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Improving vortex tube performance based on vortex generator design

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ABSTRACT

The effect of vortex generator parameters (Cold orifice angle, Cold orifice diameter and Nozzle area) on vortex tube performance is investigated experimentally. Vortex tube is connected to a natural gas pipeline with constant pressure of 4 bars. To improve vortex tube efficiency, six generators with different cold orifice angle, five generators with different cold orifice diameter and three generators with different nozzle area are studied for each experiment part. Results show variation of nozzle area has no effect on optimum cold mass fraction while cold mass angle and cold mass diameter move this point. Increment in cold orifice diameter increases optimum cold mass fraction and decreases cold temperature. As the angle of cold orifice increases, more mass flow passes through cold outlet and optimum cold mass fraction also increases. The expansion of the gas in the diffuser type cold orifice is investigated as the dominating reason for the different vortex tube performance. These mentioned designing parameters of vortex generator affect the flow pattern and efficiency of vortex tube as a consequence. For cold orifice angle of 4.1°, cold orifice ratio of 0.64 and nozzle area ratio of 0.14, highest efficiency is achieved.

1. Introduction

The VT (vortex tube) is a simple device without a moving part which is capable of separating a high pressure gas into hot and cold gas streams. The VT, also known as RHVT (Ranque–Hilsch Vortex Tube) was first discovered in 1933 by Ranque. The German physicist Hilsch [1] worked on the vortex tube and improved the designing parameters, provided comprehensive experimental and theoretical studies intend to improve the efficiency of the vortex tube. He methodically inspected the effect of the inlet pressure and the geometrical parameters of the VT on its performance and presented a possible explanation of the energy separation process. There have been a lot of researchers since then which studying vortex tube aiming to explain the reason of energy separation or enhance its performance. These studies could be divided into two categories as experimental and theoretical study.

Many researchers studied experimentally vortex tube and tried to demonstrate in what dimensions the best performance achieved. Thermophysical parameters and geometrical parameters are important factors affecting the VT performance (Saidi and Valipour [2]). They have classified the parameters affecting vortex tube performance as the thermophysical parameters such as inlet gas pressure, type of gas and cold mass fraction, moisture of inlet gas and the geometrical parameters, i.e., diameter and length of main tube and diameter of the outlet orifice were designated and studied. Xue and Arjomandi [3] studied the effect of the angle of rotating flow on the performance and efficiency of the vortex tube. To find best vortex angle, they used different vortex angle generators. A smaller vortex angle presented better performance for the heating efficiency of the vortex tube. Dincer et al. [4] have also experimentally studied the influences of the position, diameter, pressure and number of nozzles and angle of a mobile plug on the RHVT performance. The most efficient combination of parameters is obtained for a plug diameter of 5 mm, tip angle of 30 °C or 60 °C, by keeping the plug at the same position, and letting the air enter into the vortex tube through 4 nozzles. A series of experiments have carried out by Aydın and Baki [5] to study effects of the length of the pipe, the diameter of the inlet nozzle, and the angle of the control valve on the performance of the counter flow vortex tubes for different inlet pressures. Experiments demonstrated that the higher the inlet pressure, the greater the temperature difference of the outlet streams. It is also shown that the cold fraction is an important parameter influencing the performance of the energy separation in the vortex tube. Optimum values for the angle of the control valve, the length of the pipe, and the diameter of the inlet nozzle are obtained. Farzaneh-Gord and Kargaran [6] proposed the possibility of using vortex tube instead of throttling valves in natural gas pressure reduction points. They have studied VT performance with low pressure natural gas stream experimentally.

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Nomenclature

$A_{\text{Hot Tube}}$	hot tube area
A_{Nozzles}	nozzle area
d	cold orifice diameter (m)
D	vortex tube diameter (m)
h	enthalpy (kJ/kg)
h_c	cold enthalpy (kJ/kg)
h_{in}	inlet enthalpy (kJ/kg)
h_{cs}	cold enthalpy in isentropic process (kJ/kg)
Δh	enthalpy difference (kJ/kg)
Δh_{cs}	enthalpy difference in isentropic process (kJ/kg)
$\dot{m}_c$	cold mass flow rate (kg)
$\dot{m}_{in}$	inlet mass flow rate (kg)
P	pressure (MPa)
p_a	atmosphere pressure (MPa)
p_{in}	inlet pressure (MPa)
R	universal gas constant (kJ/kmol K)
s	entropy (kJ/kg K)
T	temperature (K)
T_c	cold temperature (K)
T_h	hot temperature (K)
T_{in}	inlet temperature (K)
ΔT_c	cold temperature difference (K)
ΔT_h	hot temperature difference (K)

Greek letters

α	cold orifice angle
β	ratio of cold orifice diameter to vortex tube diameter
γ	specific heat ratio
η_{isen}	isentropic efficiency
μ_c	cold mass fraction
ζ	nozzle area ratio

Farzaneh-Gord et al. [7] studied natural gas temperature behaviors in a VT and investigated the effects of hot tube length on VT efficiency. Further, the amount of cooling capacity created by natural gas as it passes through a VT has been calculated.

In parallel with experiments, researchers have carried out many theoretical activities, most are based on results obtained from the related experimental work and some are based on numerical simulations. A few attempts of applying numerical analysis tried to demonstrate the separation phenomena. Some other works tried to study vortex tube from a thermodynamic aspect view. Lewins and Bejan [8], Saidi and Allaf Yazdi [9], Farzaneh-Gord et al. [10] and Xue et al. [11] used the second law of thermodynamics to show temperature separation effect and measured losses in terms of exergy destruction, which provide direct measure of thermodynamic inefficiencies then resulted in several formulas for estimating the performance and efficiency of VT under different operating conditions, which induced the optimum ratios of VT dimensions corresponding to the highest efficiency. Different modeling technics are also used to optimize vortex tube dimensions. For example, by using 81 experimental data sets in the training step, heating and cooling performances of vortex tubes were experimentally investigated and modeled with Fuzzy modeling (Berber et al. [12]).

According to numerical and experimental attempts, various theories have been proposed in the literature to explain the "temperature separation" effect. These studies discuss the different theories to describe why this phenomenon happens in RHVT. Xue et al. [13] presented a critical review of explanations on the working concept of a vortex tube. They discussed hypotheses of pressure, viscosity, turbulence, temperature, secondary circulation and acoustic streaming. Based on the observed velocity, turbulence intensity, temperature and pressure distributions, Xue et al. [14] proposed multi-circulation as the main reason for thermal separation. They demonstrated that kinetic energy transformation outwards from the central flow contributes to the temperature separation. Briefly, the aim of all studies can be considered in three viewpoints; firstly, to find empirical expressions for geometrical/thermophysical parameters which can be used for improving the VT performance; secondly, to apply the VT for wide application purposes, like cooling and heating, cleaning, purifying and separation. In one application, vortex tube is used as a spot cooling device for a tractor cabinet (Kabeel et al. [15]).

Due to low efficiency of a vortex tube with natural gas as working fluid in previous studies (Farzaneh-Gord and Kargaran [6], Farzaneh-Gord et al. [7] and Farzaneh-Gord et al. [10]), the current research has been carried out to improve the performance of VT with natural gas as the working fluid. Study concentration focuses for vortex generator dimensions and designs. Different designing parameters of vortex generator (cold orifice angle, cold orifice diameter and nozzle area) affect the flow pattern in the vortex generator and consequently have an effect on efficiency of vortex tube. Different vortex generators have been considered. Six generators with different cold orifice angle, five generators with different cold orifice diameter and three generators with different nozzle area are examined. The natural gas from a pressure line extracted and directed into the vortex tube. It should be pointed out that the effects of cold orifice angle on the vortex tube performance have not been investigated in all previous researches. Also, cold orifice diameter and nozzle area effects on vortex tube performance for natural gas as working fluid are not studied yet. The variation of cold mass fraction at maximum temperature separation point is of our interest.

2. Theoretical issue

In order to study the performance of a VT, some terms (e.g. cold temperature difference, hot temperature difference, cold mass fraction and isentropic efficiency) should be defined firstly as follow:

a) The cold temperature drop (or difference) and the hot temperature difference of the vortex tube are defined as follows respectively:

$$\Delta T_c = T_{in} - T_c \tag{1}$$

$$\Delta T_h = T_h - T_{in} \tag{2}$$

b) Cold mass fraction is defined as the ratio of cold mass flow rate to inlet mass flow rate. By using a valve at the hot tube end, the passing mass of two ends is controlled.

$$\mu_c = \frac{\dot{m}_c}{\dot{m}_{in}} \tag{3}$$

The cold mass fraction which causes highest cold temperature drop is called *optimum cold mass fraction* here.

c) Cold orifice ratio is the ratio of cold orifice diameter to vortex tube diameter

$$\beta = \frac{d}{D} \tag{4}$$

d) Isentropic efficiency is expressed as the ratio of enthalpy change of inlet and cold outlet to the enthalpy change of the isentropic expansion of the process.

$$\eta_{\text{isen}} = \frac{\Delta h}{\Delta h_{\text{cs}}} = \frac{h_{\text{in}} - h_{\text{c}}}{h_{\text{in}} - h_{\text{cs}}} \tag{5}$$

By supposing the stream as an ideal gas the equation becomes

$$\eta_{\text{isen}} = \frac{T_{\text{in}} - T_{\text{c}}}{T_{\text{in}}\left(1 - \left(\frac{p_{\text{a}}}{p_{\text{in}}}\right)^{\frac{\gamma-1}{\gamma}}\right)} \tag{6}$$

e) Nozzle area ratio defines as the ratio of nozzle area to the vortex tube area.

$$\zeta = \frac{A_{\text{Nozzles}}}{A_{\text{Hot Tube}}} \tag{7}$$

f) Cold orifice angle (α) defines as the angle between orifice wall and horizontal line along orifice centerline (see Fig. 2).

3. Experimental setup

In Fig. 1, the size of the vortex tube under investigation (inlet diameter, vortex tube length and diameter, vortex chamber length and diameter and cold orifice diameter) has been shown. The vortex generator has depicted in Fig. 2 indicated that there are 6 nozzles allowing the working fluid enter the vortex chamber. The diameter of vortex chamber and cold orifice diameter are shown Fig. 1. High pressure natural gas pipeline equipped with a pressure regulator is used as the source of the working fluid. As depicted in schematic diagram of the experimental setup, Fig. 3, a pressure regulator was placed before the inlet to maintain constant input pressure. A pressure gauge is also employed to measure the inlet pressure.

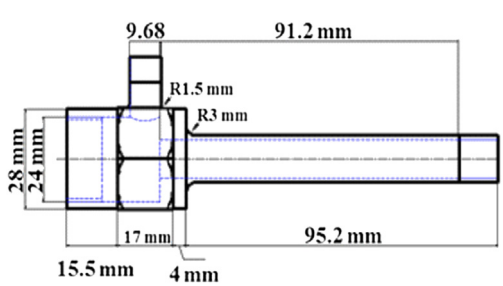

Fig. 1. The vortex tube dimensions under investigation.

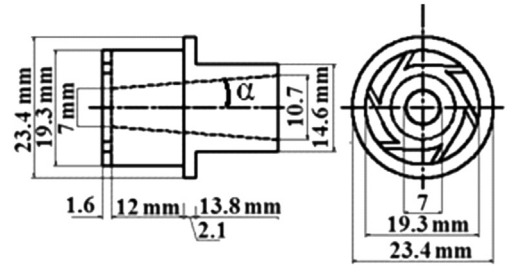

Fig. 2. A vortex generator with cold orifice angle 3.6°.

The natural gas is introduced tangentially into the vortex tube and separated into hot and cold streams and finally discharged into atmosphere. The cold flow in the central region streams out of the tube, while the hot flow in the outer periphery part exits the hot end of vortex tube. The temperatures of the inlet and outlet streams were measured with thermocouples. The cold and hot outlet mass flow is measured by two diaphragm gas meters. The uncertainties of the measurement instruments are given in Table 1. In each test, a valve controls the hot flow. At the beginning of experiment, the valve is fully closed and opened gradually. Cold mass fraction decreases until the valve continues to be opened. When the valve is fully opened, no change happens and cold mass fraction becomes constant. Then, to achieve smaller values of cold mass fraction another valve manipulated to prevent passing out the cold flow.

Error analysis is estimated by using the method proposed by Moffat [16]. Temperature and pressure are directly logged in file with accuracy of 0.1 °C and 0.01 bar respectively. The maximum possible errors in the case of temperature and pressure measurement are:

$$\frac{\partial T}{T} = \sqrt{\left(\frac{\partial T_{\text{PT100}}}{T_{\text{min}}}\right)^2 + \left(\frac{\partial T_{\text{log}}}{T_{\text{min}}}\right)^2} = \sqrt{\left(\frac{0.5}{12}\right)^2 + \left(\frac{0.1}{12}\right)^2} = 0.04 = 4\% \tag{8}$$

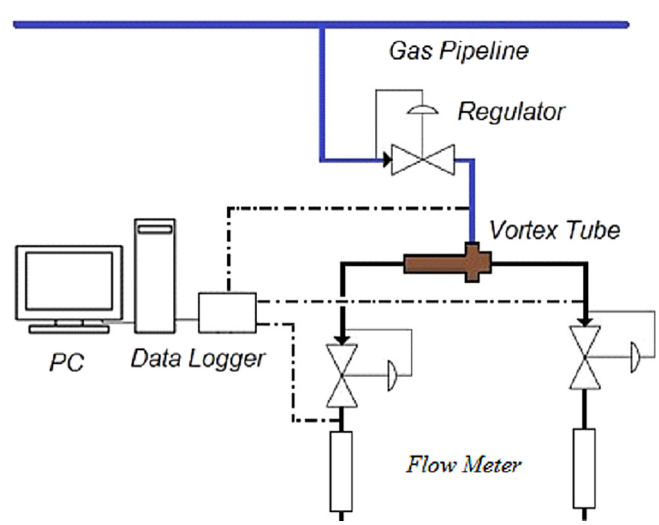

Fig. 3. Schematic diagram of the experimental setup.

Table 1
Characteristics and uncertainties of the measurement instruments.

Instrument	Range	Uncertainty
Temperature sensors	−200 °C to 850 °C	0.5 °C
pressure gauge	0–10 bar	1% full scale = 10,000 Pa
Flow meter	0.16–25 m^3/h	2%

$$\frac{\partial p}{p} = \sqrt{\left(\frac{\partial p_{trans}}{p_{min}}\right)^2 + \left(\frac{\partial p_{log}}{p_{min}}\right)^2} = \sqrt{\left(\frac{0.01}{4}\right)^2 + \left(\frac{0.01}{4}\right)^2} = 0.004 = 0.4\% \quad (9)$$

4. Result and discussion

In these experiments, activities focused on the effects of vortex generators parameters including cold orifice angle, inlet nozzle area and cold orifice diameter on performance of vortex tube. The optimum cold mass fraction has been found for all cases and the influence of the vortex generators parameters on its value is investigated. These three parameters have influence on interchange of mass and energy inside of vortex tube. The effects of inlet nozzle area and cold orifice diameter have been previously studied by many researches such as Singh et al. [17], Im and Yu [18] and Nimbalkar and Muller [19] for air as the working fluid. But the influence of these parameters on variation of optimum cold mass fraction has not been studied. Also here, working fluid is considered as the natural gas which makes a big different between current and previous reteaches.

4.1. Cold orifice angle effect on vortex tube performance

In this section, the effects of cold orifice angle (α) are examined experimentally on the vortex tube performance. Six vortex tube generators are used in these experiments with cold orifice angles of 0.7, 1.6, 2.6, 3.6, 4.1 and 5.1°. In Table 2, the specifications of these vortex generators are presented. A detailed dimension of one with cold orifice angle of 3.6° is shown in Fig. 2. For other generators, the output diameters are varied to create the desired dimensions. It should be pointed out that the effect of cold orifice angle has not been studied in previous researches.

Cold orifice is the output conduit and cold outlet flow leaves vortex tube through this region. Thus change of orifice angle influences the flow structure inside of vortex tube and thermal performance as a consequence. Figs. 4 and 5 show the cold and hot temperature difference variation against cold mass fraction change respectively. According to Figs. 4 and 5, for each generator, hot temperature difference rose steadily over the whole period of cold mass fraction while cold temperature drop shows ascending and descending behavior and has a maximum value at a different specific cold mass fraction in the range of 0.58–0.65. The same trends in hot and cold temperature different are seen in many previous researches (Markal et al. [20]). It could be realized that the cold orifice angle has an effect on temperature differences. The examination of cold orifice angle effects show that highest temperature differences could be obtained at $\alpha = 4.1°$. As both figures show, the cold orifice angle has smaller effects on temperature difference for cold mass fraction lower than 0.6. At low values of cold mass fraction, cold mass flow could easily pass through the cold orifice and thus the cold orifice angle has small effect on thermal separation within the VT.

Table 2
Dimensions of vortex generators.

Generator model	Input diameter (mm)	Output diameter (mm)	Cold orifice angle (α)
1	7	7.76	0.7°
2	7	8.64	1.6°
3	7	9.68	2.6°
4	7	10.70	3.6°
5	7	11.22	4.1°
6	7	12.28	5.1°

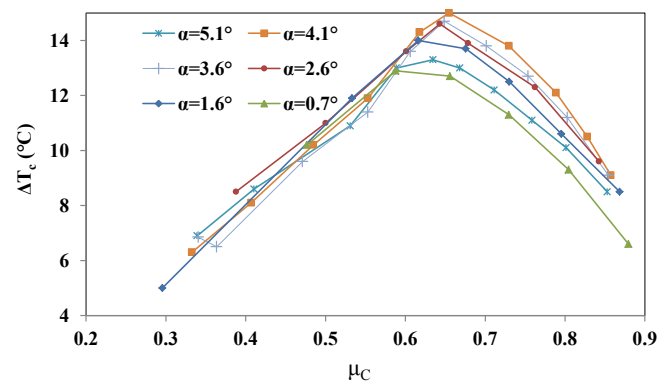

Fig. 4. Effects of cold orifice angle on cold temperature variation.

Fig. 6 provides isentropic efficiency of the VT calculated based on equation (6). The efficiency has the same trend as cold temperature difference (see Fig. 4). Attention to equation (6) states the similar trend. Denominator of the equation is constant and efficiency is directly proportional to cold temperature difference. By increasing angle from 0.7° to 4.1°, the occurrence of optimum cold mass fraction on cold mass fraction axis increases. Angle 4.1° offers highest efficiency and largest optimum cold mass fraction among six generators. This enhancement can be attributed to the flow structure and resistance along the cold orifice. For a vortex tube, Beran and Culick [21] stated that flow structures differ inside the vortex tubes due to initial swirl intensity, the swirl decay rate and various pressure drops inside the tube.

In the experiments, optimum cold mass fraction is achieved when both hot and cold valves were fully opened. Change in occurrence of optimum cold mass fraction is due to pressure balance inside and outside of vortex tube as mentioned by Love [22] and Piralishvili and Fuzeeva [23]. As cold mass fraction varies from optimum cold mass fraction to 1, Nimbalkar and Muller [19] stated that total pressure drop across vortex tube is due to the addition of pressure drop at the inlet, pressure drop due to the generator, pressure drop the flow circulation, pressure drop

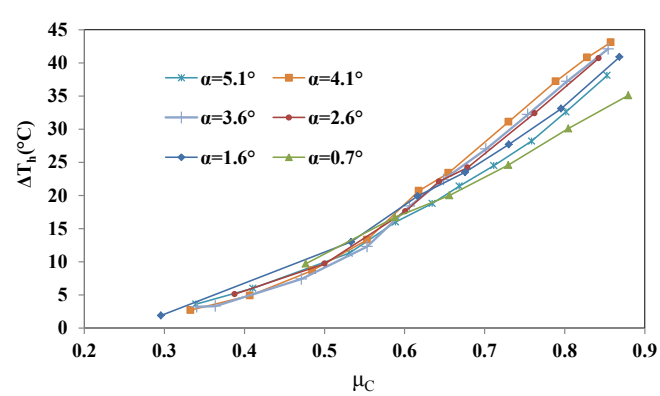

Fig. 5. Effects of cold orifice angle on hot temperature variation.

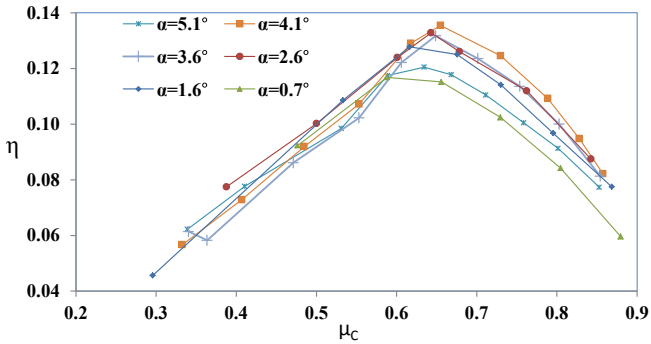

Fig. 6. Effects of cold orifice angle on the VT efficiency.

through the cold-end orifice and pressure drop due to the hot fraction control valve. Pressure drop across the generator are affected by the geometry of the generator and the number of nozzles. In current section, the dimension of the generators (through varying the cold orifice angle) is the only change which has effects on pressure drop. The figure show that generator with orifice angle 4.1° has highest efficiency which means this generator creates the least pressure drop in cold region and consequently its occurrence of optimum cold mass fraction has the largest value. This generator presents the highest efficiency among other generators at cold mass fraction of 0.65 with efficiency of 13%. For greater and smaller angles than 4.1°, the efficiency degrades.

It could be realized that the maximum efficiency is achieved for cold mass fraction varying between 0.58 and 0.65. The cold mass fraction which causes maximum efficiency (or cold temperature drop) is called optimum cold mass fraction here. The cold orifice angle not only has effect on efficiency value but also on the position of the occurrence of optimum cold mass fraction. Nikolaev et al. [24] observed the range of 0.6–0.7 for optimum cold mass fraction. Poshernev and Khodorkov [25] stated optimum cold mass fraction is occurred between 0.5 and 0.6. Nimbalkar and Muller [19] are the only researchers which reported a fixed value of 0.6 for optimum cold mass fraction.

The results of this section show that the employment of the different cold orifice angles lead to different tube performances. The configuration of the cold orifice angle appears like a diffuser connected to the cold orifice; hence the expansion of the gas in the diffuser is the dominating reason for the different tube performance. Whenever the kinetic and potential energies of the working fluid are negligible, the enthalpy represents the total energy of a fluid. For this high-speed flow through cold orifice, the potential energy of the fluid is negligible comparing to the kinetic energy. Consider the steady flow of a gas through diffuser where the flow takes place adiabatically with no work and no change in potential energy. In the absence of any heat and work interactions and any changes in potential energy, the stagnation enthalpy of a fluid remains constant. Thus any decrease in fluid velocity in orifice creates an equivalent increase in the static enthalpy of the gas and consequently increase in temperature. But this theory cannot explain difference temperature rise due to orifice angle increase. Therefore another explanation should be brought into attention for this temperature change.

A comprehensive numerical analysis of conical diffuser for natural gas is presented by Wen et al. [26]. Depending on the geometry and Reynolds number, several regime flows can exist in a diffuser. When the opening angle of the diffuser is very low, the boundary layers are thin and the effective area of the channel and the geometrical area both grow similarly. By increase in angle of diffuser, large amplitude fluctuations appear with repeated regions of reversed velocity along the wall of the diffuser. By continuing the increase in diverging angle, a region of back flow created (Greitzer et al. [27]). Sparrow et al. [28] noted that some sources state that flow in conical diffusers separates if the divergence angle exceeds 7°, while other sources indicate a critical divergence angle of 15°. They themselves found that flow separation occurred for a diffuser expansion angle of 5° for inlet Reynolds numbers less than about 2000. They numerically studied divergence angles of 5°, 10° and 30°, and Reynolds numbers varying from 500 to 33,000, showing that the flow separated at lower Reynolds numbers for small divergence angles. Furthermore, the extent of the separated flow region was shown to generally decrease as the Reynolds number increased. Results showed separation at all investigated Reynolds numbers. As obvious, in diffusers, separation of the flow can occur if cone angle and area ratio be selected improperly. This separation can cause steady-state or intermittent separation of the flow from the diffuser walls and higher losses in downstream. Generally, proper diffuser design requires that the equivalent cone angle be constrained below a certain value for a given area ratio. The results observed in vortex tube experiment prove that difference in vortex tube performance is due to diverging angle of cold orifice and consequently the expanding process occurs in diverging duct. The determination of the diffuser loss parameter is presented by Eckert et al. [29]. They stated that energy loss along diffusers depends on the cross-sectional shape and equivalent cone angle of the section. For a conical diffuser the expansion functions are, for $0^\circ < 2\alpha < 3^\circ$, $3^\circ < 2\alpha < 10^\circ$ and $2\alpha > 10^\circ$:

$$\text{for}3^\circ < (2\alpha) < 10^\circ K_{\text{exp}} = 1.70925 \times 10^{-1} - 5.84932 \times 10^{-2}(2\alpha) + 8.14936 \times 10^{-3}(2\alpha)^2 + 1.34777 \times 10^{-4}(2\alpha)^3 - 5.67258 \times 10^{-5}(2\alpha)^4 - 4.15879 \times 10^{-7}(2\alpha)^5 + 2.10219 \times 10^{-7}(2\alpha)^6 \tag{10}$$

$$\text{for}0^\circ < (2\alpha) < 3^\circ K_{\text{exp}} = 1.033395 \times 10^{-1} - 1.19465 \times 10^{-2}(2\alpha) \tag{11}$$

$$\text{for}(2\alpha) > 10^\circ K_{\text{exp}} = -9.66135 \times 10^{-2} + 2.336135 \times 10^{-2}(2\alpha) \tag{12}$$

By applying above equations for orifice angles of Table 3, loss coefficient due to expansion is calculated and tabulated. For each cold orifice angle, the maximum efficiency is also presented. First increase in angle shows increase in expansion loss through the diverging orifice due to proper expansion occurs downstream of diffuser. The least expansion loss occurs at cold orifice angle 4.1° and due to better expansion in this case, the best performance achieves for this angle.

Paying attention to Fig. 6 show that relation of maximum optimum cold mass fraction and cold orifice angle. The highest value

Table 3
Expansion losses and maximum efficiency for cold orifice angles.

η_{max} (%)	K_{exp}	Cold orifice angle (α)
11.7	0.104	0.7°
12.8	0.065	1.6°
13.2	0.041	2.6°
13.3	0.017	3.6°
13.5	0.005	4.1
12	0.142	5.1°

of maximum optimum cold mass fraction achieves when diverging angle of orifice is 4.1°. By increases the orifice angle, this parameter increases for angle 4.1° and then it decreases again. This implies that the least pressure is created in this angle. A good conformity is observed between the data and Eckert's relation.

4.2. Cold orifice diameter effect on vortex tube performance

In this section, the influence of COD (cold orifice diameter) is studied on vortex tube performance and on the variation of optimum cold mass fraction. Many researchers including Nimbalkar and Muller [19] and Liu and Liu [30] studied experimentally cold orifice diameter and tried to explain its effect on VT performance. In this study, the cold orifice angle is zero degree. Five generators considered for these experiments, have cold orifice diameters of 5.6, 6.0, 6.4, 7.7 and 8.2 mm. Diameter of vortex hot tube is 10 mm. Inlet pressure is kept constant as 4 bars.

Variation of cold and hot temperature difference against cold mass fraction depicted in Figs. 7 and 8 respectively. The effects of cold orifice diameter could also be examined in these figures. The trends of temperature differences are similar to Figs. 4 and 5 which discussed in previous section. Note from Fig. 7, the minimum temperature achieves at various values of cold mass fraction for each COD. As cold orifice diameter increases, optimum cold mass fraction value increases. The variation in minimum temperature is due to structure of flow within VT. Part of main flow returns from the hot end of the VT centrally flow towards the cold end side. Diameter of this return flow depends on cold orifice diameter. Consequently for each cold orifice diameter, there will be an optimum diameter for return flow. This phenomenon causes that optimum cold mass fraction varies with cold orifice diameter. Comparing the results for orifice diameters, it could be realized that the highest cold temperature difference occurs for $\beta = 0.64$. Eiamsa-ard and Promvonge [31] reviewed the results of previous researches and pointed out that the optimum value for β in which highest cold temperature difference occurs, is equal to 0.5. The difference between optimum β value in this research and Eiamsa-ard and Promvonge [31] may be due to difference in internal path of flow and in working fluids. Also, their results are presented for air while here the natural gas is selected as the working fluid.

An increase in the cold mass fraction in a range of 0.3–0.8 is seen to increase the value of hot temperature difference (Fig. 8). According to Fig. 8, generator with cold orifice ratio of 0.64 presents highest separation in heating. Cold orifice ratios higher and lower than 0.64, cause disorders affecting the separation performance of vortex tube. As shown in Figs. 7 and 8, an increase in cold orifice diameter from the optimum cold orifice ratio (0.64), resulted to a sharp decrease in cold and hot temperature difference. By

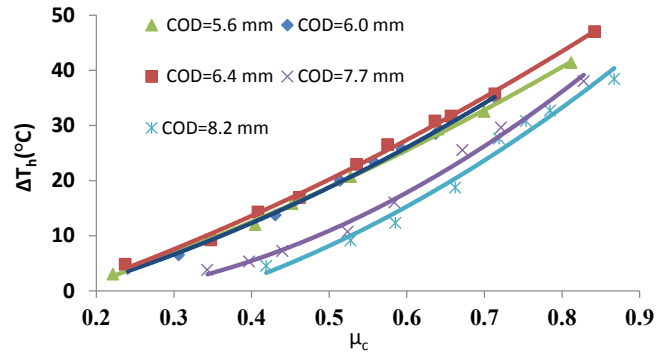

Fig. 8. Effects of cold orifice diameter on hot temperature variation.

increasing cold orifice diameter, the VT performance decreases considerably. In larger cold orifice diameters, more flow escapes directly from the cold orifice without coming into the vortex tube and interacting with interval flow. Consequently decreasing in interaction of flows, causes decrease in VT performance. Comparing Figs. 7 and 8 shows cold temperature difference is more sensitive to cold orifice diameter than hot temperature difference as mentioned by Bovand et al. [32].

Fig. 9 provides isentropic efficiency for five different generators with various cold orifice diameters. Among five tested generators highest efficiency achieves for cold orifice ratio of 0.64. For this generator efficiency reaches 22% at cold mass fraction of 0.53. Respective of optimum orifice diameter, for 10% decrease and increase in cold orifice diameter, efficiency decreases about 17% and 40% respectively. Comparing Fig. 7, variation of optimum cold mass fraction is observed while Nimbalkar and Muller [19] believed that cold orifice diameter has no effect on VT efficiency. They stated that at constant cold mass fraction of 0.60, the vortex tube achieves ideal operating conditions, and hence the efficiency reaches to its maximum value irrespective of orifice diameter. Nimbalkar and Muller [19] believed that when there is a decrease in cold fraction, axial stagnation point moves towards the hot end, and due to the stretching of the central recirculating core, radial stagnation point moves towards the axis of the tube. They stated that for the ideal separation of cold and hot flow streams, there are fixed critical locations for the axial and the radial stagnation points. But clearly with variation of cold orifice diameter and consequently the variation of tangential velocity of flow in the tube (Eiamsa-ard and Promvonge [31]), there is not a fixed point for axial and radial stagnation point at which highest temperature separation occurs.

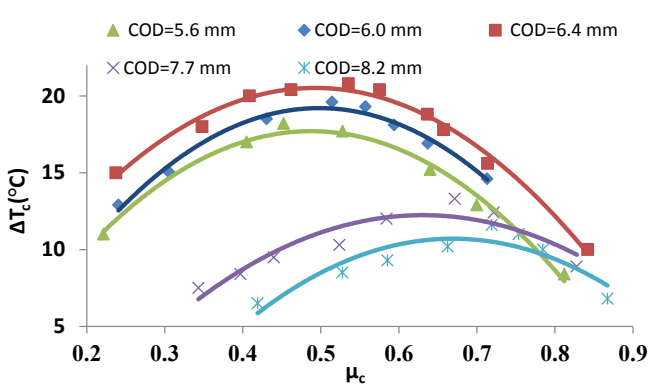

Fig. 7. Effect of cold orifice diameter on cold temperature difference.

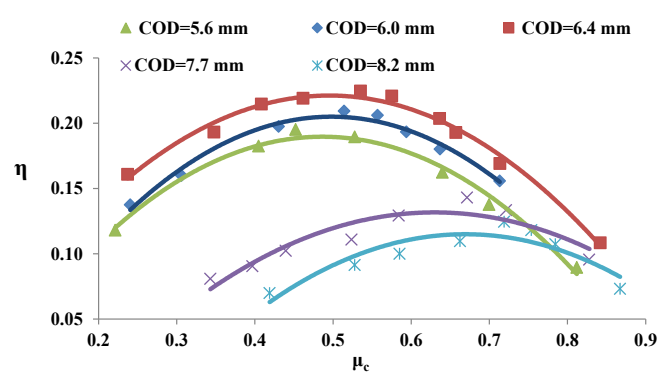

Fig. 9. Effects of cold orifice diameters on the VT efficiency.

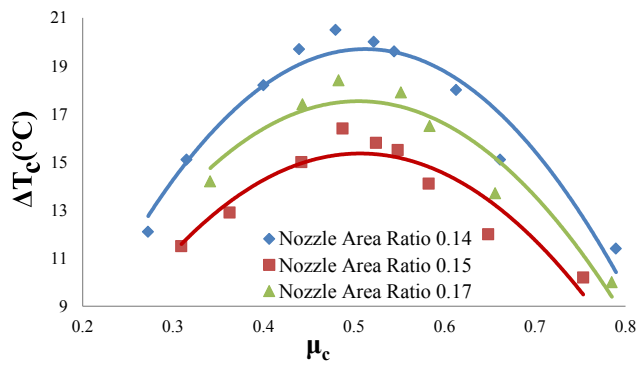

Fig. 10. Effects of nozzle area on cold temperature variation.

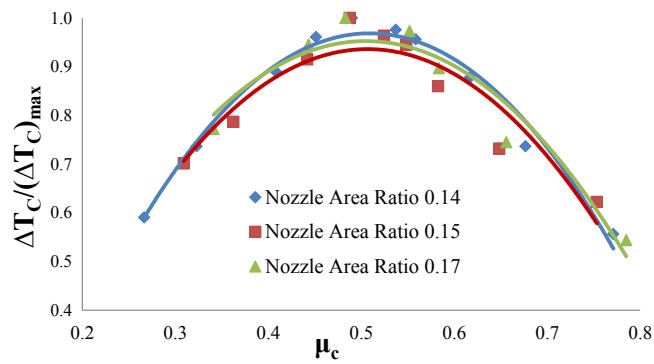

Fig. 12. Effects of nozzle area on the non-dimensional cold temperature.

4.3. Nozzle area effect on vortex tube performance

For 3 generators with different nozzle area ratio, the experiments repeated to investigate the influence of nozzle area on variation of optimum cold mass fraction. The inlet nozzle plays an important role to get the best performance of VT. For example, the convergent nozzles can improve VT performance in comparison with common nozzles (Rafiee and Rahimi [33]). Yilmaz et al. [34] stated that a very small inlet nozzle would give rise to considerable pressure drop in the nozzle itself, leading to low tangential velocities and hence low temperature separation. A very large inlet nozzle would fail to establish proper vortex flow resulting again in low diffusion of kinetic energy and therefore low temperature separation. There have been many investigations aiming to study the effect of inlet nozzle area on the performance of the VT. Aydın and Baki [5] proposed the ratio of inlet nozzle diameter to vortex tube diameter 0.33 respectively while Takahama [35] proposed that the mentioned ratio should be less than 0.2 in order to have larger temperature differences. By looking in previous articles, Yilmaz et al. [34] stated that the optimum ratio of ζ is 0.25. The effect of convergent nozzles and different nozzle numbers on vortex tube performance experimentally performed (Rafiee and Rahimi [33]). To study the effect of nozzle area on optimum cold mass fraction, three nozzle area ratios are considered as 0.14, 0.15 and 0.17. Figs. 10 and 11 present effects of nozzle area on cold and hot temperature variation. As depicted in Figs. 10 and 11, highest temperature separation occurs for nozzle area ratio of 0.14. As Fig. 10 shows minimum temperature occurs at the same cold mass fraction. It demonstrates that variation of nozzle area could not change the optimum amount of mass exiting from cold end and thus no change would occur in optimum cold mass fraction, while variation in cold orifice angle or cold orifice diameter has effect on optimum cold mass fraction. When cold orifice angle or diameter changes, balance of pressure inside vortex tube varies. With this change, optimum cold mass fraction varies. Cold orifice angle and diameters are parameters affect the mass flow inside vortex tube while nozzle area has no influence on exit mass flow and it just controls the input mass.

Fig. 12 confirms Stephan's theory for nozzle area variation. Stephan et al. [36] represented a general mathematical formulation of the energy separation process taking place in a vortex tube. The similarity relation showed the ratio of the actual temperature drop of the cold flow to the maximum temperature drop of the cold flow (non-dimensional cold temperature) can be represented just as a function of the cold mass fraction. Variation of nozzle area does not change the pressure balance and consequently no change occurs in optimum cold mass fraction.

The observation to be made with the help of Fig. 13 is the comparison of efficiency for three cases. For each nozzle area, efficiency reached to its maximum at cold mass fraction of 0.5. Nozzle area ratio of 0.14 showed maximum temperature separation and efficiency reaches 0.19. Due to very small inlet nozzle, pressure drop increases in the nozzle and leads to low tangential velocities and hence low efficiency. A very large inlet nozzle would fail proper swirl flow formation and resulting in low efficiency.

Concentrating on cold and hot temperature difference for all experiments shows that there are similar trend. As cold mass fraction increases, both temperature differences increase. For very low cold mass fraction, there is weak secondary circulation and thus low heat transfer occurs between inner and outer fluid layers inside of vortex tube (Ahlborn and Groves [37]). By increasing cold mass fraction, hot flow gets energy from cold flow and warms up and consequently the cold flow cools down. By increasing cold mass fraction, hot mass flow reduces and heat addition also increases the hot temperature difference. For cold mass fraction higher than optimum value, any increase in cold mass fraction resulted to

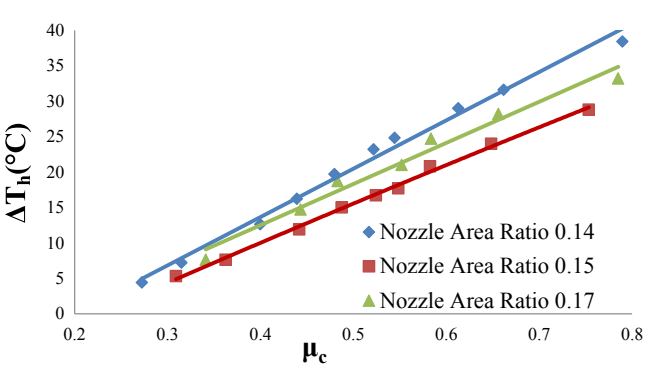

Fig. 11. Effects of nozzle area on hot temperature variation.

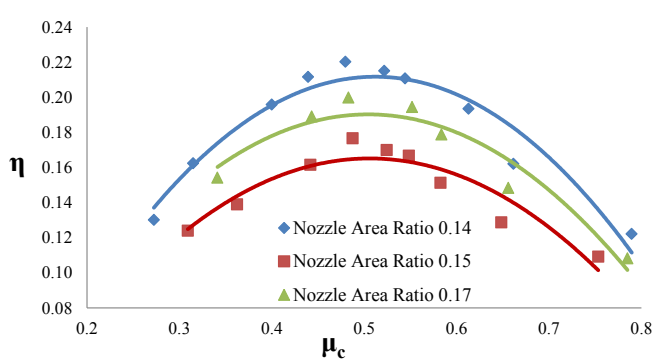

Fig. 13. Effects of nozzle area on the VT efficiency.

weakness of secondary circulation and consequently less heat transfer occurs between central and periphery streams. Due to increase in cold mass flow and decrease in strength of secondary circulation, cold temperature difference starts decreasing. This is why there is an optimum point at which maximum cold temperature difference occurs. Nikolaev et al. [24], Poshernev and Khodorkov [25] and Nimbalkar and Muller [19] reported different optimum cold mass fraction. It is here believed that vortex tube design and pressure distribution are the reasons for variation in optimum cold mass fraction. The design of vortex tube affects the flow distribution through the vortex tube. In addition, pressure balance varies between two cold and hot outputs and consequently cold mass fraction changes during these manipulations.

5. Conclusion

Vortex tube is extensively used in industry for spot cooling. There are potential applications of VT in natural gas pressure reduction points. Consequently a wide area of usage could be established for this instrument. In this study, a vortex tube has examined with natural gas as the working fluid. Temperature and pressure along the instrument setup were measured by thermocouples and pressure gauges. The experiment on vortex generator performed in three sections and the effects of different vortex generator parameters on variation of optimum cold mass fraction are studied. First part focused on influence of cold orifice angle. For this goal, six generators with different angles are constructed and tested. The experiments depicted that variation of cold orifice angle influences vortex tube performance and best separation achieves for angle 4.1°. With respect of orifice angle, each generator has its own optimum cold mass fraction. The reason for this displacement in cold mass fraction is the change of pressure distribution inside the vortex tube. As cold orifice angle increase from 4.1°, increment in optimum cold mass fraction occurred. The change in optimum cold mass fraction with variation of diverging angle of cold orifice shows pressure balance inside vortex tube affected by this parameter. As orifice angle increases from 0.7° to 4.1°, the optimum cold mass fraction increases but with more increase in angle, the amount of cold mass decreases.

Second part of experiments performed in order to see how vortex tube performance and optimum cold mass fraction are influenced by cold orifice diameter for natural gas as working fluid. The route of cold orifice affects the formation of flow pattern through vortex tube. When cold orifice diameter increases, the ability of vortex tube in directing flow through cold orifice increases and therefore optimum cold mass fraction increase. Highest efficiency attained for cold orifice ratio of 0.64. Smaller cold orifice ratio would give higher back pressure in the VT and which resulted in low efficiency. Larger cold orifice ratio would tend to draw flow directly from the inlet and yield weaker swirl velocities in the VT, resulting again in low efficiency.

Finally nozzle area effect on vortex tube performance examined. Three nozzles are considered with different nozzle area. Conversely, three generators presented their best performance at a fixed optimum cold mass fraction. This constant optimum point indicates the best flow pattern is the same for these generators and nozzle area has no effect on optimum cold mass fraction while variation of cold orifice angle and cold orifice diameter move this point.

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