EFFECT OF THE OUTLET NOZZLE DIAMETER ON THE PERFORMANCE OF DIVERGENT VORTEX TUBE

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ABSTRACT

The aim of the present work is to study the effect of different outlet nozzle diameters, by using one or two nozzles on the performance of divergent vortex tube and also to determine the effects of various parameters on the vortex tube cooling performance such as: refrigeration capacity, coefficient of performance, isentropic efficiency. The experimental investigation was carried out on a divergent vortex tube rig manufactured for the present study covering all tests. The effect of different outlet nozzle diameters (d_n= 4 mm, d_n = 5 mm, and d_n= 6.5 mm) on the performance of the vortex tube is described by using one nozzle or two nozzles by varying the pressure of the inlet air and cold air mass ratio (μ_c) within the ranges ($P_{i\ abs}$=2 - 7 bar) and (μ_c= 0 - 1). The outlet nozzle diameter, (d_n= 5 mm), and the outlet cold diameter (d_c= 10 mm), when using two nozzles, give high temperature separation and may considered to be the optimum for different pressures of the inlet air regarding all the size tube diameter. The experimental study predicts two empirical results between the outlet nozzle diameter (d_n), number of nozzles (N), inside vortex tube diameter (D), and length of vortex tube (L) as:

$$\frac{N d_n^{\ 2}}{D^2} = 0.105 \qquad \text{for (N=1) and;}$$

$$\frac{N d_n^{\ 2}}{D^2} = 0.211 \qquad \text{for (N=2)}$$

الخلاصة

الهدف من البحث الحالي هو دراسة تاثير القطر الداخلي للمنافث باستخدام ,منفث واحد ، ومنفثان على أداء الأنبوب الدوامي المنفرج ، وكذلك دراسة مختلف العوامل التي توثر على الأداء التبريدي مثل سعة التبريد ، معامل الأداء ، الكفاءة الايزنتروبية .أنجزت التجارب من خلال بناء جهاز متكامل للأنابيب الدوامة المنفرجة ، صمم

وصنع خصيصاً لهذا البحث، حيث غطى جميع التجارب. تمت دراسة تأثير تغير القطر الداخلي للمنافث (d_n= 4 mm , 5 mm , 6.5mm) وذلك باستخدام منفث واحد ، ومنفثان ، على أداء الأنبوب الدوامي المنفرج ، مع تغير الضغط المطلق للهواء الداخل ونسبة كتل الهواء البارد الى الهواء الداخل (μ_c) ضمن المدايات (2-7 bar) و (μ = 0 -1) على التوالي. عند استخدام فتحة المنفث (d_n = 5 mm) وقطر فتحة البارد الخارج (d_c = 10 mm) وباستخدام منفثان تعطي اعلى فصل في درجة الحرارة ويمكن اعتبراها الامثل لضغوط المختلفة الداخلة الى الانبوب الدوامي . تم الحصول على علاقتين تجريبية تربط قطر المنفثات ، عدد المنافث ، القطر الداخل للأنبوب (D)، طول الأنبوب. (L) بحيث تعطي افضل تصميم للأنبوب الدوامي المنفرج ذو طاقة فصل عالية، كالتالي :

$$\frac{N d_n^{\,2}}{D^2} = 0.105 \qquad N=1$$

$$\frac{N d_n^{\,2}}{D^2} = 0.211 \qquad N=2$$

KEY WORDS

Counter Flow Vortex Tube , Thermal / temperature Separation , Design Parameters

INTRODUCTION

A vortex tube uses compressed air as a a power source, has no moving parts and produces hot air from one end and cold air from the other, as shown in **Fig.1**.

There is one widely accepted explanation of the phenomenon: **(Prasad 1963 , vortex tube theory 2000).** Compressed air is supplied to the vortex tube and passes through nozzles that are tangent to the internal counterbore. These nozzles set the air in a vortex motion. The spinning stream of air turns and passes down the hot tube in the-form of a spinning shell, similar to a tornado. A valve at one end of the tube allows some of the warmed air to escape. What does not escape, heads back down the tube as a second vortex inside the low-pressure area of the larger vortex. The inner vortex losses heat and exhausts through the other end as cold air.

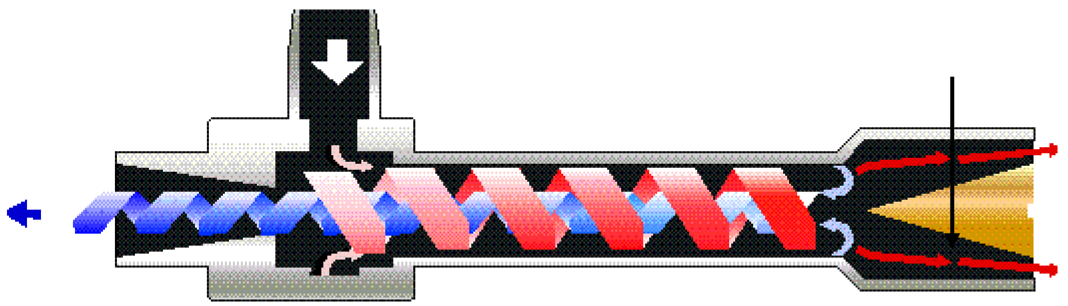

Fig.1: Diagrammatic view of the pattern in a counter-flow tube .

While one air stream moves up the tube and the other down it, both rotate in the same direction at the same angular velocity. Due to the principle of conservation of angular momentum the speed of the inner vortex remains the same. Angular momentum has been lost from the inner vortex. The energy lost shows up as heat in the out vortex. Thus the outer vortex becomes warm, and the inner vortex is cooled.

A number of researches mention the effects of tapering the vortex tube, with conflicting results. **(Martynovskii and Alekseev 1957)**, find a contracting hot tube to be preferable and get optimum performance for (40 < L/D < 50) which is comparable to **(Hilsch's 1947)** comment that (L/D) should be around (50) for good temperature separation.

(Raiskii and Tunkel 1974) investigated the influence of the vortex tube configuration (cylinder, diffuser and step), on the vortex-temperature gas-separation process. Results show that long cylindrical tubes are most effective in a broad range of variations of structural and modal parameters, noting that the majority of the energy transfer appears to occur within the first five diameters of the tube.

(Soni and Thomson 1975) employed a systematic method of experimental design to determine the optimum performance of a vortex tube as a function of the pertinent design parameters, especially the individual experiments were stipulated according to the simplex modification of the method of evolutionary operations, **(EVOP 1962).**

The results indicate the following optimum design parameters for maximum (ΔT_c) and maximum isentropic efficiency (η_{ise}).

A_N (Nozzle / tube area) = 0.11 ± 0.01 (0.08 ± 0.001)

A_Φ (Orifice area / tube area) = 0.08 + 0.01 (0.145 ± 0.035)

(Takahama and Yakosawa 1981) investigated the measurements of the swirling flows inside the divergent chambers. The results show that the shorter divergent chambers (L=900 mm) have a higher angular velocity near the center of the back flow and the swirl intensity increases in divergent chambers in the direction of flow with relatively large divergent angles.

(Mitushen and Mohammed 1992), part I, designed " Laser Doppler Velocimeter" that provides three velocity components of the flow inside the counter flow vortex tube. Results indicate a reverse flow from the axial velocity profiles in the central region (r < 4 - 5) mm of the vortex tube and the maximum axial velocity was is to the tube wall.

(AL-Abriy 1997) carried out an experimental study on the effect of tangential nozzles number on the performance of vortex tube by varying the number of tangential nozzles to become between (1-8) nozzles under inlet pressure of (3-7) bar at cold air mass fractions of (μ_c = 0-1).

A laboratory device, "Hilton Co. Manufacture" and rig employed by **(Al-Jielawe 1994)** were used to carry out the experiments studying.

Results show that there is an optimum value of number of nozzles (N = 8 and d_c = 0.8 mm) to get high performance.

(AL-Barwari 2004) studied the experimental study on the thermodynamic properties of vortex tubes with a divergent chamber. The experimental investigation was carried on a divergent vortex tube rig especially designed for the present study covering all tests. The effect of different cold outlet diameters (d_c=10 mm, d_c=12 mm, and dc=14 mm) on the performance of the vortex tube is described .The results show a pronounced influence of divergent vortex tube (θ°= 1.72, L= 460 mm), on the energy separation performance.

REFRIGERATION PERFORMANCE

Performance of simple refrigeration plant is usually characterized by means of the Coefficient of Performance (C.O.P) defined for the cyclic work-absorbing device **Fig.2** as **(vortec corporation catalogue 1992).**

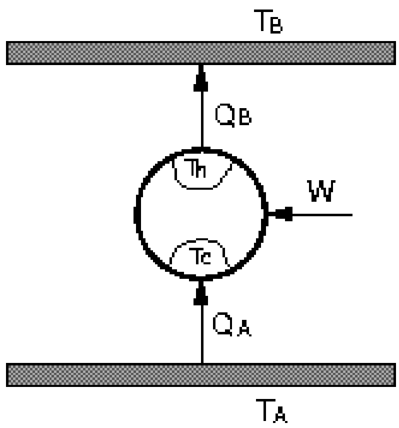

Fig (2) cyclic work absorbing device.

$$COP = \frac{Q_A}{W_{act}} \quad (1)$$

The process of ideal compression of the inlet air mass from the atmospheric pressure (P_a) to high pressure (P_i) is represented by the line (T_i - T_f) in **Fig.3** and the actual compression is represented by the line ($T_i - T_{f'}$) . [13]

The COP of the vortex tube is defined as the ratio of the actual cooling effect to the work input to the air-compressor.

The compressed air has to be cooled at constant pressure in a cooler to the initial temperature (T_i) before being sent to the vortex tube. The refrigeration capacity (actual cooling) of the vortex tube is simply equal to Q_r.

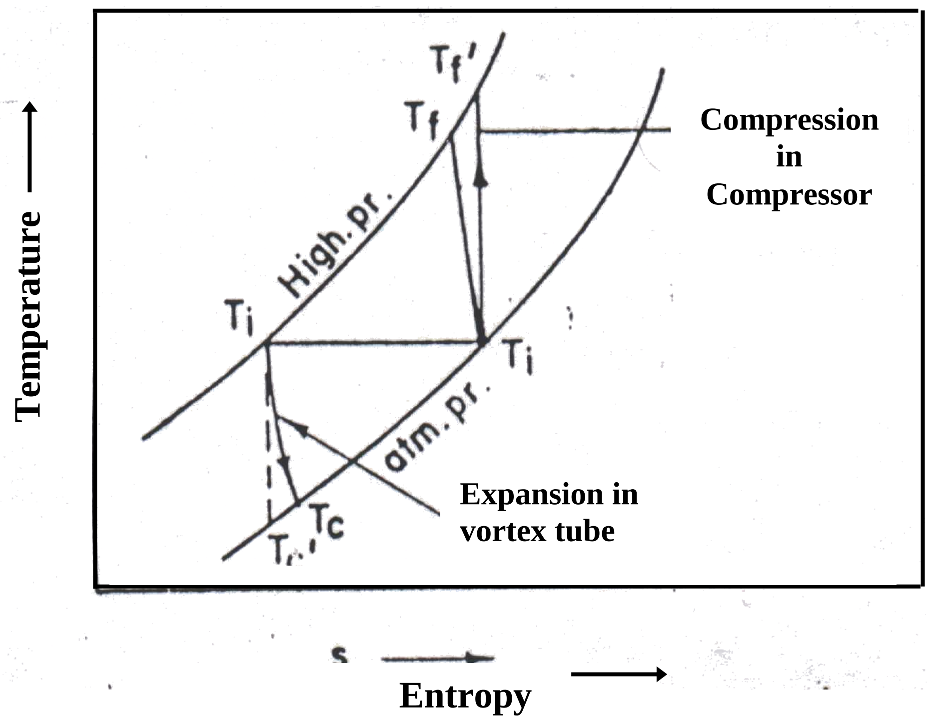

Fig.3: Compression and expansion

process in a vortex tube

$$Q_r = m_c\, C_p\, (T_i - T_c) \qquad \textbf{(2)}$$

Thus:

$$COP = \frac{m_c\, C_p\, \Delta T_c\, \eta_{comp}}{m_i C_p T_i \left[\left(\frac{P_i}{P_a} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right]} \qquad (3)$$

If the process undergoes an isentropic expansion from the inlet pressure (P_i) to atmospheric pressure (P_a) at the cold end then the static temperature drop due to expansion is given by:

$$\Delta T_{c'} = T_i - T_{c'} = T_i \left[1 - \left(\frac{P_a}{P_i} \right)^{\frac{\gamma-1}{\gamma}} \right] \qquad (4)$$

The actual temperature drop due to expansion occurred in vortex tube is ΔT_c the ratio of ΔT_c to $\Delta T_{c'}$ is called " Relative Temperature Drop".

$$R.T.D = \frac{\Delta T_c}{\Delta T_{c'}} \qquad (5)$$

The adiabatic (isentropic) efficiency of the vortex tube is defined as:

$$\eta_{ise} = \frac{\textit{Actual}\ \text{Cooling effect obtained in the vortex tube}}{\text{Ideal cooling effect possible with adiabatic expansion}}$$

$$\eta_{ise} = \mu_c \frac{\Delta T_c}{\Delta T_{c'}} \qquad (6)$$

Sub. **Eq. (4)** in **(6)** one gets:

$$COP = \mu_c \left(\frac{\Delta T_c}{\Delta T_{c'}}\right) \eta_{comp} \left(\frac{P_a}{P_i}\right)^{\frac{\gamma-1}{\gamma}} \qquad (7)$$

By substituting the value of $\left(\mu_c \frac{\Delta T_c}{\Delta T_{c'}} \right)$ from **Eq.(6)**, in **Eq.(7)**, one gets:

$$COP = \eta_{ise}\ \eta_{comp} \left(\frac{P_a}{P_i}\right)^{\frac{\gamma-1}{\gamma}} \qquad (8)$$

EXPERIMENTAL APPARATUS AND RIG LAYOUT

The test rig is designed and manufactured to fulfill the requirements of the test system for different divergent vortex tube. The main rig, and the apparatus of the system illustrated in **Figs. 4** and **5** and photograph (1).

RESULTS AND DISCUSSION

Effect of cold air mass ratio (μ_c) on the temperature difference ($\Delta T_h, \Delta T_c$), isentropic efficiency (η_{ise}) coefficient of performance (COP), and refrigeration capacity (Qr), for the inlet air pressure change and outlet nozzle diameter (d_n) , are investigated .

Fig.6 at ($P_{i\ abs}$ = 7 bar), number of nozzle = 1, and, (μ_c =0.408), shows, that the best energy separation, i.e highest temperature separation is for the outlet nozzle diameter (d_n=6.5 mm), and (d_c = 10 mm), in respect with all selected outlet nozzle diameters, which give the minimum values of the cold air temperature differences (ΔT_c),as shown in **Table (1)**

Table (1): Results of One Nozzle

Number of nozzles = 1, D= 20 mm, L/D = 23, L = 460, (θ° = 1.72), T_{in} = 30 °C d_c=10 mm					
d_n = 4 mm		d_n = 5 mm		d_n = 6.5 mm	
Minimum T_c (°C)	Maximum T_h (°C)	Minimum T_c (°C)	Maximum T_h (°C)	Minimum T_c (°C)	Maximum T_h (°C)
1.2	55.2	-14.5	71.2	-20	72.3

Fig.7, at ($P_{i\ abs}$ = 7 bar), (number of nozzles = 2), (μ_c= 0.409). shows clearly, that the best performance is for the outlet nozzle diameter (d_n = 5 mm), and (d_c= 10 mm), which gives the minimum values of the cold air temperature .i.e maximum temperature differences (ΔT_c), and the maximum value of the hot temperature, i.e the highest value of the hot temperature, as shown in **Table (2):**

Table (2): Results of Two Nozzle

Number of nozzles = 2, D = 20 mm, L/D = 23, L = 460, (θ° = 1.72), T_{in} = 30°C d_c = 10 mm					
d_n = 4 mm		d_n = 5 mm		d_n = 6.5 mm	
Minimum T_c(°C)	Maximum T_h(°C)	Minimum T_c(°C)	Maximum T_h(°C)	Minimum T_c(°C)	Maximum T_h(°C)
-20.2	70.5	-35.5	81.2	-34.1	77.5

In **Figs. 8, 9** and **10** samples are selected to illustrate the effect of outlet nozzle diameter on isentropic efficiency with two nozzles at outlet cold diameter (dc=10 mm) as illustrated in **Table (3)** for different outlet nozzles diameters.

Table (3): Isentropic Efficiency Results

One nozzle				Two nozzles			
d_c= 10 mm				d_c= 10 mm			
	d_n= 4 mm	d_n= 5 mm	d_n= 6.5 mm	d_n= 4 mm	d_n=5 mm	d_n= 6.5 mm	$P_{i\ abs}$

η_{ise} %	10.58	18.25	22	22.9	28.2	27.9	2
μ_c	0.601	0.718	0.706	0.714	0.703	0.714	
η_{ise} %	6.22	13.38	18.1	18.8	20.6	21.8	7
μ_c	0.607	0.617	0.707	0.715	0.702	0.713	

In **Figs. 11, 12** and **13** samples are selected to show the effect of changing (μ_c) on (COP) of vortex tube (θ°= 1.72), at (L/D= 23) for different pressure ranges between (2-7) bar, with two nozzles at outlet cold diameter (d_c = 10 mm).

The best values were acquired for the above mentioned outlet nozzle diameters, as illustrated in **Table (4):**

Table (4): Effect of Cold Air Ratio Optimum Results of COP

One nozzle				Two nozzles			
d_c=10 mm				d_c=10 mm			
	d_n=4 mm	d_n=5 mm	d_n=6.5 mm	d_n=4 mm	d_n=5 mm	d_n=6.5 mm	$P_{i\ abs}$
COP	0.05	0.094	0.126	0.127	0.180	0.159	2
μ_c	0.701	0.616	0.606	0.714	0.712	0.604	
COP	0.035	0.079	0.09	0.094	0.125	0.108	7
μ_c	0. 702	0.612	0.605	0.712	0.711	0.603	

Figs. 14, 15 and **16**, show the effect of changing (μ_c) on the refrigeration capacity for vortex tube (θ°= 1.72) at (L/D= 23) for the pressure range between ($P_{i\ abs.}$= 2-7) bar, with two nozzles at outlet cold diameter (d_c = 10 mm).

The best values acquired for the above mentioned outlet nozzle diameters at highest pressure ($P_{i\ abs}$=7bar) and minimum at pressure ($P_{i\ abs}$= 2 bar), are shown in **Table (5):**

Table (5): Effect of Cold Air Ratio Optimum Results of Qr

One nozzle				Two nozzles			
d_c=10 mm				d_c=10 mm			
	d_n=4 mm	d_n=5 mm	d_n=6.5 mm	d_n=4 mm	d_n=5 mm	d_n=6.5 mm	$P_{i\ abs}$
Qr, watt	15.7	33.25	60.03	54.81	150.5	149.51	2
μ_c	0.601	0.616	0.706	0.718	0.714	0.714	
Qr, watt	308	610	1180	1284	1875.8	1866	7
μ_c	0.611	0.617	0.712	0.716	0.713	0.714	

CONCLUSIONS

- The outlet nozzle diameter (d_n= 6.5 mm), and the outlet cold diameter,(d_c= 10), when using **one nozzle** give a high temperature separation and may considered to be the optimum for different pressures of the inlet air regarding the size of tube diameter.

- The outlet nozzle diameter, (d_n= 5 mm), and the outlet cold diameter (d_c= 10 mm), when using **two nozzles**, give high temperature separation and may considered to be the optimum for different pressures of the inlet air regarding all the size tube diameter.

- The isentropic efficiency of the divergent vortex tube (θ°= 1.72),(L/D = 23), (L = 460 mm), (d_c = 10 mm), (d_n = 6.5 mm), when uesing one nozzle and two nozzles, increase with cold air mass ratio (μ_c) and reaches the highest value at (μ_c = 0.6-0.72). On the other hand they decrease with the inlet air pressure increase. Divergent vortex tube (θ°= 1.72) has a highest isentropic efficiency attain: (η_{ise} = 27.9 %) at (μ_c = 0.714), when using **two nozzles**.

- The coefficient of performance (COP) of the divergent vortex tube (θ°= 1.72), (L/D = 23), (L = 460 mm), (d_c = 10 mm), (d_n = 6.5 mm), when using one nozzle and two nozzles, increase with cold air mass ratio (μ_c) and reaches the highest value at (μ_c = 0.6-0.72). On the other hand they decrease with the inlet air pressure increase. Divergent vortex tube (θ°= 1.72) has a highest coefficient of performance attain: (COP = 0.159 %) at (μ_c = 0.604), when using **two nozzles.**

-The refrigeration capacity of the divergent vortex tube (θ°= 1.72) when using one nozzle and two nozzles is increased with cold air mass ratio (μ_c) till reaching the highest value at (μ_c = 0.6 - 0.72). They increase with the inlet air pressure increase and has a higher refrigeration capacity attain: (Qr = 1866 Watt) at (μ_c = 0.714), when use **two nozzles**.

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NOMENCLATURE

A_c , A_h	Cross sectional area of the cold end orifice and hot end of the outlet air	`m^2
A_i	Cross sectional area of inlet pipe	m^2
COP	Coefficient of performance	-
Cp	Specific heat at constant pressure	J/kg. K
D	Inner diameter of vortex tube	mm
d_c	Diameter of cold end orifice	mm
d_h	Diameter of hot end outlet	mm
d_n	Diameter of nozzle outlet	mm
h_i	Specific enthalpy of the inlet air	J/kg
L	Length of vortex tube	mm
L/D	(Length / inner diameter) of the vortex tube	-
$\dot{m}$	Mass flow rate per nozzle	kg/s
$\dot{m}_I$, $\dot{m}_c$, $\dot{m}_h$	Mass flow of inlet air, cold air and hot air	kg/s
P_a	Ambient pressure	bar
P_i , P_c , P_h	Pressure of the inlet air, cold air and hot air	bar
Q_r	Refrigeration capacity	Watt
Q	Heat exchanged between the system and its surrounding	Watt
T_a	Ambient temperature	^oC
T_i, T_c , T_h	Temperature of the inlet air, cold air and hot air	^oC
T_o	Temperature at the nozzle outlet	^oC

ΔT_c	Temperature difference of cold air = $T_c - T_i$	^oC
$\Delta T_{c'}$	Isentropic (maximum) temperature difference	^oC
ΔT_h	Temperature difference of the hot air = $T_h - T_i$	^oC
η_{comp}	The isentropic efficiency of the compressor	%
η_{ise}	Isentropic efficiency of the vortex tube	%
θ	Angle of divergence	degree
μ_c	Cold air mass ratio = $\frac{m_c}{m_i}$	-
ρ	Density of the air	kg/m^3
c, h, i	Subscripts refer to cold, hot and inlet air	-

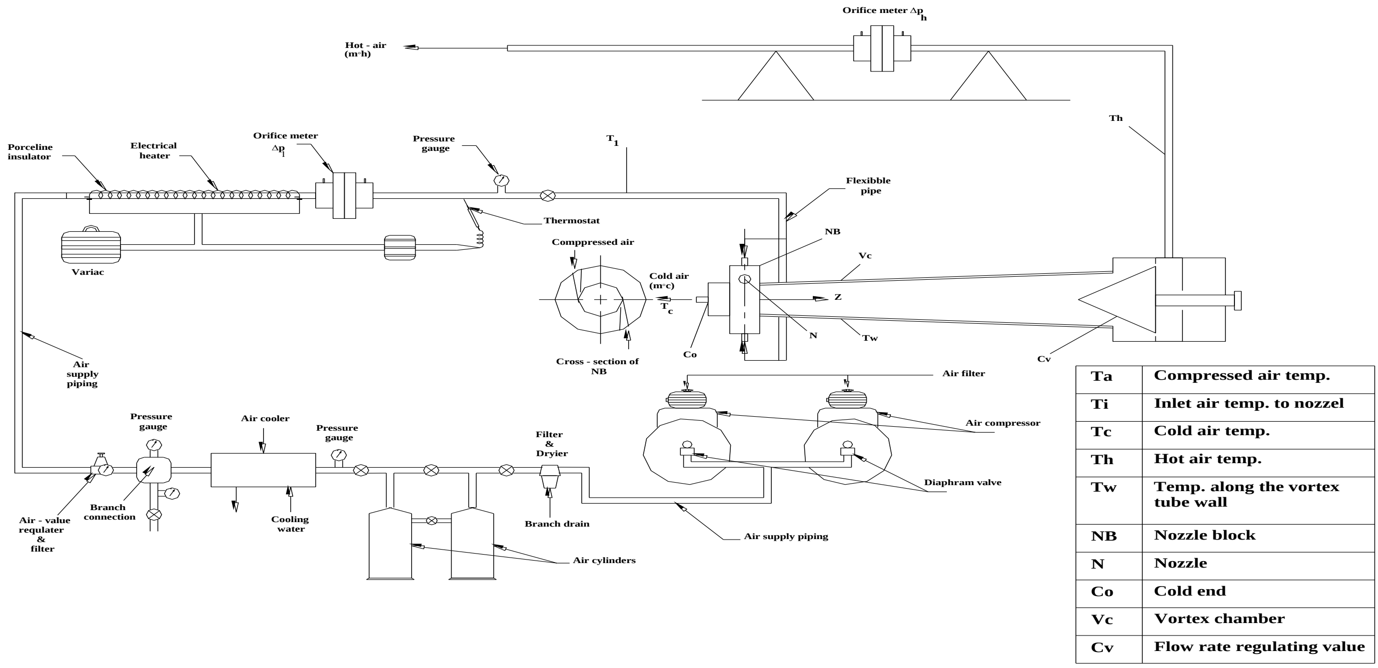

Fig. (4) Schematic diagram of the experimental rig.

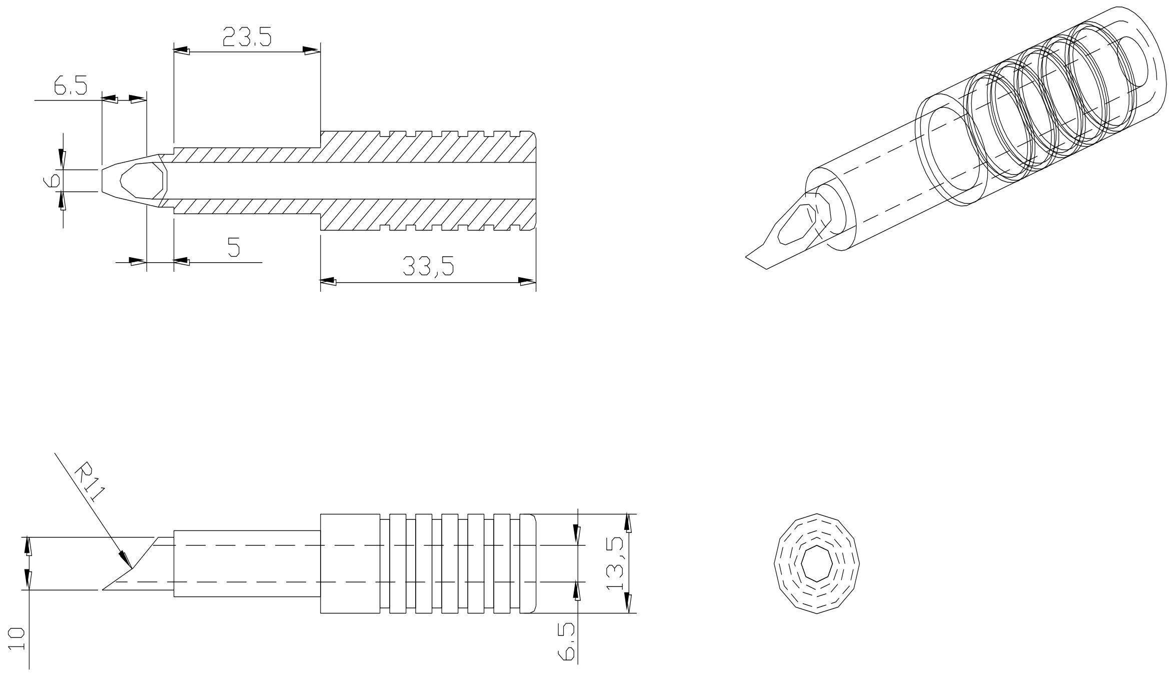

Fig. (5) Nozzle

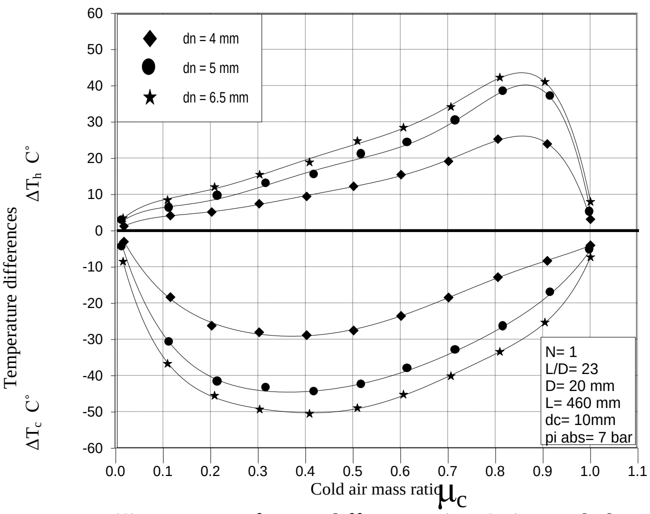

Fig. (6) Variation of temp. differences ΔT_c & ΔT_h with the mass ratio(μ_c)for vortex tube (θ°=1.72) ,with outlet diameter (d_n) of the vortex tube as a parameter.

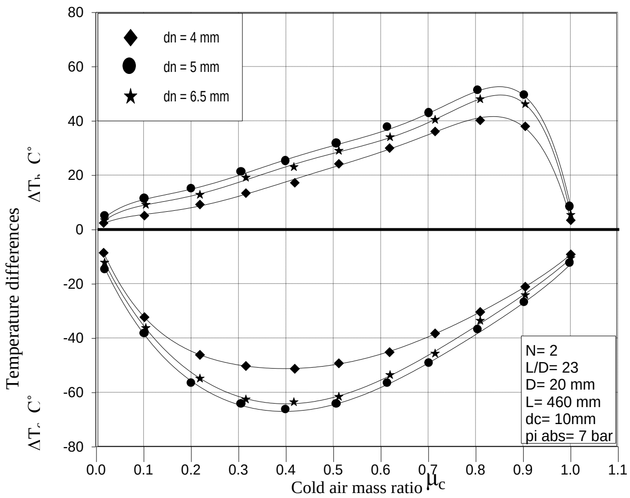

Fig. (7) Variation of temp. differences ΔT_c & ΔT_h with the cold air mass ratio(μ_c)for vortex tube (θ°=1.72) ,with outlet nozzle diameter (d_n) of the vortex tube as a parameter.

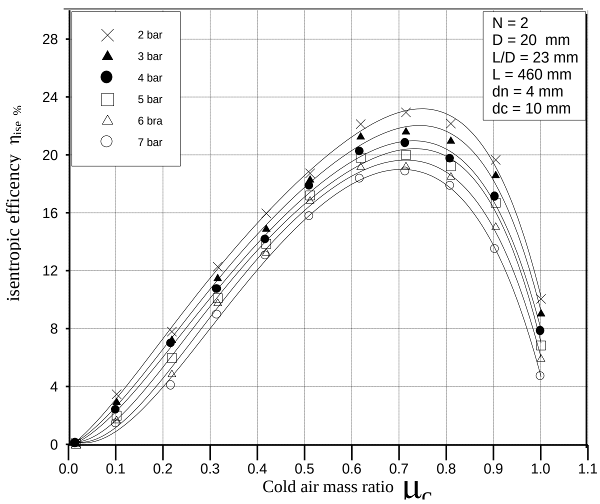

Fig. (8) The relation between isentropic efficiency (η_{ise}) and cold air mass ratio (μ_c) for vortex tube (θ° =1.72) , with the absolute pressure (p_i) of the inlet air as parameter.

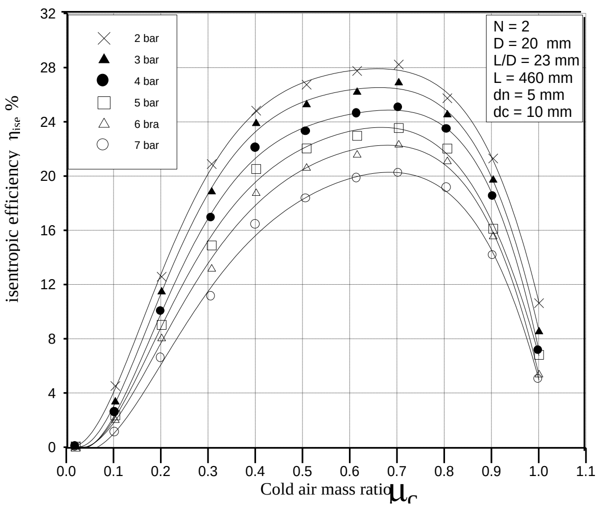

Fig. (9) The relation between isentropic efficiency (η_{ise}) and cold air mass ratio (μ_c) for vortex tube (θ° =1.72) , with the absolute pressure (p_i) of the inlet air as parameter.

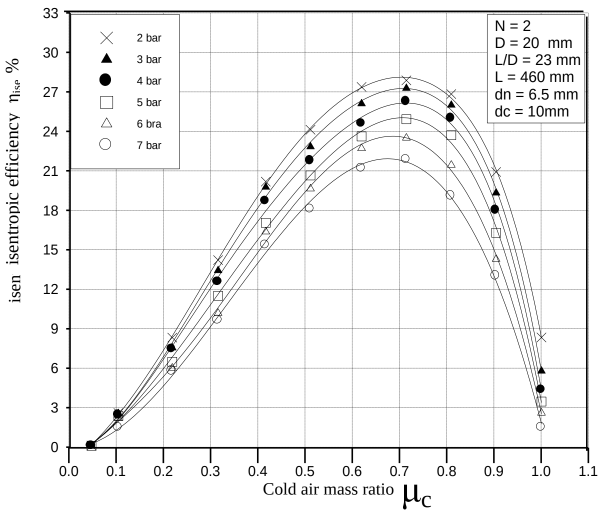

Fig. (10) The relation between isentropic efficiency (η_{ise}) and cold air mass ratio (μ_c) for vortex tube (θ° =1.72) , with the absolute pressure (p_i) of the inlet air as parameter.

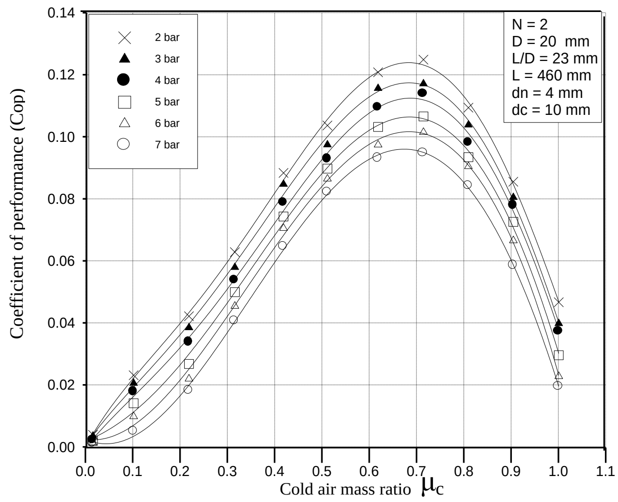

Fig. (11) The relation between coefficient of performance (Cop) cold air mass ratio (μ_c) for vortex tube (θ°= 1.72), with the absolute pressure (p_I) of the inlet air as a parameter

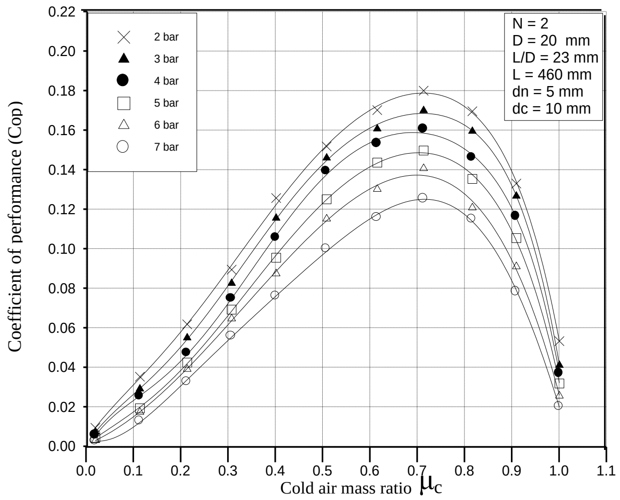

Fig. (12) The relation between coefficient of performance (Cop) and cold air mass ratio (μ_c) for vortex tube (θ°= 1.72), with the absolute pressure (p_I) of the inlet air as a parameter

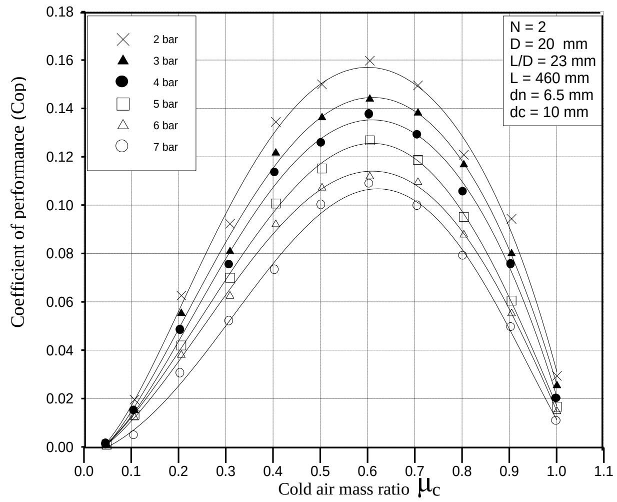

Fig. (13) The relation between coefficient of performance (Cop) and cold air mass ratio (μ_c) for vortex tube No.2 (θ°= 1.72), with the absolute pressure (p_I) of the inlet air as a parameter

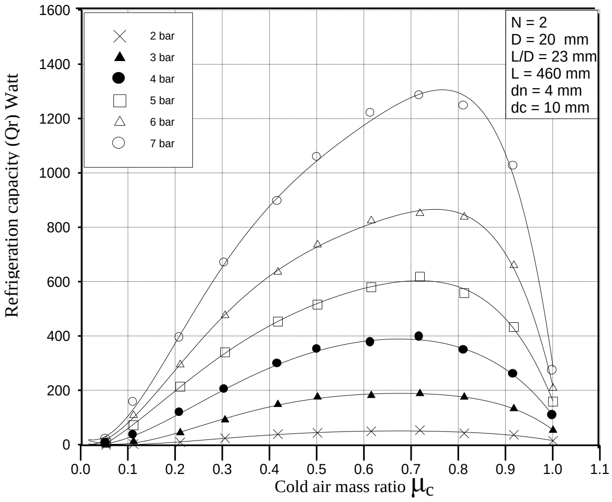

Fig. (14) The relation between Refrigeration capacity (Qr) and cold air mass ratio (μ_c) for vortex tube (θ˚= 1.72), with the absolute pressure (p_i) of the inlet air as a parameter.

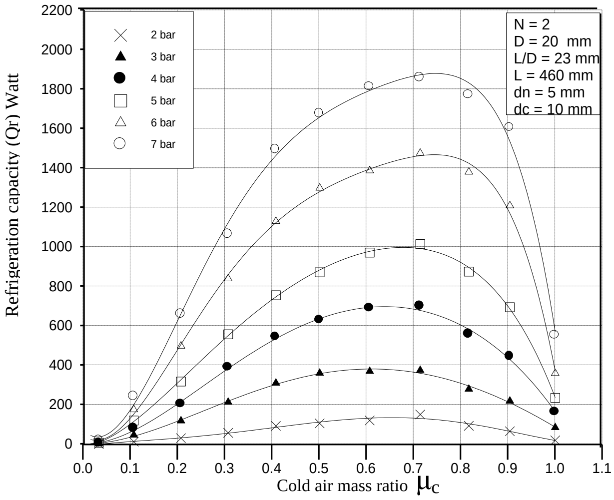

Fig. (15) The relation between Refrigeration capacity (Qr) and cold air mass ratio (μ_c) for vortex tube (θ˚= 1.72), with the absolute pressure (p_i) of the inlet air as a parameter.

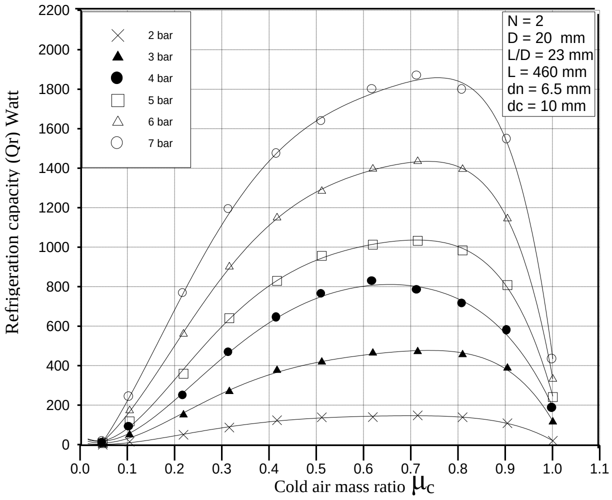

Fig. (16) The relation between Refrigeration capacity (Qr) and cold air mass ratio (μ_c) for vortex tube (θ˚= 1.72), with the absolute pressure (p_i) of the inlet air as a parameter.