MILITARY TECHNICAL COLLEGE

CAIRO - EGYPT

PERFORMANCE CHARACTERISTICS OF ENERGY SEPARATION IN DOUBLE STAGE VORTEX TUBES

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ABSTRACT

The performance characteristics of double stage vortex tubes have been investigated and compared with that for a single stage tube operating at the same inlet conditions. The effect of the diameter ratio of the second stage to the first stage tube (D2/D1) together with the operating pressure ratios for each stage were investigated.

The analytical study of the process of energy separation taking place in the double stage tube have shown that the performance of the first stage is always higher than that of the second stage tube. This conclusion was also verified experimentally. On the other hand, the experimental data have shown that the maximum temperature drop produced by the double stage vortex tube improves at higher inlet pressures (Po) and excels that of conventional single stage tubes when that pressure exceeds 3 bar. Best cooling performance for the double stage tube is achieved when the diameter ratio D2/D1 approaches unity. Also, the optimum value for the intermediate pressure (Pc1) for the double stage tube is expressed by:

$$P_{c1} = P_o^{0.45}$$

Experimental results have also indicated that; under all conditions of operation, the refrigerating effect produced by the double stage operation is always less that for the single stage operation.

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INTRODUCTION

The vortex tube is a simple refrigerating device that converts a compressed gas stream to two streams of higher and lower temperatures than that of the compressed stream. The vortex tube has been the subject of many investigations in which attempts have been made to improve its performance as a refrigerating device. The present work is an extension to those efforts which aim at clarifying and improving the design and performance of this tube.

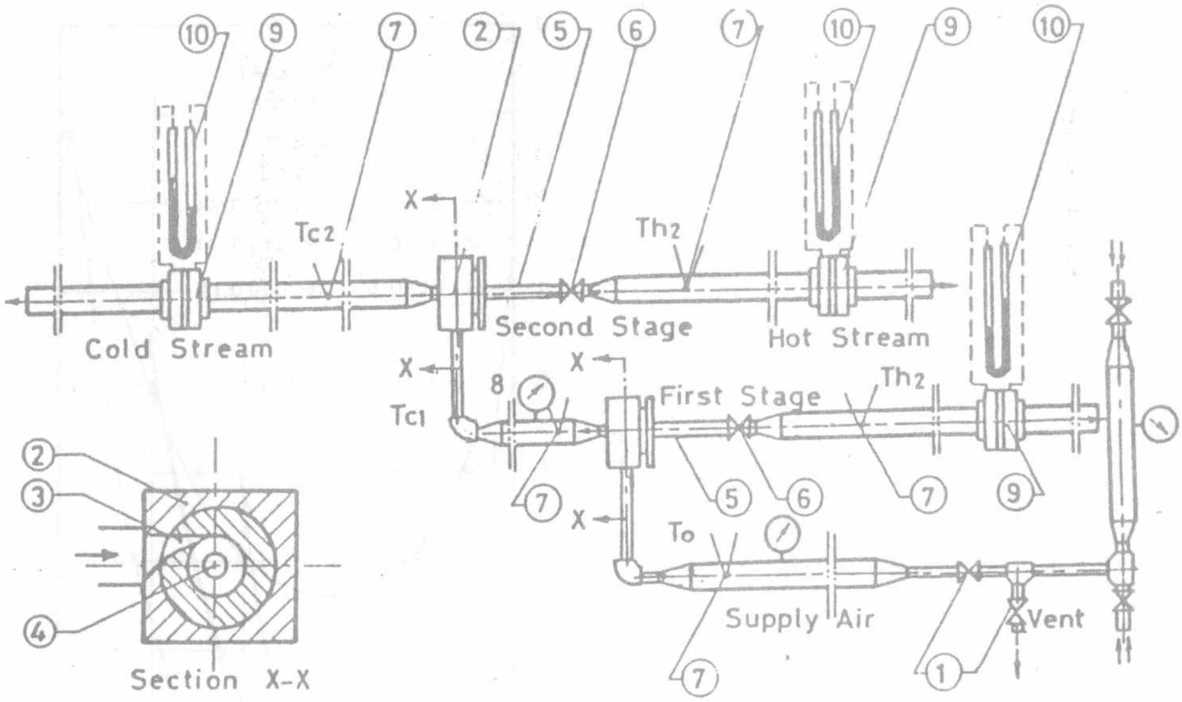
The performance of vortex tubes is found to depend on many parameters which can be divided into two categories: geometrical parameters and conditions of operation. The first category includes the effect of the tube shape, length, diameter, cold orifice diameter, type of hot valve and number, size and shape of inlet nozzles. The effect of these parameters on the tube performance as a cooling device have been investigated experimentally in many researchworks [eg. 1 - 6]. The second category includes the effect of the inlet pressure, type of working medium, its humidity and cold mass fraction. These parameters were the subject of many investigations [eg. 7 - 11].

However, there is a parameter that combines the above two categories, that is the multistage operation of the vortex tube. In this configuration; two vortex tubes are arranged in series such that the cold stream of the first stage tube is introduced to the second stage tube. This configuration was first suggested by Negm [6] who noticed that the rate of increase in the temperature drop of the cold stream is lower at higher inlet pressures. It was thus expected that the multistage operation of vortex tubes would give greater temperature drop than the single stage for the same inlet pressure. However, this conclusion have not been verified experimentally. Also, no attempt have been made so far either to study the multistage operation of vortex tubes or to provide design data for such configuration. The main objective of the present work is therefore to verify such conclusion and to provide basic data for optimal design and operation of the two stage vortex tubes.

EXPERIMENTAL APPARATUS

The main consideration in the design of the test apparatus was to be so versatile that various combinations of double stage vortex tubes could be tested at different conditions of operation. The test apparatus, shown schematically in Fig. 1 is composed of a pressure control circuit (1) which supplies a constant pressure air stream to the main block (2). This block houses the inlet nozzle-vortex chamber combination (3) and the cold end orifice (4). The hot tube (5) is provided with a throttle valve (6) at one end and is fitted to the main block at the other end via a special flange. Two identical sets of vortex tubes were prepared for testing. Each set consists of four, geometrically similar vortex tubes having inside diameters D of 11.5, 14, 16 and 20 mm. The design details of these tubes are as follows:
Tube length, L = 24D. Nozzle profile: simple with rectangular tip (width/height = 2). Tip area, $A_n = 0.12\ A_w$ (A_w : tube cross sectional area). Cold orifice diameter, $D_c = 0.5D$. Throttle valve type: globe.

The experimental program on double stage vortex tubes included testing of 12 different combinations of vortex tubes. For each combination; temperatures of the inlet air (T_o), first and second stages cold and hot



1. Pressure control circuit
2. Main block
3. Nozzle vortex-chamber
4. Cold end orifice
5. Hot tube
6. Throttle valve
7. Thermocouple probe
8. Pressure guage
9. Orifice meter
10. Manometer

Fig. 1 Experimental Apparatus

streams (Tc1, Th1 , Tc2 and Th2) were measured via a set of thermocouple probes (7). Also, inletpressure for each stage (Po and Pc1) were measured by pressure guages (8). The mass flow rates for the first stage hot sream (Mh1), second stage hot and cold streams (Mh2 and Mc2) were measured by a set of standard orifice meters (9) and manometers (10). Throughout the testing program the vortex tubes together with all piping system were perfectly insulated.

RESULTS AND DISCUSSIONS

The experimental program on double stage vortex tubes was divided into two parts. The first part was devoted to study the effect of the pressure ratio on each stage on the overall performance of the double stage tube. Whereas the influence of the diameter ratio of the second to the first stage tubes (D2/D1) was established in the second part of the work.

I. Effect Of Pressure Ratio:

The target of this set of tests was to investigate the effect of the pressure ratio on the overall performance of the double stage operation. Tests were excuted on the 16 mm diameter tube with the ratio of the second stage diameter to the first stage diameter D2/D1 = 1. Test pressures were 3,4 and 5 bar.

a. Temperature drop of cold stream:

Figure 2 shows the temperature drop attained by the double stage vortex tube (To-Tc2) versus the overall cold fraction U at inlet pressure Po = 5 bar. In this figure, the cold fractions of the first stage tube U1 is the parmeter. Qualitatively similar results are obtained for Po = 3 and 4 bar, however the values of the temperature drop produced are smaller. In Fig. 2, the curve in dashed lines represents the performance of the corresponding single stage tube (D = 16 mm) under the same condition of operation. Such curve is taken as the basis for comparison between the single and double stage operations. Note that the values of overall cold fraction U for the double stage operation is: U = U1 . U2 .

The graphs of Fig. 2 indicate that the maximum temperature drop of the double stage tube exceeds the maximum drop of a single stage tube at a specific value of the cold fraction U but not over the whole range of change of U. As can be seen; when U1 = 1 (i.e., all the air passes through the cold orifice of the first stage tube then go to the second stage); the performance of the double stage is lower than that of the single stage. This is attributed to the small temperature drop produced by the first stage tube as the process taking place inside it approaches that of the throttling process (no energy separation).

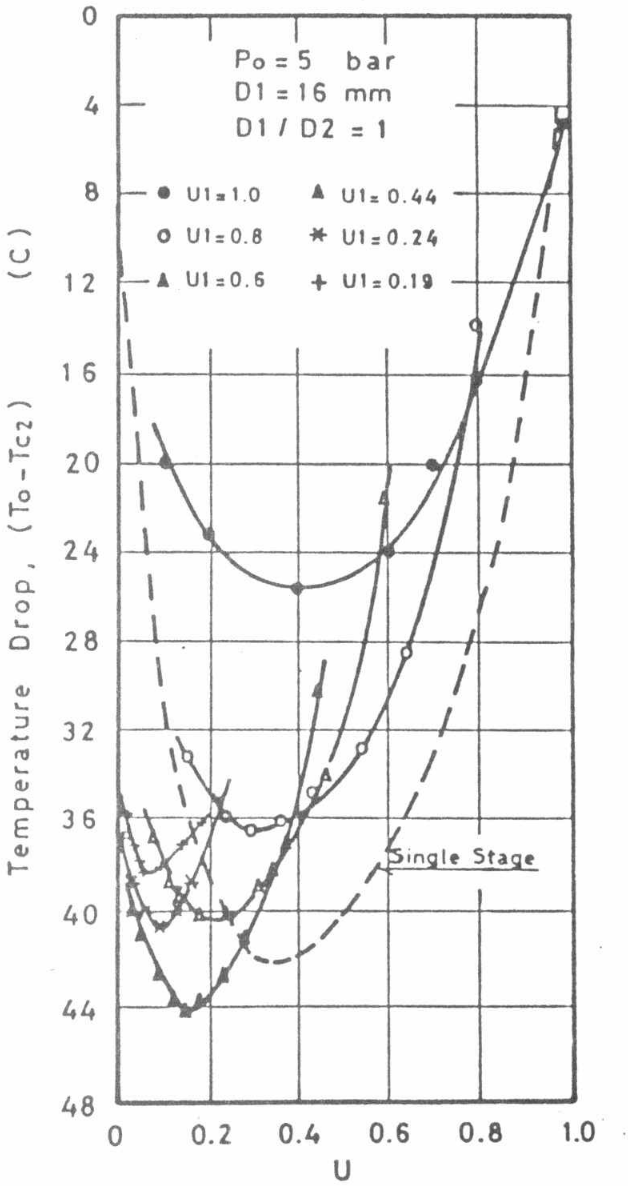


Fig. 2 Temperature Drop Vs Overall Cold Fraction

As the cold fraction of the first stage decreases (U1 < 1); the temperature drop of the first stage tube Δ Tc1 increases and consequently the temperature drop of the double stage (To - Tc2) increases. At a value of U1 = 0.44; the temperature drop (To - Tc2) approaches its maximum value which exceeds that for the single stage operation. However, when the value of U1 is decreased further (U1 < 0.4); the value of (To - Tc2) is decreased again and the performance of the double stage operation deteriorates. This deterioration is attributed to the depenedence of the pressure ratio of the second stage tube N2 on the cold fraction of the first stage U1. Figure 3 shows such relationship between N2 and U1 for different values of the overall pressure ratio N. As can be seen, at values of U1 less than 0.4; the resulting values of N2 become relatively small. Consequently, the energy separation process taking place inside the second stage tube becomes rather deficient and the overall performance deteriorates.

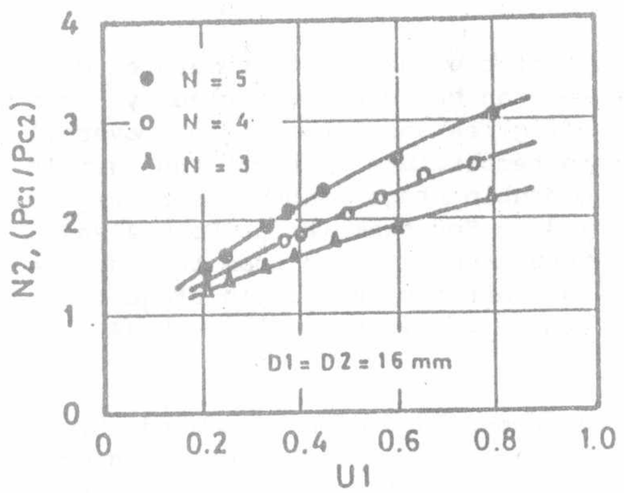


Fig. 3 First Stage Cold Fraction Vs Second Stage Pressure

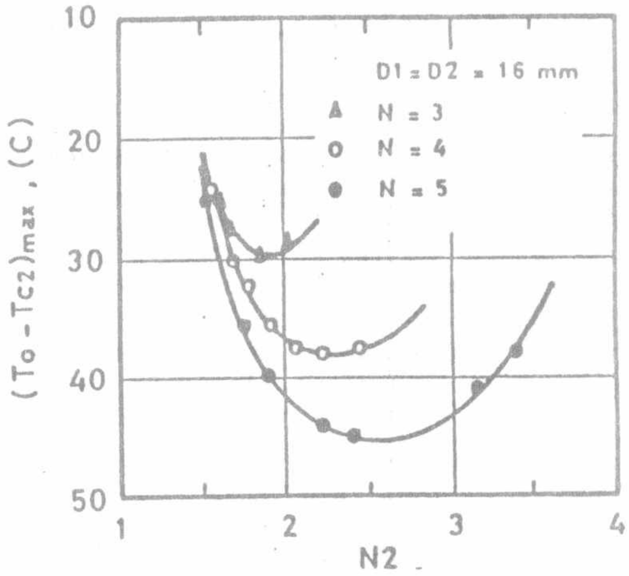
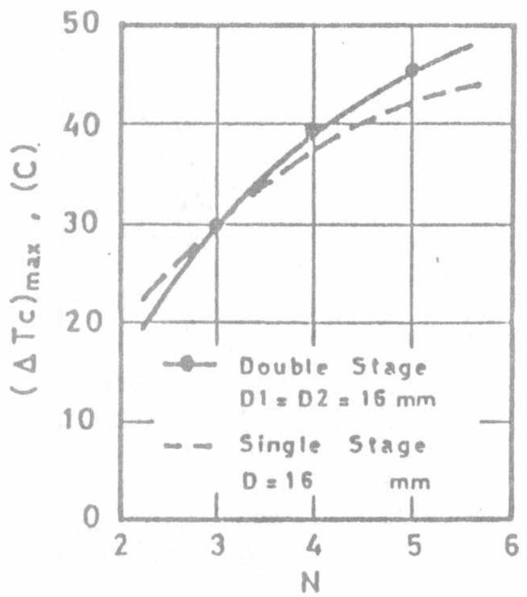
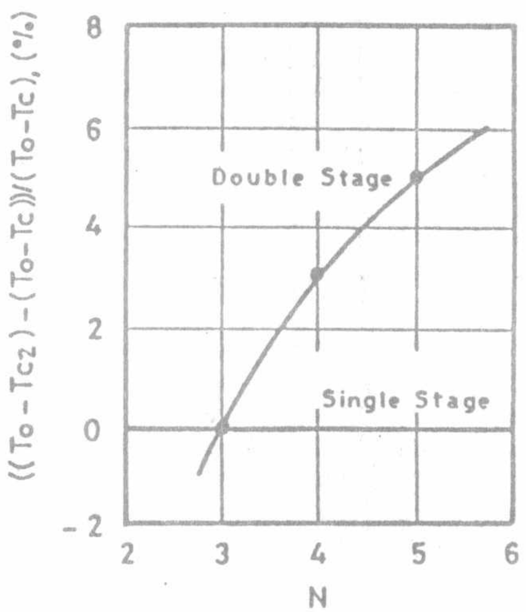


Fig. 4 Maximum Temperature Drop Vs Pressure Ratio (N2)



a. Maximum Temperature Drop Vs Pressure Ratio (N)



b. Maximum Percentage Temperature Gain From Double Stage Tube

Fig. 5 Comparison Between Maximum Temperature Drop For Single And Double Stage Vortex Tubes

Figure 4 shows the relationship between the maximum temperature drop of the double stage tube $(T_o - T_{c2})_{max}$ and the second stage pressure ratio N2, for different overall pressure ratios N (3, 4 and 5). The graphs of this figure indicate that the maximum temperature drop always takes place at :

$$N2 = N^{0.45}$$

The effect of the overall pressure ratio N on the maximum temperature drop produced by single and double stage tubes can be seen more clearly from Figs. 5-a and b. The graphs of these figures show that; for overall pressure ratios less than 3 ($N < 3$) the temperature drop of the single stage tube is better than that for the double stage tube. For higher values; $N > 3$; the situation is reversed. For example; at Po = 4 and 5 bar respectively; the maximum temperature drop attained from the double stage tube is about 3% and 5% higher than that for the single stage tube. These two values take place at values of U = 0.15 and U1 = 0.44 as previuosly shown in Fig. 2.

It can therefore be concluded that the temperature drop produced by a double stage tube becomes higher than that for single stage tube at higher pressure ratios $N > 3$. This is due to the fact that, at low pressure ratios ($N < 3$) the vortex motion inside the second stage tube becomes so weak that no gain can be obtained from it. As the pressure ratio increases ($N > 3$); the second stage tube will have a strong vortex and higher values for the temperature drop can thus be obtained from it.

b. Refrigerating effect and coefficient of performance:

The experimental results of the present work have shown that the refrigerating effect of the double stage operation is always less than that for the single stage operation. Fig. 6 shows the relationship between the refrigerating effect produced by the two stage vortex tube RE and the overall cold fraction U (U= U1.U2) where:

$$RE = U1 .\ U2 .\ (T_o - T_{c2}) \qquad (1)$$

The graphs of Fig. 6 show that; at different values of cold fraction U1; the refrigerating effect RE is always less than that for the single stage tube. This is due to the fact that the refrigerating effect is proportional to the mass flow rate of the cold stream of the second stage which is represented by U1.U2 kg per kg compressed air. The value of U1 is always less than unity, hence the product U1.U2 is always less than U for the single stage tube. At the same time, the temperature drop of the cold stream produced by the double stage tube is not large enough

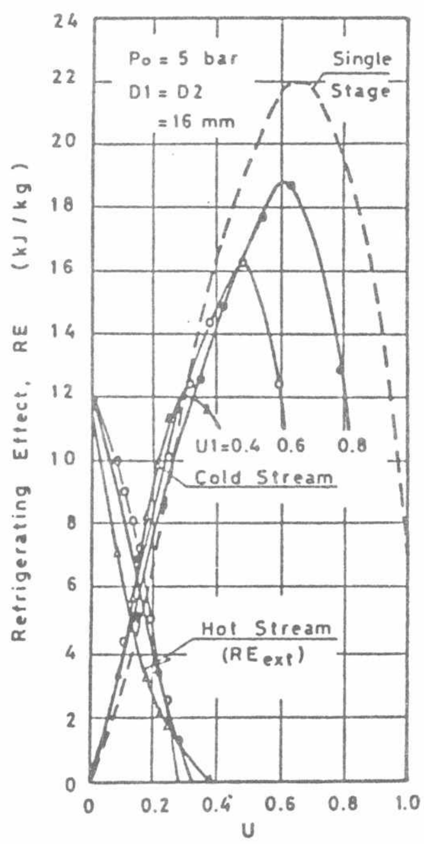


Fig. 6 Refrigerating Effect: RE Vs Overall Cold Fraction

to compensate for the smaller value of U1.U2 with the result of a reduction in the refrigerating effect. For example; at Po = 5 bar; the maximum refrigerating effect is produced at U1 = 0.8 and is equal to 18.5 kj/kg which is only 80% of that can be obtained from a single stage tube Consequently, the coefficient of performance of the double stage operation which is directly proportional to the refrigerating effect is always less than that for the single stage tube as can be seen from Fig. 7.

However, it should be pointed out that, at values of U2 less than about 0.3 , the temperature of the second stage hot stream (Th2) becomes less than that of the inlet air to the system (To). Therefore, the amount of "hot" air escaping through the throttle valve of the second stage tube represents an extra refrigerating capacity (REext) that can be obtained from such configuration:

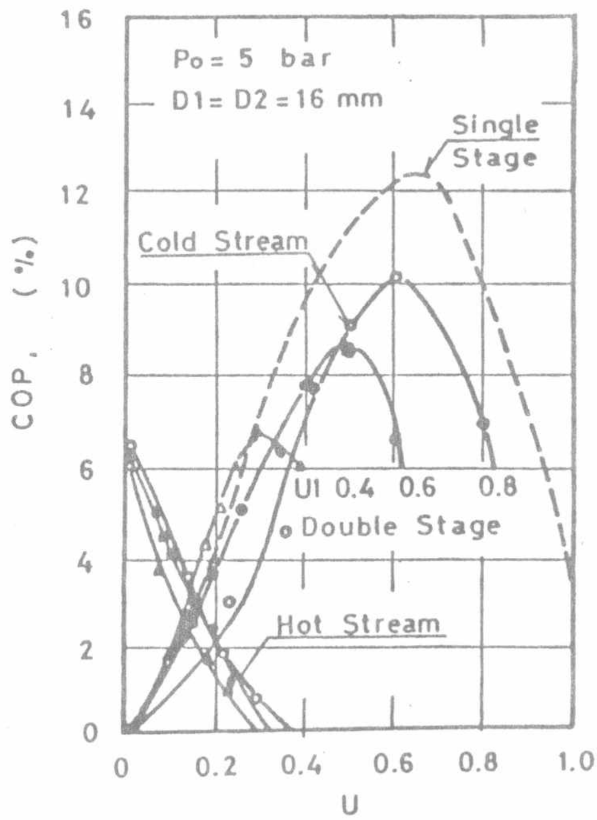


Fig. 7 Coefficient Of Performance Vs Overall Cold Fraction U

$$REext = U1 \cdot (1 - U2) \cdot (To - Th2) \qquad (2)$$

Values of both REext and the corresponding values for the COP are given at the left hand sides of Figs. 6 and 7 respectively.

II. Effect Of Diameter Ratio D2/D1:

The main objective of this set of tests is to determine the optimal diameter ratio for the second to the first stage tubes that gives the maximum temperature drop compared with that produced by the single stage operation. For these tests; different combinations of vortex tubes were prepared from the available tubes to cover a relatively wide range of diameter ratios from D2/D1 = 0.575 to 1.74. All tubes were tested at overall pressure ratio N = 4 (Po = 4 bar).

The experimental results of these tests are summarized in Fig. 8 in which the abscissa represents the diameter ratio D2/D1. The vertical axis represents the ratio between the maximum temperature drop produced by each of the 12 tube combinations (To - Tc2) to the maximum drop obtained by a single stage tube having the same diameter as the first stage tube (D1) . As can be seen, the maximum value of the ratio [(To - Tc2)max] / [(To - Tc)max] takes place at diameter ratio D2/D1 = 1 and is approximately 103%

It could also be noted from Fig. 8 that; for smaller values of D2/D1 (< 0.6) the double stage operation would produce much lower temperature drop than the single stage tube. This is attributed to the high resistance to air flow of the second stage tube resulting from its small size. Such resistance forces most of the compressed air supplied to the system to escape from the hot valve of the first stage tube.

For larger values of diameter ratios (D2/D1 > 2); the performance of the double stage tube approachs that of the single stage tube. This is attributed to the relatively large volume provided by the second stage tube which causes the pressure inside it to approach the atmospheric pressure.

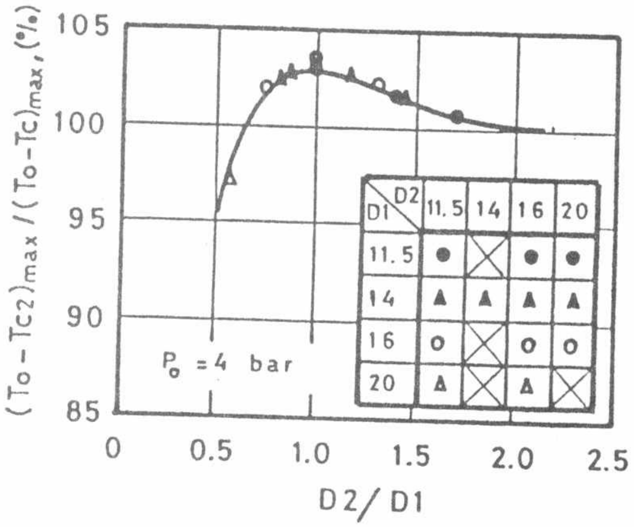
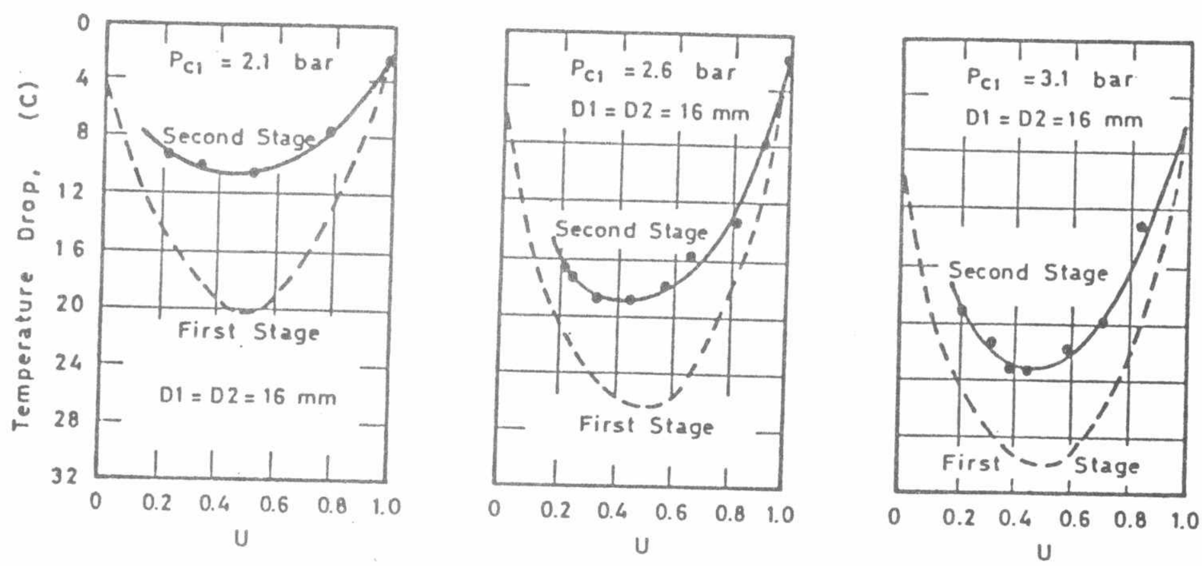


Fig. 8 Effect Of Diameter Ratio On Max. Temperature Drop

III. Temperature Drop At Second Stage Tube:

The experimental results on double stage vortex tubes have shown that the second stage temperature drop (Tc1 - Tc2) is always less than that for the first stage tube (To -Tc1) for same pressure ratio (N1=N2). Figures 9-a through c show the temperature drop of the second stage tube as compared



a. U1 = 0.4 b. U1 = 0.6 c. U1 = 0.8

Fig. 9 Comparison Between Temperature Drop Produced By First and Second Stage Vortex Tubes with that for the first stage tube for same value of N. Fig. 10 shows the ratios of the temperature drop of the second stage to that for the first stage tube plotted against the cold fraction U. It can be noted that the temperature drop ratio($\Delta Tc2 / \Delta Tc$) is always less than unity. Also this ratio decreases as the cold fraction U1 decreases. These two results can be ascribed to the difference in temperature level of the inlet air to each stage. The inlet air temperature to the first stage tube (To) is always higher than that for the second stage tube (Tc1).

Such effect of the temperature level of the inlet gas on the tube performance can be seen from the entropy analysis of the system. As has been shown previously [7], the minimum change in entropy of the cold stream produced by geometrically similar vortex tubes (Δ Sc) min can be expressed as:

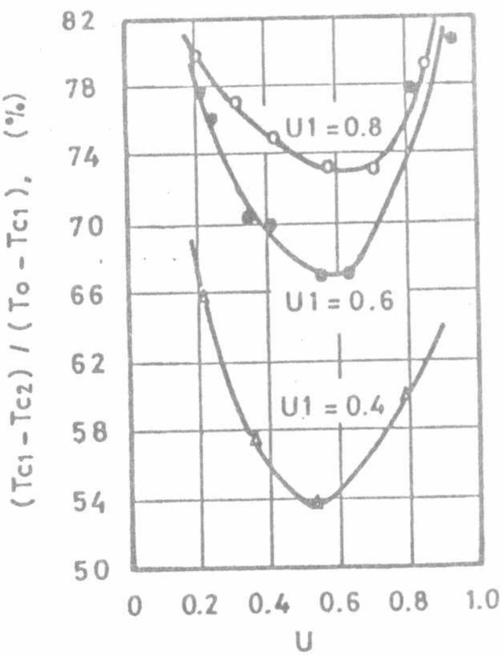


Fig. 10 Ratio Between Second To First Stage Temp. Drop

$$(\Delta Sc)min / B = a \tag{3}$$

where,
$$B = R . \ln N \tag{4}$$

and a is a constant: $a = 0.67$

Equation 4 indicates that the value of B is constant for same working medium (expressed by the gas constant R) and same pressure ratio N.

Now consider two air streams having inlet temperatures of To1 and To2, where To1 > To2; thus:

$$(\Delta Sc1)\ min / B = (\Delta Sc2)min / B = a \tag{5}$$

$$(\Delta Sc1)\ min = (\Delta Sc2)\ min \tag{6}$$

Hence;

$$Cp.\ln (Tc1\ min / To1) + R.\ln (N) = Cp.\ln (Tc2\ min / To2) + R.\ln (N) \tag{7}$$

$$Tc1\ min / To1 = Tc2\ min / To2 \tag{8}$$

$$\Delta Tc1\ max / To1 = \Delta Tc2\ max / To2 \tag{9}$$

Equation 9 indicates that the maximum temperature drop preoduced by a voretx tube (To-Tc)max is directly proportional to the temperature level of the inlet gas (To). Hence, for the case under consideration:

$$(\Delta Tc1)\ max > (\Delta Tc2)\ max$$

This conclusion is in full agreement with the experimental results of the double stage vortex tubes.

CONCLUSIONS

The performance of different combinations of double stage vortex tubes have been investigated experimentally under various conditions of operation. The following conclusions may be drawn:

1. Best performance (maximum temperature drop of the cold stream) can be achieved under the following conditions:
a- Diameter ratio of the second to the first stage tubes D2/D1 should be as close as possible to unity.
b- Pressure ratio for the second stage tube N2 is given by:

$$N2 = N^{0.45}$$

where N is the overall pressure ratio (Po/Pc2).

2. Under optimal conditions; the maximum temperature drop produced by the double stage vortex tube is higher than that by the single stage tube. This is true at higher overall pressure ratios $N > 3$. However at lower ratios $N < 3$; the situation is reversed. Test results have shown that the percentage gain in temperature drop at pressure ratios of 4 and 5 are 3 and 5% respectively. Therefore, multistaging of vortex tubes would be more advantageous when exceptionally high inlet pressures are available (i.e., $Po > 10$ bar).

3. Under the same conditions of operation; the refrigerating effect and coefficient of performance of the double stage tube is less than that for single stage tube. At pressure ratio N=5; the percentage reduction in the maximum refrigerating effect is 20%. At lower values of N; such reduction is higher.

4. The temperature drop produced by the second stage vortex tube (Tc1 - Tc2) is usually less than that by the first stage tube (To - Tc1) under same values of pressure ratios (N1 = N2) and cold fractions (U = U1.U2). This is attributed to the lower temperature level of the inlet air to the former tube (Tc1) than that for the latter tube (To).

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NOMENCLATURE

COP	Coefficient of performance	
Cp	Specific heat at constant pressure	[J/kg.K]
D	Tube diameter	[mm]
N	Overall pressure ratio = N1 . N2	
N1	Pressure ratio for first stage tube = Po / Pc1	
N2	Pressure ratio for second stage tube = Pc1 / Pc2	
P	Pressure	[bar abs.]
RE	Refrigerating effect = U . (To - Tc)	[kJ / Kg supply air]
S	Specific entropy	[J / Kg.K]
T	Temperature	[K]
R	Gas constant	[J / kg.K]
ΔT	Temperature difference	[C]
U	Overall cold fraction = U1.U2	[Kg cold stream / Kg supply air]
U1	Cold fraction of first stage tube	
U2	Cold fraction of second stage tube	

Subscripts:

c	Cold stream
h	Hot stream
max	Maximum value
o	Inlet condition
1	First stage tube
2	Second stage tube