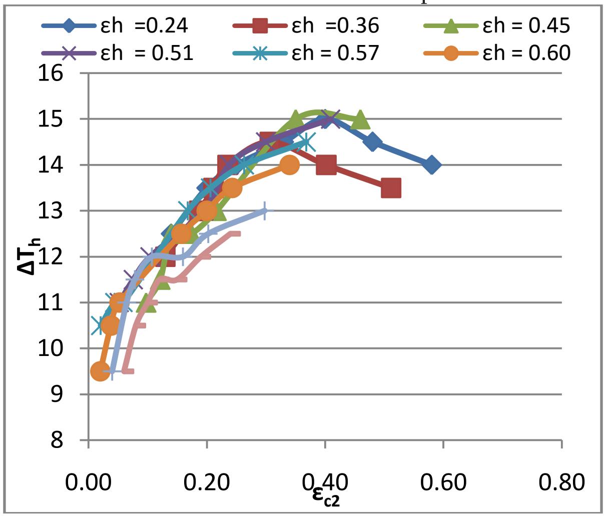


Fig 6: Effect of cold fraction-II on temperature rise for different hot fractions

4.2 Effect of cold fraction on Co-efficient of Performance (COP)

4.2.1 COP of the tube through cold end-II

Fig 7 shows the effect of cold fraction-II on COP_{R2} at end-II. It shows that the COP increases with increase of cold fraction. The COP is higher for higher hot fractions at low cold fraction. For emend cooling higher COP at higher cold fractionate is required. But it is observed that at higher hot fractions the COP lessen for $\varepsilon_{c2} > 20\%$. This is due to the shutdown of solid cone which stop the flow of the first forced vortex stream that contribute radial inwards near the hollow cone end which disorder the stream through end-II. Also for higher cold fraction-II (indicate lower cold fraction-I), the current through the solid cone outgoing is retard which abstain the flow through cold end-I. This cause the air particles that approach the solid cone to scatter radial outwards which in turn penetrate the band of entrance air and disorder the whole stream model at the entering.

Whereas at frowning hot fractions the COP is commendable at higher cold fractions particularly 0.45 hot fraction fetters the maximum COP of 0.0711 at 0.45 cold fraction-II. With low hot fraction, the stagnation point move towards the nozzle end axially and stretches towards the core which advance the firm secondary stream that escapes through cold end-II. Also at too higher cold fraction-II, the passage through end-II is ample and if the opening at the end-II is not furnish free evade the superfluity flow penetrate the hot band at the circumference which in turn disorder the require flow model. This is recognized in the results that COP begin diminish at higher cold fraction-II for all observe hot fractions. Cold fraction-II of 0.45 renders optimal performance of the tube through cold end-II. This is due to the provision of correct free escape of the flow through end-II.

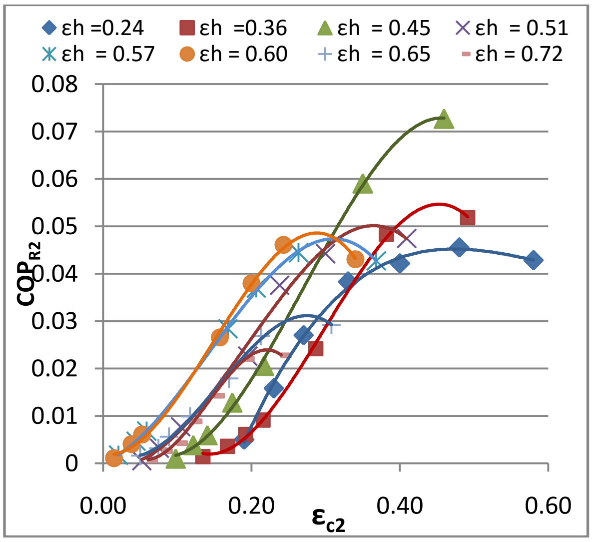


Fig 7: Effect of cold fraction-II on COP at cold end-II for different hot fractions

4.2.2 COP of the tube through cold end-I

Fig 8 shows the effect of cold fraction-I on the COP_{R1}. It is observed that COP increases with increase of cold fraction. The tendency of COP deviation is same at all examine hot fractions. Results show that the COP increments at slow rate at low cold fraction and the same took at a sharp rate at higher cold fractions. Because, at low cold fraction-I (higher cold fraction-II), the energy transfer from the second forced vortex to first forced vortex is higher that induce to decay in temperature drop at end-I. So the COP at low cold fraction-I increases at a slow rate. At the other side at higher cold fraction through end-I denote low cold fraction at end-II. Energy transpose from second forced vortex to first forced vortex is low, so the temperature drop in end-I is more effectual and COP enhance with fast rate. The highest COP of end-I gain is 0.0512 at 0.36 hot fraction and 0.49 cold fraction-I.

No noticeable vary in act of the tube through cold end-I is observed even with alteration of hot fraction. Because, at low hot fraction ample air is present at the core for the forming of second forced vortex stream which arise in higher energy carry over to the air at midst core towards cold end-I. So, though even energy on-pass from first forced vortex stream at middle core to the boundary is superior, finally ensue in low temperature drop and henceforth low COP. Whereas at the higher hot fraction, the air at the end of free vortex flow escapes generously through the slot of hollow cone at the periphery, which results in formation of weak forced vortex flow towards cold end-I and proceed in low COP.

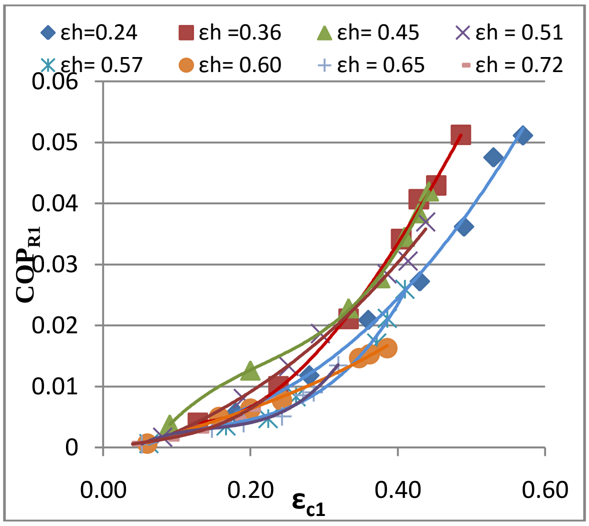


Fig 8: Effect of cold fraction-I on COP at cold end-I for different hot fractions

4.3 Effect of end obstruction through hot exit

Secondary circulation flow rise to be the characteristics of cold end orifice diameter [16] as accustomed by our studies and could be a performance debasing mechanism. At higher hot fraction stagnation point [16] occurs near the hot end, on the other side, it occurs at the nozzle end for lower hot fraction. Also, earlier studies reveal that stagnation point stretches towards the axis when it occurs near the nozzle end and expands radially as moves away from the nozzle. In the existing model of vortex tube the secondary flow gives rise to swirl loop flow near hot end [3] that disturbs the desired vortex flow. The dilapidation could be due to the relocate of colder fluid elements near the cold end exit zone in the course of the swirling secondary loop to the warmer flow region compel a decrease in the hot-end temperature and the transfer of warmer flow elements back to the cold end exit zone induce an increase in cold end exit temperature of the flow.

Stagnation point is the initiation for the present modified vortex tube and entity of stagnation execute the essential role in the formature of require vortex flow and thereby to procure efficient temperature separation. Keeping stable all the parameters, alteration of flow rates through hot and cold outgoing influence the event of the stagnation point and thus the growth of secondary flow. Especially the flow band at the end of free vortex, converging and guide to form forced vortex flows is decisive.

In the present modified vortex tube, the existence of secondary flow is utilized for further higher level of temperature separation. Here for higher temperature drop the secondary flow should originate towards the nozzle end and thereby can get enough time for energy carry over. However, again, it is observed that too close to the nozzle ensue in mixing up of cold air through the end-I with inlet air and disturbs the flow. Also the secondary flow cannot be continued in move far distance, due to its strong circumference stream. Therefore, moderate hot fraction renders higher temperature drops.

Thus to intensify the convergent flow at the hot exit, an obstacle is furnish by the abrupt blockage (diminishing the hot outgoing diameter). Through this the stream particles near the wall get blocked and addressed on to the tapered surface of the hollow cone valve, which thereby directed in the opposite order (towards the nozzle end) at the middle centre to form the require flow model. Also abrupt choking at the hot outgoing ensue in pressure raise up and assist the shifting of stagnation point towards solid cone end. Fig 9 discloses the outcome of cold orifice diameter at end-II on the temperature drop through end-II with and without end obstacle (say without end obstruction and with end obstruction of 0.159, 0.305 & 0.437) at the opening of hot exit. It is observed that temperature drop enhance by providing fine obstruction at the periphery of the hot exit and diminish again with too much extend in end obstruction area for observe cold orifice diameters. Compare to zero end obstruction, 0.305 end obstruction results in advance of temperature drop by 73.6%, 44.3% & 19.4% using end-II orifice diameter of 3mm, 4mm & 5mm. Also it is observed that increase of cold orifice diameter initially improves the temperature drop and diminish with extend of orifice diameter beyond 4mm.

Existence of stagnation point [3] closely hot end also cause the same stagnation point to expand towards the wall and precedence to the shaping of second forced vortex flow with higher diameter and vice versa. If the diameter of second forced vortex flow is higher than the diameter of opening at end-II, the transpose of colder fluid elements near the cold end exit band through the swirling secondary loop penetrate in to the warmer flow region causing decay in the hot end temperature. Where as in reversal casing with the diameter of second forced vortex flow lower than the diameter of orifice at end-II, the transfer of warmer stream elements back to the cold end-II outgoing band causing an increase in cold end-II exit temperature of the flow and also the air particles retard by hollow cone at the periphery escapes instantly through end-II without taking part in temperature separation which finally degrades the performance of tube. This is confirmed with the results showing again declination in temperature drop using 0.437 end obstruction compared to 0.305 & 0.159 end obstructions. So finally the cold end-II orifice diameter that provides sufficient passage (equal to the diameter) second forced vortex flow gives the effective temperature drop. Providing 0.305 end obstruction and increasing end-II orifice from 3mm to 4mm leads to increase of temperature drop by 24.2%.

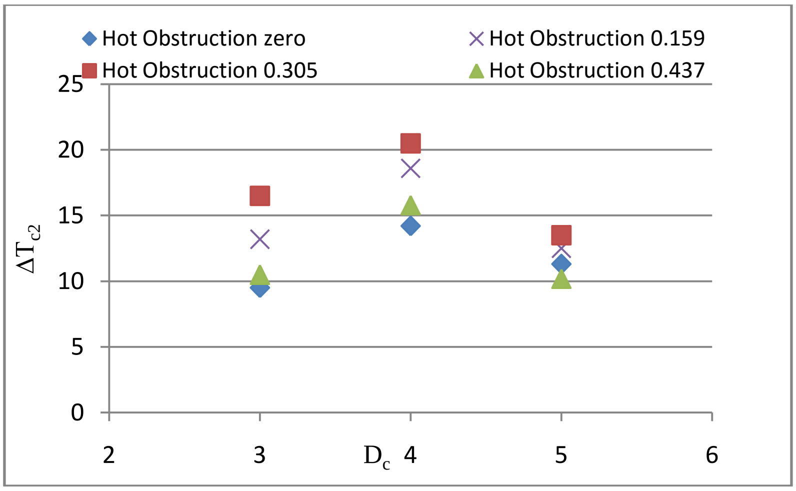


Fig 9: Effect of orifice diameter on temperature drop-II for different hot end obstruction

Either smaller or higher orifice diameter (Dc) results in poor performance because smaller orifice diameter tends the excess cold air in the passage through cold end-II, to enter the hot zone whereas higher orifice makes the air from free vortex to escape through without taking part in energy separation. So obviously the moderate orifice diameter gives the maximum temperature drop. Too higher orifice diameter at end-II minimizes the chances of development of secondary flow [17] and there by the DFFVT does not perform effectively.

V. Conclusions

The following conclusions were drawn by conducting series of experiments on the performance of the dual forced flow vortex tube (DFFVT) under variable cold fraction, end obstruction and several cold orifice diameters.

The experimental results reveal that the temperature reduction of the cold air of the dual forced flow vortex tube is substantially influenced by the cold mass fraction. Temperature drop at the end-II increases with increase of cold fraction-II initially, attains a peak value in the range 0.19 to 0.4 cold fraction-II and the trend reverses beyond that for all considered hot fractions. Maximum temperature drop through cold end-II of 18^0 is obtained at 60% hot fraction with 24% cold fraction-II.

Temperature drop through cold end-I increases with increase of cold fraction-I. Maximum temperature drop of 10^0 is obtained at 36% hot fraction and 49% cold fraction-I (15% cold fraction-II). The mass fraction for optimum performance through the both cold end-I and II is not same. Optimum performance together at the both the ends yields a maximum temperature drop of 16^0 through cold end-II with 6^0 through cold end-I for 45% hot fraction with 34% cold fraction-II and 21% cold fraction-I. Hence with the proposed modification the cooling capacity is improved by enhancing the cold fraction to 55% (together at end I & II).

Orifice diameter of 4mm is the optimum in attaining higher temperature drop and co-efficient of performance.

Providing end obstruction, certainly influence the tube performance in attaining higher temperature separation. Providing an end obstruction of 0.305 leads to increase of temperature drop by 44.3% compared to the tube performance without end obstruction. An end obstruction of 0.305, obstruction area to tube area results in higher temperature drop of 20.05^0.

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