RESEARCH ARTICLE
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Investigation on the Vortex Thermal Separation in a Vortex Tube Refrigerator

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ABSTRACT: This paper describes the experimental study of the temperature separation phenomenon in a counter-flow type vortex tube. Effects of (1) the number of inlet tangential nozzles, (2) the cold orifice diameter, and (3) tube insulations on the temperature reduction and isentropic efficiency of the tube were experimentally investigated. The temperature drops of the cold air obtained from these tests were in good agreement with available data for comparison at a similar scale of operating conditions. The experimental results showed that the insulated vortex tube with 4 inlet nozzles and cold orifice diameter of 0.5D yielded the highest temperature reduction (temperature separation) and isentropic efficiency at about 30 °C and 33% respectively.

KEYWORDS: vortex tube, counter-flow type vortex tube, energy separation, thermal/temperature separations, temperature reduction.

INTRODUCTION

The Ranque-Hilsch vortex tube is a mechanical device operating as a refrigerating machine without any moving parts, by separating a compressed gas stream into a low total temperature region and a high one. Such a separation of the flow into regions of low and high total temperature is referred to as the temperature (or energy) separation effect. Generally, the vortex tube can be classified into two types. One is the counter-flow type (often referred to as the standard type) and the other is the parallel or uni-flow type. The counter-flow vortex tube consists of an entrance block of nozzle connections with a cold orifice, a vortex tube (or hot tube) and a cone-shaped valve. A source of compressed gas (e.g. air) at high pressure enters the vortex tube tangentially through one or more inlet nozzles at a high velocity. The expanding air inside the tube then creates a rapidly spinning vortex. The air flows through the tube rather than pass through the cold orifice located next to the nozzles because the orifice is of a much smaller diameter than the tube. Therefore, the cone valve is applied at the hot tube end in order to control the outlet hot air flow or to let the cold air pass through the cold orifice as needed. The length of the tube is typically between 30 and 50 tube diameters, and no optimum value has been determined between these limits. As the air expands down the tube, the pressure drops sharply to a value slightly above the atmospheric pressure, and the air velocity can approach the speed of sound. Centrifugal action will keep this constrained vortex close to the inside surface of the tube. The uni-flow vortex tube comprises of an entrance block of inlet nozzles, a vortex tube and a cone-shaped valve with a central orifice. Unlike the more popular counter-flow version, the cold air exit is located concentrically with the annular exit for the hot air. The operation of the uni-flow vortex tube is similar to the operation of the counter-flow one. Nowadays the property of vortex tube has a great variety of application in many industries. It has been widely used in the cooling industrial fields especially grilling, turning and welding on account of the various advantages of vortex tube such as cooling without moving part, non-electricity consuming, tiny, lightweight and inexpensive working chemical substance inside the vortex tube, uncomplicated cooling point, cleanliness, convenience and non-CFC's free from pollution.

The vortex tube was first discovered by G. J. Ranque (1933), a metallurgist and physicist who was granted a French patent for the device in 1932, and a United States patent in 1934. The initial reaction of the scientific and engineering communities to his invention was disbelief and apathy. Since the vortex tube was thermodynamically highly inefficient, it was abandoned for several years. Interest in the device was revived by Hilsch (1947), a German engineer, who reported an account of his own comprehensive experimental and theoretical studies aimed at improving the efficiency of the vortex tube. After Hilsch[1], an experimental study was made by Scheper[2] who measured the velocity, pressure, and total and static temperature gradients in the Ranque-Hilsch vortex tube, using probes and visualization techniques. Martynovskii and Alekseev[3] studied experimentally the effect of various design parameters of vortex tubes. The velocity, temperature and pressure profiles agreed with the hypothesis of Fulton[4] and supported the suggestion of the conversion of a free vortex into a forced vortex inside the tube. Blatt and Trusch[5] investigated experimentally the performance of the uni-flow vortex tube and improved its performance by adding a radial diffuser to the end of the shortened tube instead of a cone valve. Bruun[6] presented the experimental data of pressure, velocity and temperature profiles in the counter-flow vortex tube with a ratio of 0.23 for the cold to total mass flow rate and concluded that radial and axial convective terms in the equations of motion and energy were equally important. Williams[7] studied the counter-flow type vortex tube with methane and Algerian natural gas with a high methane content, in order to estimate the effects of gas supply temperature and pressure. A critical inlet Reynolds number was identified at which the separation was a maximum. Takahama et al.[8] investigated experimentally the energy separation performance of a steam-operated standard vortex tube and reported that the performance worsened with wetness of steam at the nozzle outlet because of the effect of evaporation. Takahama and Yokosawa[9] examined the possibility of shortening the chamber length of a standard vortex tube by using divergent tubes for the vortex chamber. Stephan et al.[10] measured temperatures in the standard vortex tube with air as a working medium in order to support a similarity relation of the cold gas exit temperature with the cold gas mass ratio, established using dimensional analysis. Ahlborn et al.[11] carried out measurements in standard vortex tubes to support their models for calculating limits of temperature separation. Flohlingsdorf and Unger[12] predicted numerically the compressed flow and energy separation phenomena in the vortex tube through the CFX code. Promvonge[13] introduced a mathematical model for the simulation of a strongly swirling compressible flow in the vortex tube by using an algebraic Reynolds stress model (ASM) and the k-ε turbulence model to investigate flow characteristics and energy separation in a uni-flow vortex tube. It was found that a temperature separation in the tube exists and predictions of the flow and temperature fields agree well with measurements. The ASM yielded more accurate prediction than the k-ε model. Guillaume and Jolly[14] compared the two vortex tubes placed in a charged configuration and those placed in series by connecting the cold discharge of one stage into the inlet of the following stage. From their results, it was found that for similar inlet temperatures, a two-stage vortex tube could produce a higher temperature reduction than one of the vortex tubes operating independently. Promvonge and Eiamsa-ard[15] experimentally studied the energy and temperature separations in the vortex tube with a snail entrance. In their experimental results, the use of snail entrance could help to increase the cold air temperature drop and improve the vortex tube efficiency in comparison with those of original tangential inlet nozzles.

The analysis in the previous research has been found that the vortex thermal separation phenomenon comes mainly from the diffusion process of mean kinetic energy. Low temperatures (or large temperature separation), both total and static, are found near the tube axis, becoming lower towards the orifice or the cold exit of the counter-flow vortex tube. One might want to know how the diffusion process of mean kinetic energy affects the design of the vortex tube. In general, the vortex tube is designed to obtain either (i) the maximum temperature separation or (ii) the maximum efficiency. At a given supply pressure, however, many vortex tubes with different design parameters can yield the same temperature separation. The separation phenomenon in the tube is understood clearly. If any design parameter of a particular vortex tube affects the flow field, it would certainly affect the performance of the tube.

The process of thermal gas separation in the vortex tube has caused a great deal of interest because of its extreme simplicity and seemingly paradoxical behavior. It seems possible with an input gas pressure of only a few atmospheres, to obtain an axial cool gas current whose stagnation temperature is from 15 to 70 °C below the initial stagnation temperature of the input gas. At the same time the peripherally rotating gas current leaves the tube with a stagnation temperature significantly greater than that of the input compressed gas.

This paper introduces an experimental research on temperature reduction (cold tube) and temperature rise (hot tube) characteristics in the counter-flow vortex tube. The main goals of this study were (1) to investigate the effects of vortex tube geometry such as the number of inlet nozzles, cold orifice diameter, and tube with or without insulation, on temperature separation and isentropic efficiency and (2) to study the wall temperature distribution along the whole length of the hot air tube.

Materials and Methods

Experimental Apparatus

In the design of a vortex tube, there are several tube parameters to be considered, such as (1) insulated and non-insulated tubes, (2) cold orifice diameter, and (3) the number of the inlet tangential nozzles. There are no critical dimensions of these parameters that would result in a unique value of maximum temperature separation. Knowledge of the temperature separation phenomenon suggests a relative design procedure for the vortex tube with the physical realities of its operation. For fixed inlet conditions (supply pressure), a very small diameter of the vortex tube would offer considerably higher back pressures and, therefore, the tangential velocities between the periphery and the core would not differ substantially due to the lower specific volume of air (still high density) while the axial velocities in the core region are high. This would lead to low diffusion of kinetic energy which also means low temperature separation. On the other hand, a very large tube diameter would result in lower overall tangential velocities both in the core and in the periphery region, which would produce low diffusion of mean kinetic energy and also low temperature separation.

A very small cold orifice diameter would give a higher back pressure in the vortex tube, resulting, as discussed above, in low temperature separation. On the other hand, a very large cold orifice diameter would tend to draw air directly from the inlet and yield weaker tangential velocities near the inlet region, resulting in low temperature separation. Similarly, 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. The inlet nozzle location should be as close as possible to the orifice to yield high tangential velocities near the orifice. A nozzle location away from the orifice would lead to low tangential velocities near the orifice and hence low temperature separation.

In this experimental test run, the vortex tube with the inner tube diameter of 16 mm and length of 880 mm was made of Plexiglas which can be divided into two sections; one is the hot air tube of 720 mm (45D) in length and the other is the cold air tube of 160 mm (10D) in length. The cold orifice plate was mounted between the hot and cold tubes made of Plexiglas of 2 mm in thickness. The vortex tube was insulated with 20 mm thick Aero-flex pipe insulation. The change of the cold orifice diameters was varied from 0.4D to 0.9D. The inlet nozzle chamber mounted between the hot and cold tubes was made of Plexiglas of 20 mm in thickness and inlet nozzles with a diameter of 2 mm (D/9) were placed equally around the chamber periphery, depending on the number of the nozzles used.

The counter-flow vortex tube and the arrangement of the experimental system are depicted in Figures 1 and 2. The experimentation started when a compressed air from a compressor flows through the control valve and the pressure gauge and then air-filter before entering the vortex tube. Inside the vortex tube, the air is separated into two currents and escapes into the atmosphere through hot and cold tubes. The cold air would flow out from the cold orifice plates installed near the inlet nozzles, whereas the hot air escapes from the end of hot tube equipped with the cone-shaped valve. The mass rates of the flow of the cold air and hot air discharges are measured by standard pipe orifice flow meters and their ratio, called a cold mass fraction is changed by regulating the cone-shaped valve opening. In the experiments, the cold discharge thermocouple is installed downstream of the cold discharge orifice. Cold air temperature is measured at the middle of the cold air tube while the hot discharge thermocouples are located immediately upstream of the cone-shaped valve and the hot air temperature is measured at a 1/8 tube radius from the inner wall of the hot air tube. All temperature data were measured with iron-constantan thermocouples type K and recorded with a signal-processing unit on a personal computer.

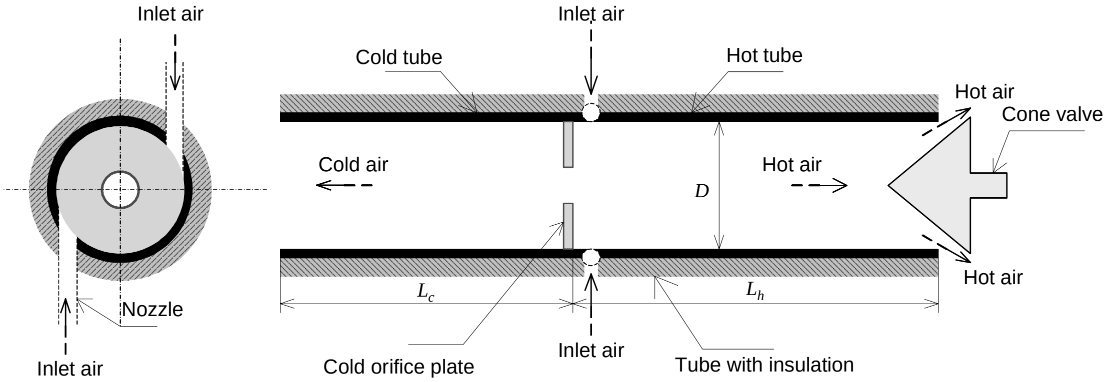


Fig 1. Basic operation of the Ranque-Hilsch standard vortex tube or counter-flow vortex tube.

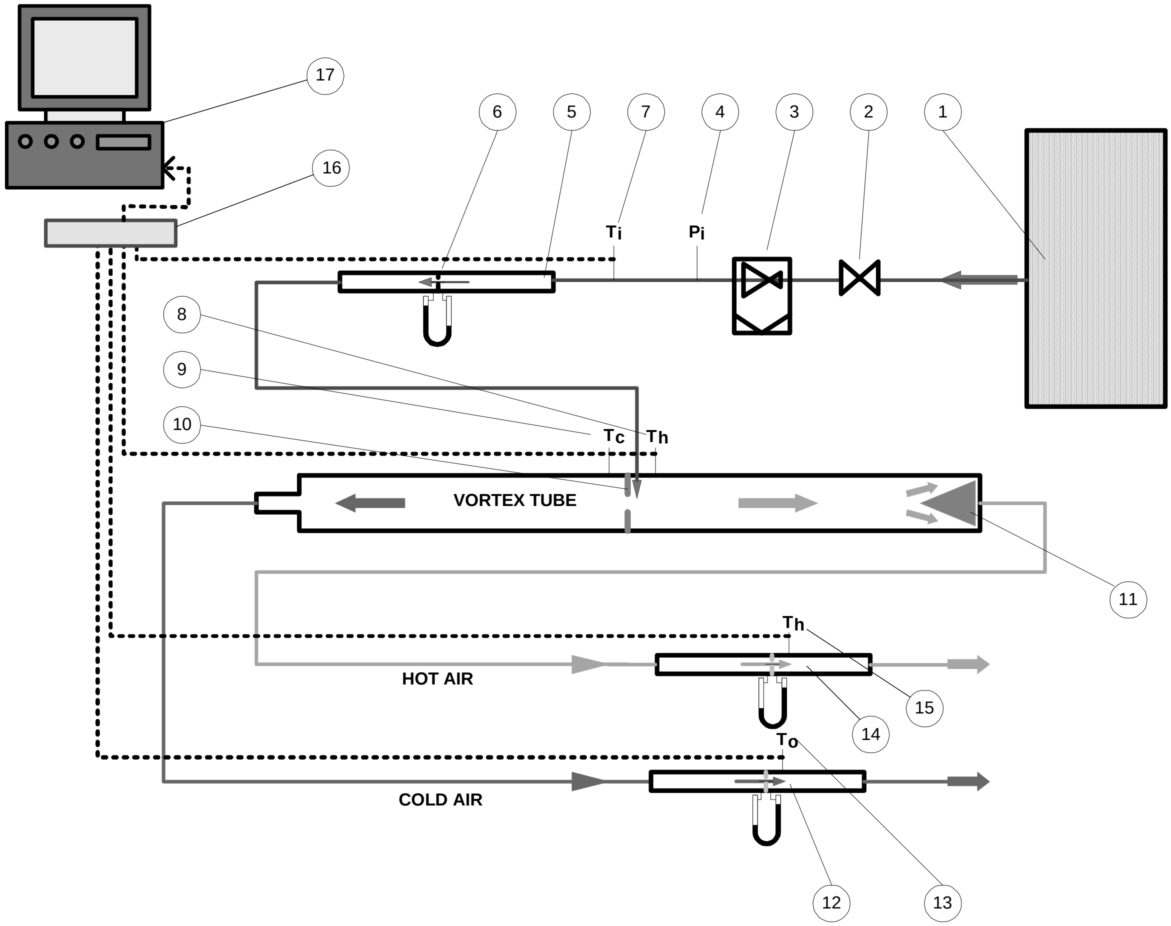


Fig 2. Experimental apparatus, 1) air compressor, 2) control valve, 3) air filter, 4) pressure gage, 5-6) a set of orifice flow meters, 7-9) thermocouple probe, 10) cold orifice plate, 11) cone-shape valve, 12 and 14) orifice flow meter, 13 and 15) thermocouple probe, 16) data logger, and 17) PC computer.

Data Reduction Equations

The most important parameter indicating the vortex tube performance is the cold mass fraction which can be expressed as;

$$\mu_c = \frac{\dot{m}_c}{\dot{m}_i} \tag{1}$$

Cold air temperature drop is expressed as:

$$(\Delta T)_c = T_i - T_c \tag{2}$$

Hot air temperature difference is expressed as:

$$(\Delta T)_h = T_h - T_i \tag{3}$$

To calculate the cooling efficiency of the vortex tube, the principle of adiabatic expansion of ideal gas will be used. As the air flows into the vortex tube, the expansion in isentropic process occurs. This can be written as follows:

$$(\Delta T)_c = \eta_{is}(\Delta T)_{isen} \tag{4}$$

Temperature difference in isentropic process is:

$$(\Delta T)_{is} = T_i\left(1-\left(\frac{P_a}{P_i}\right)^{\left(\frac{\gamma-1}{\gamma}\right)}\right) \tag{5}$$

Isentropic efficiency is expressed as:

$$\eta_{is} = \frac{T_i - T_c}{T_i\left(1-\left(\frac{P_a}{P_i}\right)^{\left(\frac{\gamma-1}{\gamma}\right)}\right)} \tag{6}$$

Confirmatory Test

Before the results of experimental work are reported, the data are compared with previous measured data of Hilsch[1] and Guillaume and Jolly[14].

Table 1. Comparison of the present tube with the previous investigator.

Vortex tube characteristic	Present work	Hilsch (1947)	Guillaume and Jolly (2001)
Pressure at the inlet,(bar)	2.0	3.0	1.5
Hot tube length, L_h	45D	32D	-
Cold tube length, L_c	10D	10D	-
Vortex tube diameter, (mm)	16 (D)	9.2 (D)	6.4 (D)
Nozzle diameter,	D/9	D/9	D/3
Number of inlet nozzles,	1, 2 and 4	1	1
Cold orifice diameter,	0.5D	-	-
Vortex chamber	-	-	2D

The details of geometries and working conditions of the present vortex tube and the previous work are shown in Table 1. Figure 3 displays the temperature reduction in the cold tube against the cold mass fraction for the present tube, Hilsch's tube and Guillaume and Jolly's tube. It is worth noting that the temperature reduction distributions for all tubes show a similar trend despite different tube sizes and inlet conditions. All tubes yield a maximum temperature reduction at the cold mass fraction between 0.3 and 0.4, having a maximum temperature decrease value at about 16 ^{0}C. A close examination reveals that when using a single tangential inlet nozzle, the temperature drop profile of the present tube is very close to that of Hilsch's tube[1] but is slightly different with Guillaume and Jolly's tube[14].

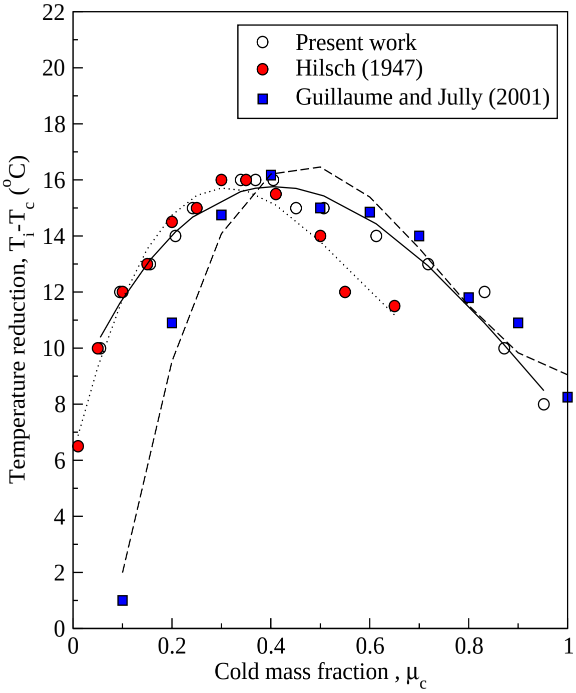


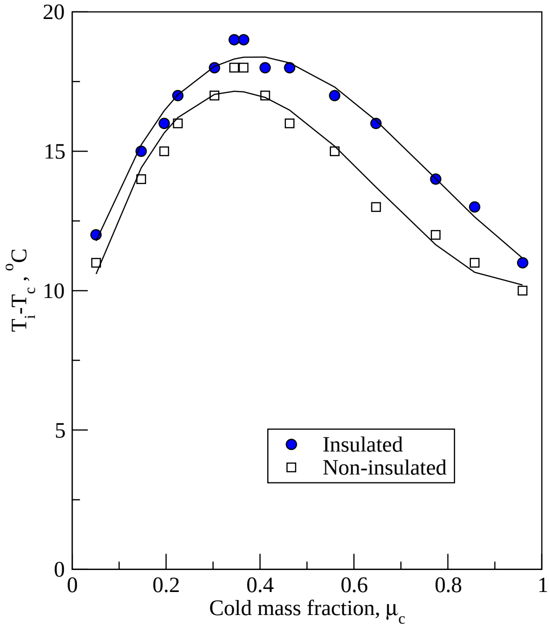
Fig 3. Comparison of temperature reduction in the cold tube for the present work and previous work.

This comparison has been made to help increase the confidence in the present measurements only.

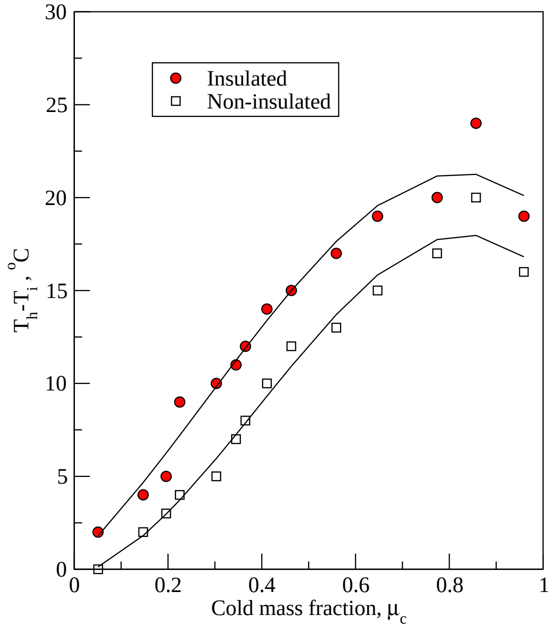
Results and Discussion

Temperature Reduction

In the present experiment, measurements of the vortex tube with and without insulation at inlet



(a) Cold tube



(b) Hot tube

Fig 4. Effect of the insulated and non-insulated tubes on temperature variation in (a) cold tube and (b) hot tube, T_i =29°C.

temperature and pressure of 29 ^{0}C and 3.5 bar respectively were made for the cold orifice diameter of 0.5D using the single inlet nozzle. The outside surface temperature of the non-insulated hot tube exposing to the surrounding was about 50 - 60 ^{0}C, but this reduced to some 32 ^{0}C when using the insulated tube. The temperature differences between the inlet and the tube temperatures against various cold mass fractions are shown in Figures 4a and 4b for the cold and hot tubes, respectively. In the figure, the insulated tube provided a higher temperature reduction than the non-insulated one. At a cold mass fraction of 0.345, the highest temperature reductions at the cold tube of the insulated and non-insulated tubes were 19 and 18 ^{0}C, respectively. In addition, in the hot tube the maximum temperature increases of the insulated and non-insulated tubes were found to be 24 and 20 ^{0}C, respectively, at a cold mass fraction of 0.857. The average temperature differences between the insulated and non-insulated tubes were in a range of 2 to 3 ^{0}C for the cold tube, and of 2 to 5 ^{0}C for the hot tube. This is because the insulated tube gave a less energy loss to the surroundings than the non-insulated one, causing the higher temperature difference within the tube. For the cold mass fraction ranging from 0.1 to 0.4, the temperature reduction in the cold tube increased, but then decreased for the cold mass fraction over 0.4. In the hot tube, for the cold mass fraction ranging from 0.1 to 0.8, the temperature proportionally increased, but then decreased rapidly for the cold mass fraction above 0.8.

The experimental result of temperature drops for different cold orifice diameters ranging from 0.4D to 0.9D using one inlet nozzle is depicted in Figure 5. The highest temperature reduction was achieved for all cold orifice diameters when the cold mass fraction was in a range of 0.3 to 0.4. Thus, the maximum temperature drop occurred if the cone valve was adjusted to let the cold mass flow rate leave the cold tube at 30 to 40% of the inlet air. The decrease in temperature in the cold tube was found to be 18, 19, 15, 14, 12, and 10 ^{0}C for using cold orifice diameter of 0.4D, 0.5D, 0.6D, 0.7D, 0.8D and 0.9D at the cold mass fraction of 0.364, 0.375, 0.381, 0.378, 0.373, and 0.372, respectively. Furthermore, the cold orifice diameter of 0.5D yielded the highest potential of temperature reduction in the cold tube than the others. Using the cold orifice diameter ranging from 0.6D to 0.9D (bigger than that of 0.5D) would allow some hot air in vicinity of the tube wall to exit the tube with the cold air. Both the hot air and cold air as flowing out were mixed together which further affected the cold air to have higher temperature. On the other hand, for a small cold orifice diameter of 0.4D, it has a higher back pressure and makes the temperature reduction at the cold tube lower.

The effect of the number of inlet nozzles on temperature reduction in the insulated vortex tube was experimentally investigated as shown in Figure 6. The increase in the number of inlet nozzles led to considerable temperature separation. In the figure, the use of 4 inlet nozzles resulted in a higher temperature

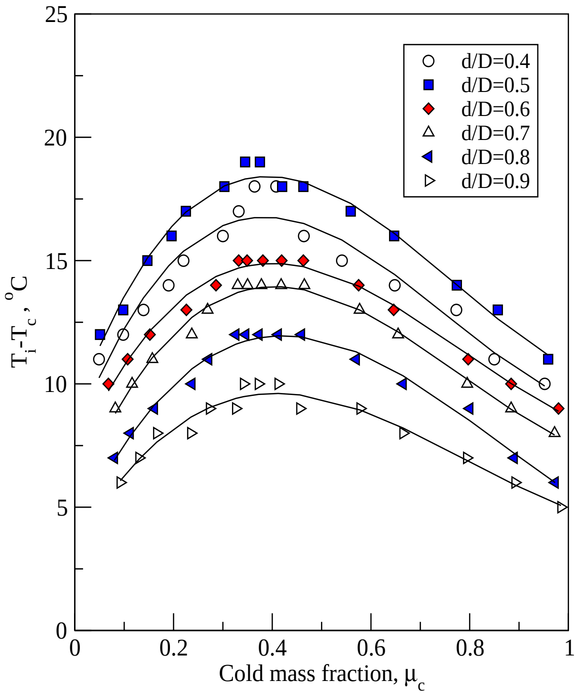


Fig 5. Effect of the cold orifice diameters on temperature reduction in the insulated vortex tube, T_i =29°C.

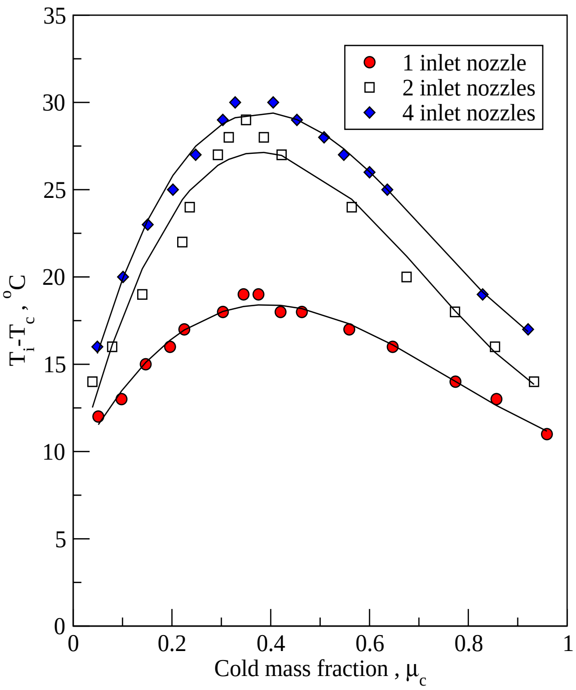


Fig 6. Effect of the number of inlet nozzles on temperature reduction in the insulated vortex tube, T_i =29°C.

reduction in the cold tube than that of 1 and 2 inlet nozzles for the cold orifice diameter of 0.5D. The highest temperature drops also were 19, 29 and 30 °C for using 1, 2 and 4 nozzles, respectively. Mostly the maximum thermal separation occurred at a cold mass fraction between 0.3 and 0.4. Changing the number of inlet nozzles from 1 to 2 and 4 helped to speed up the flow and to increase the mass flow rate and strong swirl flow into the vortex tube. In addition, this gave rise to higher friction dissipation between the boundary of the flows and a higher momentum transfer from the core region to the wall region. This reduced temperature in the tube core while increased temperature in the tube wall area.

Wall Temperature Distribution

Wall temperatures of the hot tube were measured at 15 axial stations equally spaced along the axial distance downstream of the cold orifice plate as can be seen in Figure 7. Figure 8 displays the wall temperature distributions in terms of temperature difference between the wall and the inlet at different cold mass fractions. In a range from x/D=1 to x/D=11, the wall temperature distribution tended to increase, reaching a maximum at x/D=11. After x/D=11, the wall temperature distribution tended to decrease because the hot air near the tube wall region and the cold air in the core region were mixed together due to decaying of swirl flow close to the exit of hot tube. The temperature of the tube wall increased proportionally with the cold mass fraction, except when the cold mass fraction approached unity. It can be observed that at x/D=11, the temperature at the tube wall with 4 inlet nozzles was 78 °C above the inlet temperature for the cold mass fractions of 0.829.

Isentropic Efficiency

There was a substantial increase in the isentropic efficiency and the behavior of the efficiency with cold

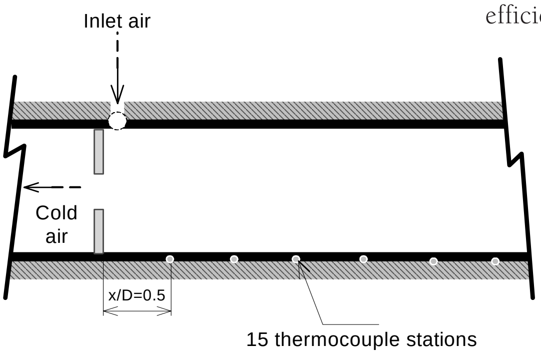


Fig 7. Arrangement of thermocouples along the hot tube wall.

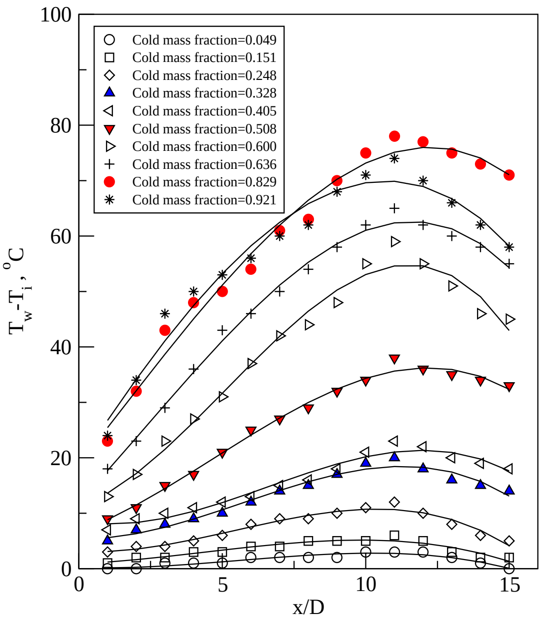


Fig 8. The axial wall temperature distribution along the hot tube of the insulated vortex tube with 4 inlet nozzles and cold orifice of 0.5D, T_i =29°C.

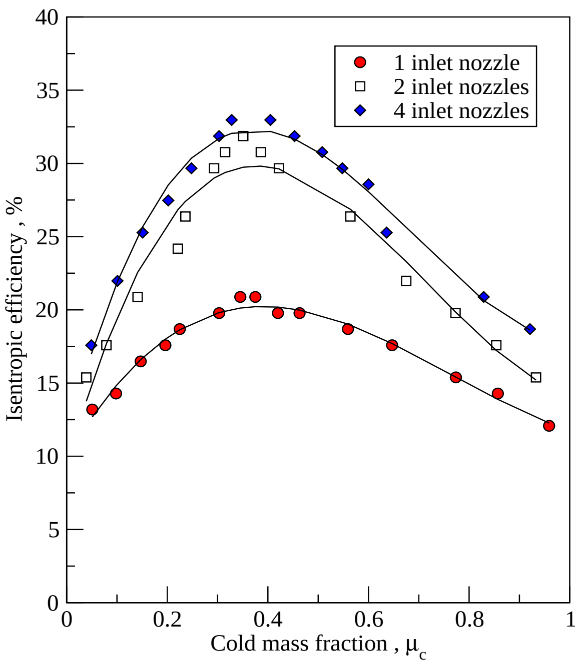


Fig 9. Relationship between isentropic efficiency and the cold mass fraction for the insulated vortex tube.

mass fraction at a different number of inlet nozzles is depicted in Figure 9. The use of 4 inlet nozzles generated a stronger swirling flow and momentum transfer than that of 1 and 2 inlet nozzles, resulting in higher thermal separation and efficiency. The isentropic efficiency for the 2 and 4 nozzles could be improved and increased around 20% and 65% better than that for the single nozzle, respectively. Moreover, the maximum isentropic efficiency of an insulated vortex tube with 4 nozzles and cold orifice diameter of 0.5D was found to be 33%.

Empirical Correlation

In the experimental optimization, the temperature reduction data using 1, 2 and 4 inlet nozzles were selected for comparison with Hilsch's work and Guillaume and Jolly's work as can be seen in Figure 10. In the figure, the maximum cold air temperature drop or thermal separation took place at the cold mass fraction ranging from 0.3 to 0.4. With the help of experimental data, the following empirical correlations of the insulated vortex tube with different inlet nozzles can be expressed:

Following present work (1 inlet nozzle), and a cubic fitting yields,

$$\frac{\Delta T_c}{\Delta T_{c,max}} = 2.0046\mu_C^3 - 4.8477\mu_C^2 + 2.8563\mu_C + 0.4641 \quad (7)$$

Following present work (2 inlet nozzles), and a cubic fitting yields,

$$\frac{\Delta T_c}{\Delta T_{c,max}} = 1.0916\mu_C^3 - 4.2581\mu_C^2 + 3.101\mu_C + 0.3499 \quad (8)$$

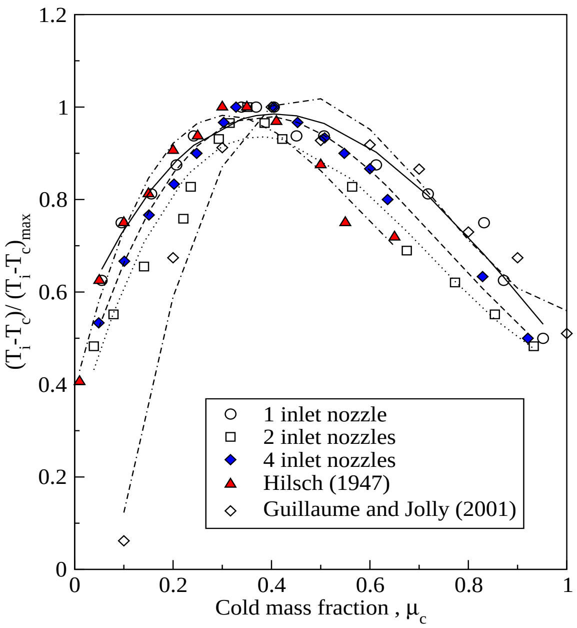


Fig 10. Comparison between non-dimensional temperature reduction with cold mass fraction of previous work and the present work.

Following present work (4 inlet nozzles), and a cubic fitting yields,

$$\frac{\Delta T_c}{\Delta T_{c,max}} = 3.79\mu_C^3 - 8.3\mu_C^2 + 4.65\mu_C + 0.21 \quad (9)$$

The above correlations can be used to help design a vortex tube system to obtain the highest temperature separation in the vortex tube with insulation.

Following Hilsch (1947), and a cubic fitting yields,

$$\frac{\Delta T_c}{\Delta T_{c,max}} = 5.78\mu_C^3 - 9.8\mu_C^2 + 4.40\mu_C + 0.39 \quad (10)$$

Following Guillaume and Jolly (2001), and a cubic fitting yields,

$$\frac{\Delta T_c}{\Delta T_{c,max}} = 5.73\mu_C^3 - 12.7\mu_C^2 + 8.1\mu_C + 0.56 \quad (11)$$

Empirical equations (7), (8) and (9) are derived from the present experimental results showing the range of highest temperature separation in the vortex tube at the cold mass fraction required. When compared to empirical correlations (10) and (11) obtained from previous works, they show similar correlations.

Conclusions

An experimental study on the temperature separation in the vortex tube has been carried out and this research finding can be summarized as follows:

1. The increase of the number of inlet nozzles led to higher temperature separation in the vortex tube.
2. Using the tube with insulation to reduce energy loss to surroundings gave a higher temperature separation in the tube than that without insulation around 2-3 ^{0}C for the cold tube and 2-5 ^{0}C for the hot tube.
3. A small cold orifice (d/D=0.4) yielded higher backpressure while a large cold orifice (d/D=0.7, 0.8, and 0.9) allowed high tangential velocities into the cold tube, resulting in lower thermal/energy separation in the tube.

Nomenclature

P_a	atmospheric pressure [bar]
P_i	inlet pressure [bar]
ΔT	temperature difference [^{o}C]
ΔT_c	temperature reduction [$T_i - T_c$, ^{o}C]
$\Delta T_{c,max}$	maximum temperature reduction [$T_i - T_{c,max}$, ^{o}C]
ΔT_h	temperature increasing [$T_h - T_i$, ^{o}C]
T_i	inlet air temperature [K]
T_c	cold air temperature at the cold tube [K]

T_h	hot air temperature at the hot tube [K]
T_w	hot wall temperature at the hot tube [K]
d	diameter of cold orifice [mm]
D	diameter of the vortex tube [mm]
L	length of the tube [mm]
μ_c	cold mass fraction [$\dot{m}_c / \dot{m}_i$]
$\dot{m}_i$	inlet mass airflow rate [kg/s]
$\dot{m}_c$	cold mass airflow rate [kg/s]
δ	diameter of inlet nozzle [mm]
γ	specific heat ratio
η_{is}	isentropic efficiency [%]

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