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Research Article

Effect of Cold Fraction and Orifice Diameter on the Performance of Modified Vortex Tube with Dual Forced Vortex Flow

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Abstract

Pressurized gas is tangentially expanded to create a high swirling motion in the vortex tube, the flow of gas splits in two parts: a free vortex as the peripheral warm stream and a forced vortex as the inner cold stream. Through certain design modification, the forced vortex flow is made to strike back through the core results in formation of one more forced vortex flow. Thus the modified vortex tube with two forced vortex flows is known as dual forced flow vortex tube (DFFVT). In the present work an attempt is made to analyze the effect of cold fraction through ends-I & II and orifice diameter at cold end-II on the performance of modified vortex tube. Series of cold orifice with different diameters at cold end-II are used for experimentation and investigated for higher temperature drops and effective performance (COP).

Keywords: *COP, cold orifice, temperature drop, cold fraction*

1. Introduction

Vortex tube is a simple device that splits compressed air into hot and cold streams. The vortex tube was invented in 1933 by French physicist George J. Ranque and then improved by Hilsch in 1947. The construction details of a vortex tube are shown in fig 1. When high pressure enters through tangential nozzle a strong vortex flow created that splits in to two streams: A warm stream escapes through conical valve at periphery and a cold stream at inner core escapes through central orifice.

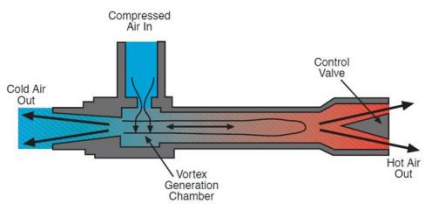

Fig 1: Schematic diagram representing flow field in vortex tube

(Sachin.*et al*, 2009) carried investigation to find the optimum cold orifice diameter of the vortex tube so that the flow can escape through the openings of the tube freely with higher temperature drops. (GurolOnal*et al*,2013) did experimental investigation to study the performance of the counter flow vortex tube with inner threaded body so as to improve the flow pattern through guiding ways at the inner surface and thereby to improve the performance of the tube. (Takahama and Yokosawa, 1981) examined the possibility of shortening the vortex tube chamber length by introducing diverging section for the vortex chamber. (Mohammad O. Hamdan*et al*, 2011 & 2012) carried experimental study on the performance of the vortex tube and found that pressure and cold fraction are important parameters that effect the performance and also found that tangential entry nozzle gives maximum energy separation. (S. Eiamsa-ard*et al*, 2010) carried investigation on vortex tube to promote its energy separation performance by cooling hot tube of vortex tube through water jacket arrangement. (KiranDevade*et al*, 2014) conducted experimental tests with series of cold orifice and hot end plugs with different dimensions to find the optimum combination for higher energy separation. (Y.T.Wu*et al*, 2007) proposed a new design for nozzle with equal gradient Mach number to reduce flow loss and also introduced a diffuser to reduce friction loss thereby to improve the performance of the tube (UpendraBehera*et al* 2008). investigated the flow behavior and energy separation in vortex tube. He studied the distribution of pressure and velocity throughout the length of the tube and

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there by suggested suitable working conditions for higher temperature drops. (Harnett and Eckert, 1957) suggested that invoked turbulent eddies are responsible for the temperature separation mechanism in the vortex tube. (HossenNezhad and Shamsoddini, 2009) carried three dimensional analysis of the mechanism of flow and heat transfer in a vortex tube.
(Pongjet and smith, 2005) investigated the vortex thermal separation in a vortex tube refrigerator, using two different tubes, insulated and non-insulated. (Eiamsa-ard and promvonge, 2007) and (Aljuwayhel, 2005) conducted experimental studies to investigate the phenomena of energy separation. They used the snail entrance method to inject the compressed air into the vortex chamber. They found that this method improved the cold side temperature drop and thereby improved the efficiency of the vortex tube as well.Arjomandi and (Yenpeng*et al*, 2007) used new hot end plug which improved the performance of vortex tube. (Behera*et al*, 2005) carried simulation of vortex tube using CFD for optimum parameters. (AbdolrezaBramo and Pourmahmoud, 2001) carried an investigation to study the effect of length to diameter on the performance of vortex tube. (Nader Pourmahmoud*et al*,2012) studied on energy separation and flow field behaviour of a vortex tube by utilizing helical nozzles. (Behera.u and p.j.paul, 2005) carried both CFD and experimentation to optimize the parameters of vortex tube. (HossenNezhad and Shamsoddini, 2009) carried three dimensional analysis of the mechanism of flow and heat transfer in a vortex tube. More references can be found in (Eiamas-ard*et al* 2008) which reviewed extensively Ranque-Hilsch effects in vortex tubes.

Literature review reveals that there is no theory so perfect, which gives the satisfactory explanation of the vortex tube phenomenon as explained by various researchers. Thus much of the design and development of vortex tubes have been based on empirical correlations leaving much scope for optimization of critical parameters.

In the present study an innovative design modification is implemented by which the forced vortex flow at cold end is made to hit back again to form one more forced vortex flow. Thus, the modified vortex tube is named as dual forced flow vortex tube (DFFVT) consists 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.

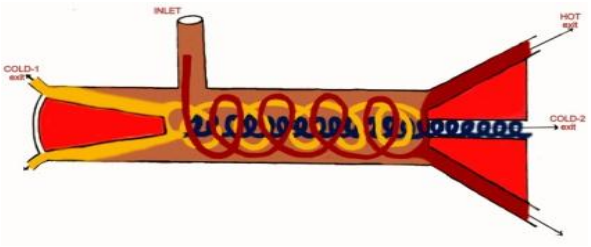


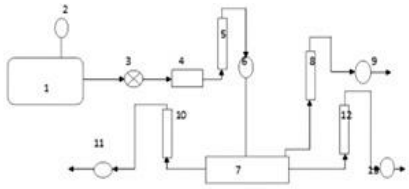
Fig 2: Flow pattern of the modified vortex tube

The objective of the present work is to introduce and study the effect of orifice opening at cold end-II on the performance of a modified vortex tube with dual forced vortex flow.

2. Experimentation

The investigational 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 enters tangentially through the nozzle attains spiral flow on the way to one end, choked-up and reversed by hollow conical valve, controls the pressure in the system. The reversed axial flow is forced to flow by forward vortex flow, moves towards the conical valve at the opposite end which is again converging to the central core and travels back as forced flow through the inner core of the hollow conical valve. Thus the modified vortex tube consists of dual forced vortex flow. Figures 2 and 3 shows a schematic plan 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 0.50C 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 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 0.50C 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 (Mohammad O. Hamdan*et al* 2011) 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 have 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 orifices at cold end-II with different diameters say 3mm, 4mm and 5mm and different mass flows by varying the openings of conical plugs.

3. 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, where as 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
T_i - Temperature of inlet air
T_{c1} – Temperature of cold outlet air at cold end-I
$\Delta T_{c2} = T_i - T_{c2}$
ΔT_{c2} – Temperature drop at exit-I in
T_{c2} – Temperature of cold outlet air at cold end-II
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
P_a = Ambientpressure
P_i = Inlet Pressure
γ = 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, (Cengel Y, Boles M *et al*, 2007):

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

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 minimum ideal compression work. While the cooling load is calculated for ideal gas as shown below for COP (ChengmingGao*et al*, 2005) at end-I:

$$COP_{R1} = \frac{\text{Cooling Load}}{\text{Isothermal Compression Energy}}$$
$$= \frac{m_c C_p (T_{in} - T_c)}{m_{in} R\, T_{in} \ln\left({}^{p_{in}}/_{p_a}\right)}$$
$$COP_{R1} = \frac{1}{r}\frac{\varepsilon_{c1}(T_{in} - T_{c1})}{T_{in} \ln \frac{p_{in}}{p_{c1}}}$$

COP (ChengmingGao*et al*, 2005) of cooling process at the end-II is expressed as follows

$$COP_{R2} = \frac{1}{r}\frac{\varepsilon_{c2}(T_{in} - T_{c2})}{T_{in} \ln \frac{p_{in}}{p_a}}$$

4. Results and Discussion

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. It is clear from the results that temperature drop at the end-II increases with increase of cold fraction-II initially, attains a peak value and the trend reverses beyond that. Max temperature drop of 11^0, 12^0, 16^0, 15^0, 17^0, 18^0, 12^0 and 11^0 is obtained at a hot fraction of 0.24, 0.36, 0.45, 0.51, 0.57, 0.6, 0.65 & 0.72. 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 more than 0.51. Results reveal that max temperature drop at cold end-II is attained at cold fraction-II ranges from 0.19 to 0.40 at all hot fractions.
The maximum temperature drop is 18^0 at 0.6 hot fraction, which is obtained at 0.24 cold fraction-II. But considering the quantity and quality together 0.45 hot fraction with 0.34 cold fraction-II is the preferred combination which gives a maximum temperature drop of 16^0. Though the temperature drop is higher at 0.6 hot fraction, it occurs at lower cold fraction-II (ε_{c2}=0.2 to 0.24).

At lower hot fraction, air particles at the hot end zone gets converged to the core and get mixed up with the flow through the end-II that leads to increase 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 decrease in temperature drop.

At higher hot fraction stagnation point (Sachin U. Nimbalkar*et al*, 2009) 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. Stagnation point is the initiation for the present modified vortex tube and existence of stagnation plays the key role in the formation of desired vortex flow and thereby to obtain effective temperature separation. Keeping fixed all the parameters, variation of cold fraction affects the occurrence of stagnation point and thus the development of secondary flow.

In the existing vortex tube model, the occurrence of stagnation point and thereby the development of secondary flow should be towards the hot end to reduce the extent of mixing of cold air elements and hot air elements (multiple circulation near hot zone), which in turn declines the performance of the tube. Thus, it is possible to reduce the effect of the existence of secondary flow but cannot avoid completely. 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 initiate towards the nozzle end and thereby can get enough time for energy transfer. However, again, it is observed that too close to the nozzle results in mixing up of cold air through the end-I with inlet air and disturbs the flow. Therefore, moderate hot fraction yields higher temperature drops.

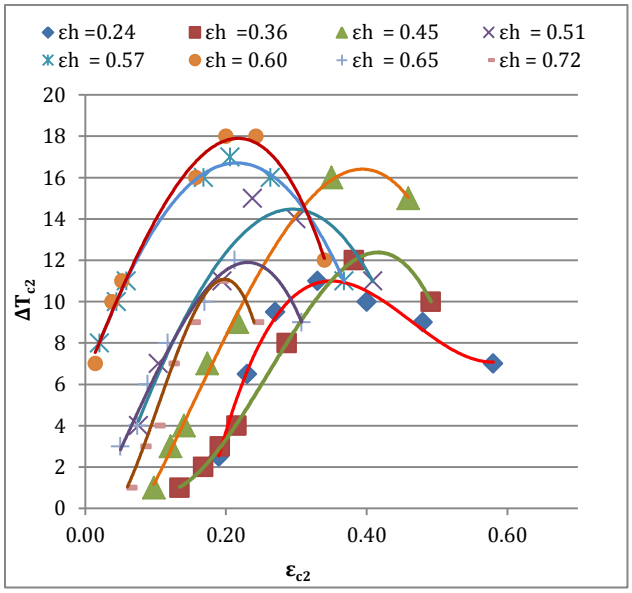


Fig 4: Effect of cold fraction-II on temperature drop through cold end-II for different hot fractions

4.1.2 Temperature drop through cold end-I

Fig 5 shows the effect cold fraction-I on temperature drop at the end-I for different hot fractions. The temperature drop increases with increase of cold fraction-I (GurolOnal*et al*, 2013). Higher temperature drop attained is found to be 8.5^0, 10^0, 9^0, 8^0, 6^0, 4^0, 4^0 and 3.5^0 at the hot fraction of 0.24, 0.36, 0.45, 0.51, 0.57, 0.6, 0.65 &0. 72. 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 optimum cold fraction for effective performance at the end-I is different from that of optimum cold fraction for effective performance at the end-II. At a lower cold fraction-I efficiency of end-I is less because abundant air is obtainable at the core towards the cold end-II results in higher transfer of energy from the second forced vortex flow at the core towards cold end-II to first forced vortex flow at middle core towards cold end-I, result in increase of temperature of air through the end-I.

Moderate hot fraction ranging from 0.35 to 0.5 gives an effective temperature drop through cold end-I. At lower hot fractions, small quantity of air is available at its periphery 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.

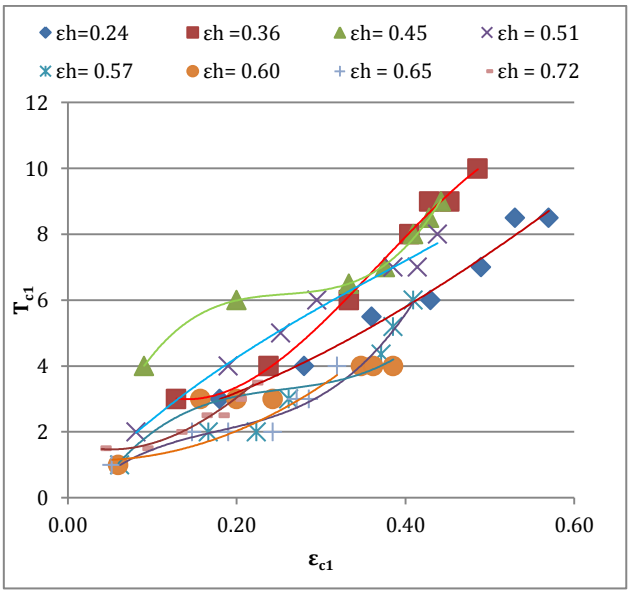


Fig 5: Effect of cold fraction-I on temperature drop through end-I for different hot fractions

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 ε_{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 (Y.T.Wu*et al*, 2007, UpendraBehera*et al*, 2008, Kalal M 2008, AlirezaHosseinNezhad, 2009). Hence cold fraction for the optimum temperature drop is enhanced and superior to earlier reports. Still higher drops at the end-I is attained with too lower cold fraction-II, but

subsequently it is not effective at the end-II, which means it works almost all like equivalent to vortex tube. Even then, it is better than a conventional type vortex tube because in the vortex tube at higher cold fraction, the secondary circulation forms which mix with hot air at the other end, whereas in this modified vortex tube, the same air escapes freely through the opening at the end-II. Comparing to conventional vortex tube, DFFVT works with higher cold fraction together at both end-I and end-II.

4.1.3 Temperature rise through hot exit

Fig 6 shows the effect of ε_{c2} on temperature rise at different hot fractions.

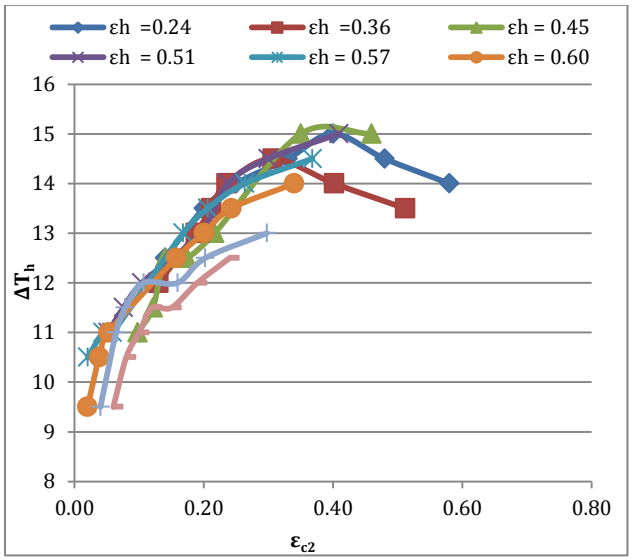


Fig 6: Effect of cold fraction-II on temperature rise for different hot fractions

The temperature rise at hot end increases with increase of cold fraction-II and the same trend is observed at all hot fractions. For a fixed hot fraction, higher cold fraction-I means lower cold fraction-II and vice versa. At a lower cold fraction-I, cooling performance at the end-II is superior and at higher cold fraction-I cooling performance at the end-I is superior. Therefore, irrespective of fraction of air through the end-I and end-II the energy transfer to the periphery is higher and thereby temperature rise at hot end is superior. But too higher hot fraction leads to the declination of temperature rise due to unavailability of ample air at the core, which has to transfer the energy to the periphery. At a lower cold fraction through end-II results in lower temperature rise because majority air passes through the end-I and thereby energy transfer mainly take place only between the cold air through the end-I and forward moving air towards hot exit. In addition, slight rise is observed due to transfer of temperature of air through the end-II to the periphery. Therefore, at all hot fractions higher cold fraction-II gives better results in temperature rise. Higher cold fraction-II means higher flow through secondary flow that pushes the air towards the wall that also enhances the temperature rise. Also, it is observed a little bit declination in the temperature rise for too higher cold fraction-II (> 40%) because particles through end-II does not have free escape and it takes its way towards hot end zone which decreases the temperature of hot air.

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.

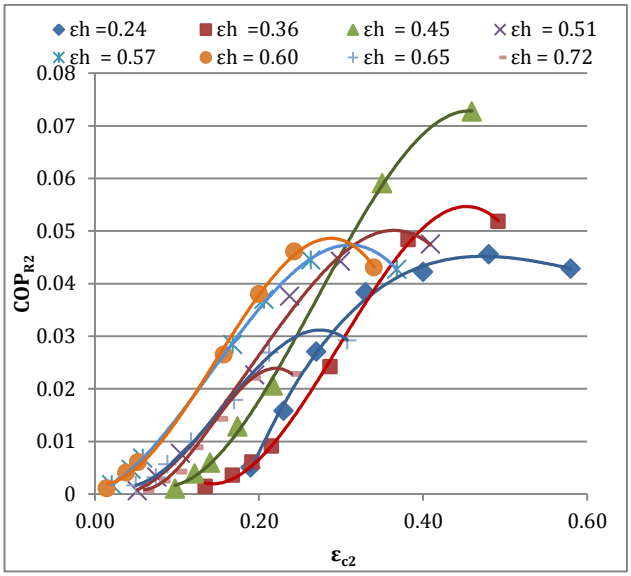


Fig 7: Effect of cold fraction-II on COP at cold end-II for different hot fractions

It shows that the COP increases with increase of cold fraction. The COP is higher for higher hot fractions at low cold fraction. For better cooling higher COP at higher cold fraction is needed. But it is observed that at higher hot fractions the COP declines for ε_{c2} > 20%. This is due to the closing of solid cone which obstructs the flow of the first forced vortex flow that tends radial inwards near the hollow cone end which disturbs the flow through end-II. Also for higher cold fraction-II (means lower cold fraction-I), the flow through the solid cone exit is obstructed which ceases the flow through cold end-I. This makes the air particles that approach the solid cone to disperse radial outwards which in turn enters the zone of inlet air and disturbs the entire flow pattern at the entrance. Whereas at lower hot fractions the COP is good at higher cold fractions especially 0.45 hot fraction gives the highest COP of 0.0711 at 0.45 cold fraction-II. At low hot fraction, the stagnation point shifts towards the nozzle end axially and stretches towards the core which enhances the strong secondary flow that escapes

through cold end-II. Also at too higher cold fraction-II, the flow through end-II is abundant and if the orifice at the end-II is not provided free escape the excess flow enters the hot zone at the periphery which in turn disturbs the desired flow pattern. This is identified in the results that COP starts declines at higher cold fraction-II for all considered hot fractions. Cold fraction-II of 0.45 yields optimum performance of the tube through cold end-II. This is due to provision of exact free escape of the flow through end-II.

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 trend of COP variation is same at all considered hot fractions. Results shows that the COP increases at slow rate at low cold fraction and the same took at rapid rate at higher cold fractions. Because at low cold fraction-I (higher cold fraction-II), the energy transfer from second forced vortex to first forced vortex is higher that leads to decrease in temperature drop at end-I. So the COP at low cold fraction-I increases at slow rate. At the other side at higher cold fraction through end-I means low cold fraction at end-II. The energy transfer from second forced vortex to first forced vortex is low, so the temperature drop at end-I is more effective and COP increases with rapid rate. The maximum COP at end-I obtained is 0.0512 at 0.36 hot fraction and 0.49 cold fraction-I.

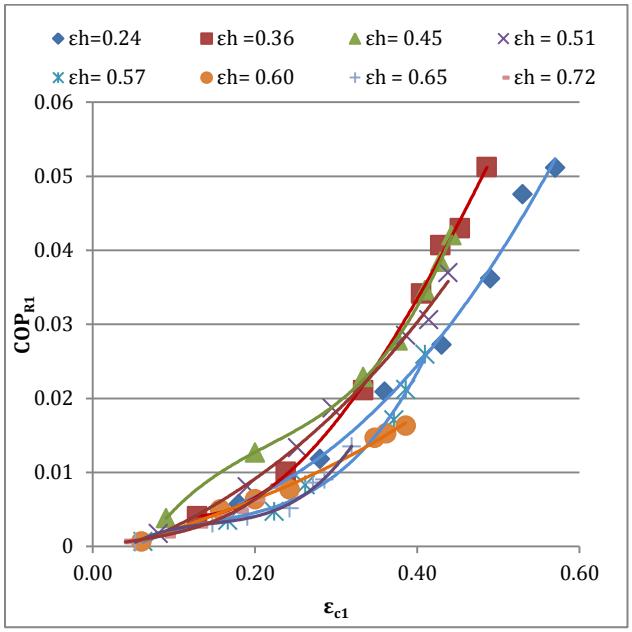


Fig 8: Effect of cold fraction-I on COP at cold end-I for different hot fractions

No remarkable change in performance of the tube through cold end-I is observed even with variation of hot fraction. Because, at low hot fraction abundant air is available at the core for the formation of second forced vortex flow which result in higher energy transfer to the air at middle core towards cold end-I. So, though even energy transfer from first forced vortex flow at middle core to the periphery is superior, finally result in low temperature drop and henceforth low COP. Whereas at the higher hot fraction, the air at the end of free vortex flow escapes freely through the opening of hollow cone at the periphery, which result in formation of weak forced vortex flow towards cold end-I and result in low COP.

4.3 Effect of the orifice diameter through cold end-II

4.3.1 Effect of the pressure

Effect of pressure on temperature drop-II using different orifice diameter is shown in fig 9. The temperature drop at end-II increases with increase of inlet pressure (Mohammad O. Hamdan*et al*, 2011 & 2012) for all cold orifice diameters at end-II. It is found that temperature drop for Dc is optimum in getting higher temperature drops. The maximum temperature drop is found to be 17^0, 25.5^0 & 13.5^0 using orifice diameter of 3, 4, and 5mm at 6bar pressure. The radii of second forced vortex is depended on mass fractions through hot end and end-I. Conducting series of tests with different mass fractions through hot and cold ends; it is found that 0.45 hot fraction with 0.34 cold fraction-II is the optimum.

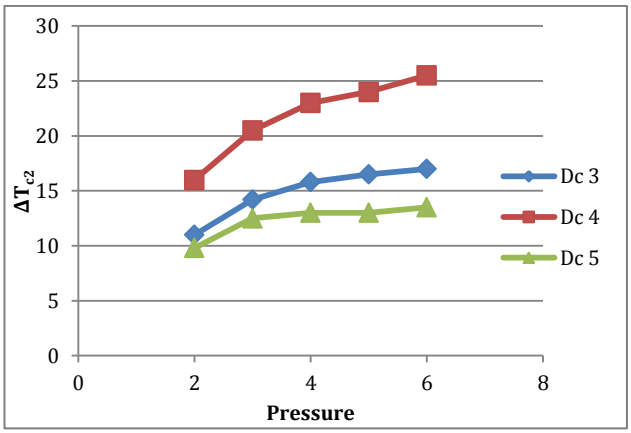


Fig 9: Effect of pressure on Temperature drop through cold end-II using different orifice diameter

At higher pressure, air particles compresses against the wall due to centrifugal force that acts radial outwards, which result in the increase of temperature of the air at the periphery. Air at the core gets colder as it expands. Also higher inlet pressure outcome with closely packed air particles in the tube, so amount of air particles that took part in energy separation increases which in turn improves the temperature drops and vice versa.

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 (KiranDevade*et al*, 2013) and there by the DFFVT does not perform effectively.

4.3.2 Optimum temperature drop and COP through cold end-II

Effect of cold fraction-II on temperature drop-II for different orifice diameters is shown in fig 10. The level of accuracy reached in predicted flow pattern also depends on the orifice of cold end-II. The temperature increases with increase of cold fraction initially up to certain extend and decreases beyond that (S. Eiamsa-ard*et al*, 2010, Y.T.Wu*et al*, 2007). This response is similar for all the considered orifice diameters say 3mm, 4mm and 5mm. The diameter of orifice should be sufficient to make a way for the air of second forced vortex flow to escape through. Results show the same that smaller diameter gives the maximum temperature drop at low cold fraction-II and vice versa. Maximum temperature drop of 15^0, 18^0, and 13.5^0 is obtained using cold orifice of 3, 4 and 5mm at end-II.

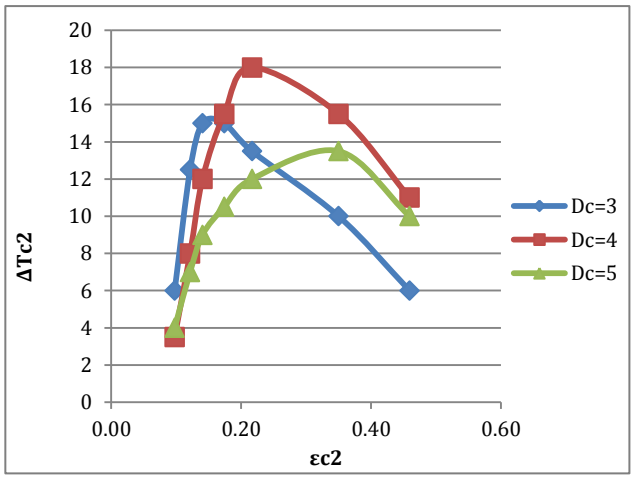


Fig 10: Effect of cold fraction-II on temperature drop through cold end-II for different orifice diameter

Small orifice diameter at cold end-II cannot provide way through for second forced vortex for free escape (Sachin U. Nimbalkar*et al*, 2009). Also large diameter gives excess passage which in turn makes the forward moving free vortex to exit through, without taking part in energy separation. Thus the diameter of orifice should be enough size for exact passage of second forced vortex flow. The same is conformed to the obtained results, that small diameter gives maximum drop in temperature at low mass fraction whereas the large diameter gives the maximum temperature drop at high mass fraction.

At a fixed hot fraction increasing of cold fraction-I decreases the cold fraction-II this in turn diminishes the amount of air particles that take part in energy separation. At the other side closing the solid cone valve (decreasing cold fraction-I), displaces the stagnation point towards the hollow cone and stretches radial outwards. Hence displacing stagnation point towards hollow end decreases the length of second forced vortex flow that in turn reflects on temperature drop declination. So, moderate cold fraction-II gives the effective temperature drop.

At low cold fraction-I, radii of second forced vortex flow is large and so the cold fraction-II and there by larger diameter of orifice at cold end-II is needed to make the same to escape through. Whereas at higher cold fraction-I, the trend is reverse.

Fig 11 shows the effect of cold fraction-II on COP_{R2}using different orifice diameter through cold end-II. Results show that the COP_{R2} increases with increase of cold fraction-II attains a peak value and decreases thereafter. A maximum COP of 0.036, 0.057 and 0.049 is obtained for cold orifice diameter of 3mm, 4mm and 5mm at corresponding cold fraction-II of 0.34. At low cold fraction all the orifice diameters show the similar performance, whereas at higher cold fraction-II, orifice of 4mm diameter performs superior to the remaining two orifices. The reason is as said above that moderate orifice diameter provides sufficient passage for free escape of flow through cold end-II.

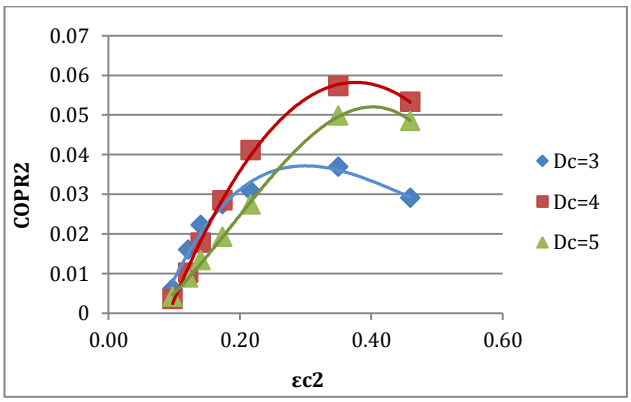


Fig 11: Effect of cold fraction-II on COP at end-II using different orifice diameters

Conclusions

A series of experiments have been conducted to investigate the performance of the Dual forced flow vortex tube (DFFVT) under variable cold fraction, inlet pressure and several cold orifice diameters.

The following were concluded from the experimental data:

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).

The inlet pressure is the driving force for the energy separation. Experimental data show that higher temperature differences are achieved as inlet pressure increases. However, the increase in COP depends on other parameters related to the vortex tube.

The effect of orifice at the cold end-II plays key role in achieving desired flow pattern and thereby effective energy separation in the tube. Smaller diameter attains higher temperature drop through end-II at low cold fraction-II whereas the larger diameter achieves the maximum temperature drops at high cold fraction-II. Orifice diameter of 4mm is the optimum in attaining a maximum temperature drop of 18^0 and co-efficient of performance of 0.057.

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