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THE VORTEX TUBE— Internal Flow Data and A Heat Transfer Theory

George W. Scheper Jr.

Gas Turbine Engineering
General Electric Company

This article continues the discussion, begun two years ago in REFRIGERATING ENGINEERING, of the thermodynamic principles which cause the hot and cold air flows through what has become known as the Hilsch Vortex Tube. The earlier articles are:

"Analysis of the Hilsch Vortex Tube" by D. S. Webster, published in February 1950

"Ranque's Tube" by C. D. Fulton, published in May 1950

"Fluid Action in the Vortex Tube" by Roy MacGee Jr., published in October 1950

"Bibliography of the Vortex Tube" by Walter Curley and Roy MacGee Jr., published in February 1951

THE vortex, or Hilsch, or Ranque, tube as it is variously called is a remarkably simple device which produces hot and cold gas streams simultaneously from a source of compressed gas. This device has no moving parts. It consists of a straight length of tubing with a concentric orifice near one end and a nozzle entering tangentially near the outer radius adjacent to the orifice plate (see Fig. 1). By throttling on the end of the tube farthest from the orifice, a cold stream is forced through the orifice, while a hot stream issues from the opposite end of the tube. Cold air temperatures 100 deg F below nozzle temperature are readily obtainable at moderate nozzle pressures with correspondingly high hot air temperatures.

Interest in the vortex tube has been increasing rapidly in this country since the publication here of the experimental work done in Germany by Rudolf Hilsch[1]. In this basic work Hilsch determined tube proportions for optimum performance, and overall performance data over a wide range of nozzle pressures. The device was actually invented in France by G. Ranque, who obtained a French patent in 1932 and a U. S. patent in 1934. Ranque described the tube in an article in 1933[2]. USP 1,952,281

That the vortex tube does actually separate a compressed gas into higher and lower stagnation temperature streams is beyond question. How it does this is at present a subject of much study and controversy. To the writer's knowledge no data have previously been published on the internal flow processes in a vortex tube.

As the subject of a master's degree thesis, the writer performed experiments on a vortex tube to define, if only approximately, the flow and temperature patterns existing under various operating conditions. This work formed the basis for the "heat transfer theory" advanced in the thesis, completed in May 1949[3]. The present paper incorporates much of the thesis work and presents the heat transfer theory in slightly modified form. This is, to the writer's knowledge, the first vortex tube theory based on forced convection heat transfer to be published.

On this theory, heat transfer occurs radially from the vortex core outward by virtue of static temperature gradients in that direction. The heat sink is provided by the outer gas layers which are at lower static temperature due to the nozzle expansion. This heat transfer raises the stagnation temperature of the outer gas which produces the hot flow, while at the same time the stagnation temperature of the core is lowered producing a cold flow through the orifice.

Test Results

Two classes of tests were used to determine the flow patterns. The first was by visual methods; the second by instrument traversing. In all of these tests a relatively large tube diameter (1½ in. ID) was used to facilitate probing the interior of the tube. The nozzle design and tube proportions were dictated by simplicity of construction and compressed air supply limitations. As was to be

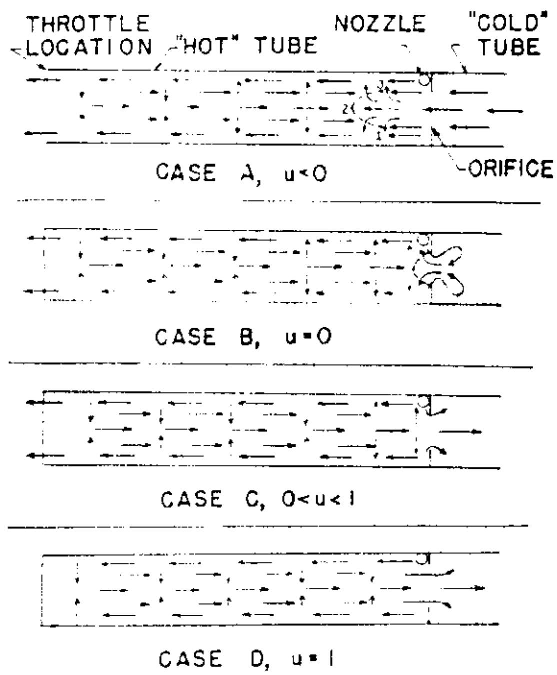

Fig. 1. Basic flow patterns for four operating conditions defined by cold fraction u.

expected, these design compromises resulted in poorer overall performance than the tubes of Hilsch[1]. However, for the purpose of this investigation, optimum performance was not considered to be of importance.

For the visual testing the tube used was glass with a brass nozzle section and clamp ring. The "hot" tube was three feet long, and the "cold" tube was eight in. long. The nozzle diameter was ¾ in., and the cold orifice diameter was ½ in. The most effective way found to visualize the flow was to use silk thread knotted loosely around a tightly stretched wire running axially through the tubes. The thread was about ⅛ in. long, pointing

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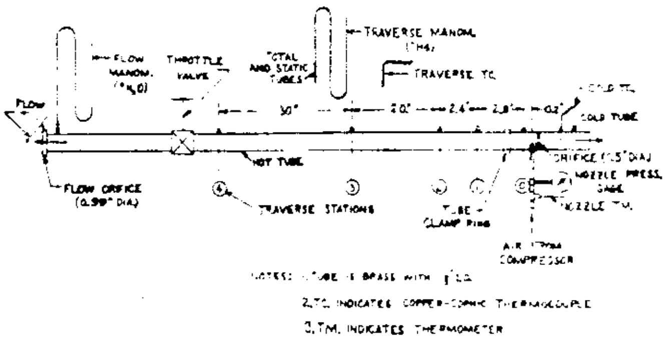
radially, and free to rotate and slide axially along the wire. In operation, the thread rotated rapidly at all axial positions as long as the center of rotation (the wire) was within about ¼ in. of the tube axis. Outside of this diameter the thread did not rotate, but assumed a tangential direction.

The most revealing feature of these tests, however, was the axial motion of the thread at various radii and degrees of hot end throttling. These results are shown in Figure 1 by arrows parallel to the tube axis. In this figure, and as used throughout this paper, the symbol μ represents the fraction of nozzle flow passing through the orifice, and its value is determined by the amount of throttling on the end of the hot tube. It is seen in Figure 1 that there exists for all values of μ a definite



NOT TO SCALE

Fig. 2. Test setup showing instrumentation.

and strong reverse flow in the core with a diameter approximately equal to the orifice diameter, and that this flow originates at a point very near the throttle end of the tube.

Chemical smoke testing with stannic tetrachloride was in general unsatisfactory due to the high degree of turbulence, even at the lowest nozzle pressures. However, the smoke did give some indication of the presence of a more r less solid rotation core flow.

Oil in the compressed air left a helical oil trace of increasing pitch toward the hot end on the tube wall. The angles of this helix indicate that near the tube wall the axial component of velocity is less than the tangential component for virtually the whole tube length.

From all the foregoing, and in particular the axial motions of the threads, a great deal can be deduced about the radial flows which occur in the tube. Figure 1 shows the four possible operating conditions of a vortex tube at constant nozzle pressure, varying only the cold fraction, μ, by throttling the hot end. This figure summarizes the internal flow configurations in a qualitative form and is basic to the theory presented later.

In Case A ($\mu < 0$) a suction is created at the orifice due to insufficient throttling. Thus, room air is drawn into the tube. This air penetrates axially to a stagnation surface of revolution such as 1-2-3, shown by the dotted curve, across which there is no axial flow. Extending from the extreme hot tube end up to the surface 1-2-3 is a core back flow caused by a sink flow near the tube end. This back flow is then returned to the outer helix flow by a source flow as surface 1-2-3 is approached. Raising the nozzle pressure moves surface 1-2-3 toward the hot end, due to the increased suction effect. Throttling the hot end moves surface 1-2-3 toward the orifice.

In Case B ($\mu = 0$) the throttling is such as to move the surface 1-2-3 partially through the orifice so that no net flow exists through the orifice. There probably exists a small recirculation through the orifice with inflow near the center and on equal outflow leaking around the edge of the orifice. The back flow and recirculation within the tube remains essentially as in Case A.

In Case C ($0 < \mu < 1$), the only operating condition at which an outflow of cold air is produced, the throttling is increased to push a net flow through the orifice. The back flow maintains itself, but as μ is raised the source flow near the nozzle decreases, probably due to the increase in the static pressure gradient near the nozzle, and the higher axial velocity ... e core flow. It is probable that a leakage exists, as ... Case B, around the edge of the orifice due to friction on the orifice plate surface lowering the velocity and hence the centrifugal force on this air, which allows the pressure gradient to force it inward.

In Case D ($\mu = 1$) the hot tube is closed off, forcing the total flow through the orifice. As in all the previous cases the core back flow is still present, but there is probably no longer a source flow near the nozzle—i.e., all radial flow is inward. The flow pattern is in this case very similar to that obtained for a cyclone separator[4] which is essentially the equivalent of a vortex tube with no hot end flow.

The flow in a vortex tube may be summarized as consisting of an outer helix flowing away from the nozzle, and an inner or core helix flowing back toward the nozzle end. The radial flow distribution depends on the amount of throttling on the hot end.

In the instrument traversing tests the glass tubes used in the visual tests were replaced by brass pipe of the same diameter, having traverse holes located along the pipe. Figure 2 is a sketch of this test setup and shows the location of the five traverse stations (numbered 0 to 4). All other instrumentation is also shown. Traverse readings were made at each of seven radial positions (numbered 0 to 6) spaced ⅛ in. apart, with position 0 at the axis of the tube. Three traverse instruments were used: thermocouple "tube," static pressure tube, total* pressure tube. Each of these tubes was made with a short length of 3/32-in. OD stainless steel tubing. For the thermocouple tube, an exposed fused junction of copper-copnic wire extended about ¼ in. from the end of the tube. The impact tube was made with a No. 80 drill hole (0.0135 in. diam) normal to the tube axis and located 3/16 in. from the plugged end of the tube. The tube was provided with a direction arm to indicate the approximate direction of the impact hole. The static tube was made with a No. 80 drill hole located in the end surface of the tube, concentric with the tube axis. Before drilling the hole, the end surface was machined flat and perpendicular to the axis after plugging the end of the tube with brazing material. The principle of this static tube is that the pressure hole is located in a flat surface which lies parallel to the rotational flow in the vortex. This can of course be valid only if the radial velocity component is negligibly small, which the writer believes to be true in the vortex tube.

It is recognized that this instrumentation, particularly the static pressure tube, is rather crude and places certain limitations on the accuracy of the data herein presented. It is felt, nevertheless, that the results obtained are sufficiently accurate to show the basic internal flow pattern. The instrumentation used was chosen for its constructional simplicity. The design and construction of a more adequate instrumentation for this complex flow field was beyond the scope of the work undertaken. In this paper a representative part only of the original data obtained is presented. Reference 3 contains the complete data obtained.

Static and total pressure vs radial position was measured at four traverse stations along the tube. The total pressures were maximum values obtained by rotating the impact tube. Angles at which this reading maximized (stream angles) were not measured but it was observed that the stream angles were essentially tangential, indicating relatively small axial velocities. This result is in agreement with the helical oil traces observed in the visual tests discussed previously.

Velocities were calculated on a compressible flow basis with the aid of Reference 5. The thermocouple temperatures were corrected for impact recovery by the use of a recovery factor of 0.65[6].

At axial stations 1 and 2 the flow field is essentially one of solid rotation up to about ⅛ in. of the tube wall (radial position 5). Between this radius and the wall the velocity distribution approaches that of a free vortex in which tangential velocity is inversely proportional to radius. The velocities obtained are essentially equal to the tangential components. At the axis of the tube, however, the axial

* Total, stagnation, and impact are synonymous terms as applied to pressure or temperature.

velocity component dominates and hence the measured velocity does not go to zero. There may also be some lack of rotational symmetry.

Figure 3 is a plot of total and static temperature vs radius for a nozzle pressure of 15 psig and cold fraction (u) of 0.26. At this condition the cold air temperature was 43 F with a nozzle air temperature of 64 F. The significance of these curves with respect to the heat transfer theory of operation lies in the fact that the cold air stream which emerges from the core of the tube has a higher static temperature than the surrounding outer helix.

Figure 4 is a plot of total temperature vs axial distance from the nozzle for radial positions 1 and 5 which were chosen to be representative of the core flow and outer helix flow respectively. The arrows on the curves indicate the axial flow direction at these two radii. In this case cold fraction (u) was zero, the nozzle conditions being identical to those of Figure 3. It is observed from Figure 4 that cold temperatures are produced even though no flow goes through the orifice. In accordance with the requirement of an overall heat balance, assuming no heat transfer between the air and the metal of the tube wall, there is no

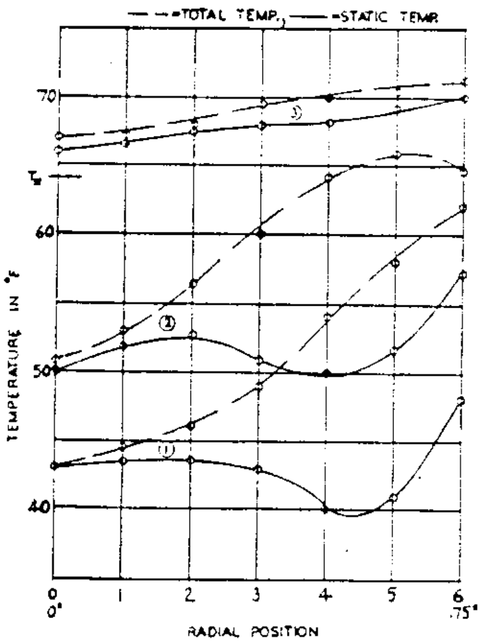


Fig. 3. Static and total temperature vs radius at three traverse stations.

net increase in temperature of the outer helix flow which issues from the throttle valve. Figures 3 and 4 lead immediately to the heat transfer theory of the vortex tube.

Before leaving this section on test results it should be pointed out that the remainder of the data for pressure, temperature, and velocity obtained in the original research at some other operating conditions is very similar to the curves herein, the main difference being a general shift of the magnitudes at a given traverse station.

Theory of Operation

It has been shown that the flow pattern in a vortex tube consists basically of two counter current axial flows, with an approximately solid rotation. The inner core axial flow emerges from the orifice with a reduced total or stagnation temperature. The axial flow of the outer helix emerges from the opposite end of the tube at the throttle valve with an increased total temperature. The core is at a higher static temperature than the outer helix. These conditions form the basis for the heat transfer theory of the vortex tube, which follows.

Referring to Figures 3 and 4, the increase of the total temperature of the outer flow is attributable only to heat transfer from the inner core, since no mechanical work is involved. The core, being at a higher static temperature than the outer helix, transfers heat by forced convection

NOMENCLATURE

A = effective surface area, sq ft
C = specific heat at constant pressure, Btu per lb, deg F
N = cooling effectiveness, dimensionless
P = nozzle inlet pressure, atmospheres absolute
Q = heat flow rate, Btu per hr
T = total (stagnation) temperature, F
t = static temperature, F
u = cold flow fraction, dimensionless
U = overall heat transfer coefficient, Btu per hr, sq ft, deg F
w = weight flow rate, lb per hr
θ = heat transfer mean static temperature difference, deg F
Δt = isentropic nozzle temperature drop, deg F

Subscripts on T or t

N = at nozzle discharge
C = cold flow at orifice discharge
H = hot flow at throttle valve discharge

to the outer helix, and as a consequence reduces its own total temperature. The analogy to a concentric pipe counterflow heat exchanger is evident. If we imagine a zero thickness tube wall, having a diameter approximately equal to that of the cold orifice, which is rotating at every axial position at the same angular velocity as the vortex, the heat transfer across this imaginary tube surface is proportional to the bulk static temperature difference

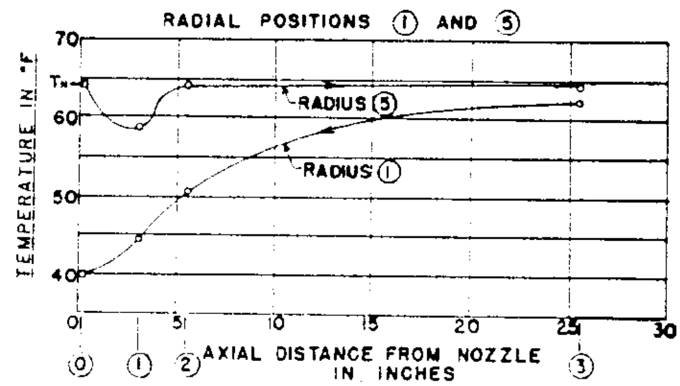


Fig. 4. Total temperature vs axial distance from nozzle at two radial position, $u = 0$.

and to the overall film coefficient of heat transfer occasioned by the relative axial velocity of the two oppositely moving streams. With the assumption of such a rotating tube, the tangential velocities do not contribute directly to the film coefficient of heat transfer, nor does the flow field become altered, as would be the case with a stationary tube due to tangential wall friction.

The break in the curve of Figure 4 near traverse station 1 is believed to be due to radial mixing effects, since it has been shown that radial outflow occurs in this region (see Fig. 1, Cases B and C).

In the following analytical treatment radial mixing is neglected for simplification, and also because its effect is believed to be of a secondary nature. Figure 5 shows the simplified flow pattern assumed and also gives qualitative hypothetical temperature vs length curves to illustrate the temperature nomenclature used, and to clarify the theoretical analysis.

To satisfy the necessary overall heat balance in the absence of mechanical work or heat transfer to the environment

$$u(T_N - T_C) = (1 - u)(T_H - T_N) \qquad (1)$$

Or,

$$u(T_H - T_C) = T_H - T_N \qquad (2)$$

As in the case of conventional heat exchangers, a cooling effectiveness can be defined, which is the ratio of the heat transferred to the maximum possible heat transferable with the available temperatures, as follows:

$$N = (T_H - T_C)/(T_H - t_N) \qquad (3)$$

In Equation 3, t_N is the lowest available static temperature in the vicinity of the nozzle. If an ideal isentropic nozzle expansion is assumed, then

$$t_N = T_N - \Delta t \qquad (4)$$

Where

$$\Delta t = T_N / C\,(f(P))$$

Combining Equations 2, 3, and 4,

$$T_H - T_N = \frac{u\,\Delta t}{(1/N) - u} \qquad (5)$$

Then using Equation 1 with Equation 5,

$$T_N - T_C = \frac{(1 - u)\,\Delta t}{(1/N) - u} \qquad (6)$$

Equations 5 and 6 give the overall performance of a vortex tube as a function of the operating variables u and Δt and the cooling effectiveness, N. Knowing the performance data for a given vortex tube, N can be calculated from Equation 5 or 6. In the hope that N could serve as a correlating factor in vortex tube performance, the writer has calculated it for many of the operating points in the rather thorough data of R. Hilsch[1]. The results are given in Figures 6 and 7, where N is plotted vs u. Figure 6 gives four curves at constant nozzle pressures, P, varying from 2.5 to 11 atmospheres absolute for the No. 2 Hilsch tube which has the following principal dimensions:

Nozzle diameter	2.3 mm
Tube diameter	9.6 mm
Orifice diameter	4.2 mm

Figure 7 gives four curves all at $P = 11$ atm abs, but with four different orifice diameters, and one curve at

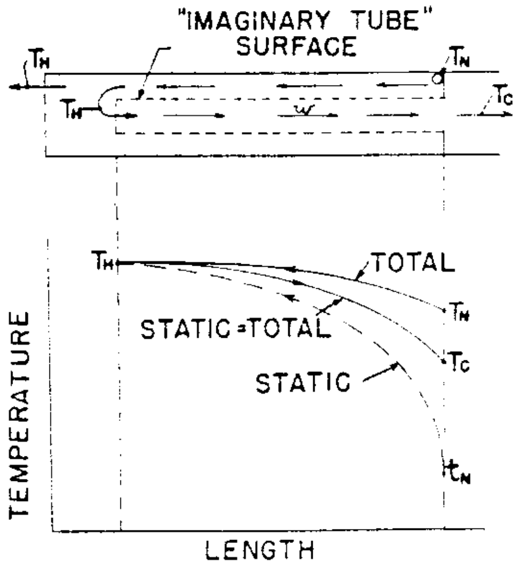


Fig. 5. Flow pattern and temperature nomenclature for theoretical derivation.

$P = 4$ atm abs with a 2.2 mm diameter orifice, all for the No. 1 Hilsch tube having the following principal dimensions:

Nozzle diameter	1.1 mm
Tube diameter	4.6 mm
Orifice diameter	varied from 1.4 to 2.6 mm

These data were calculated from points read off the published curves[1], which were reproduced to a very small size, and hence some accuracy has been lost in Figures 6 and 7. In Figure 6 the variations of N with u are well represented with straight lines. In Figure 7 the effect of orifice size on the variation of N is apparent. Comparison of the curves for the 1.4 and 1.8 mm orifices makes it appear that at some intermediate orifice size, say 1.6 mm, N would be virtually constant at a value of about 0.43.

Figure [illegible] Hilsch's data (solid curves) for his No. 2 tub[illegible] as used in Figure 6), giving temperature drop [illegible] for $P = 4$ and 11.

It now becomes necessary to point out a basic discrepancy which shows up in all the data of Hilsch when compared to the writer's experimental results. As Figure 8 indicates, the Hilsch data show the temperature drop as zero at $u = 0$. The writer's data indicate that the temperature drop is not zero at $u = 0$. Figure 4, for $u = 0$, shows that in the core just inside of the cold orifice, the air is in fact colder than it is for a value of $u = 0.26$, the latter value of u being according to Hilsch's data nearer the maximum temperature drop value. It is believed that this discrepancy lies in the location of and

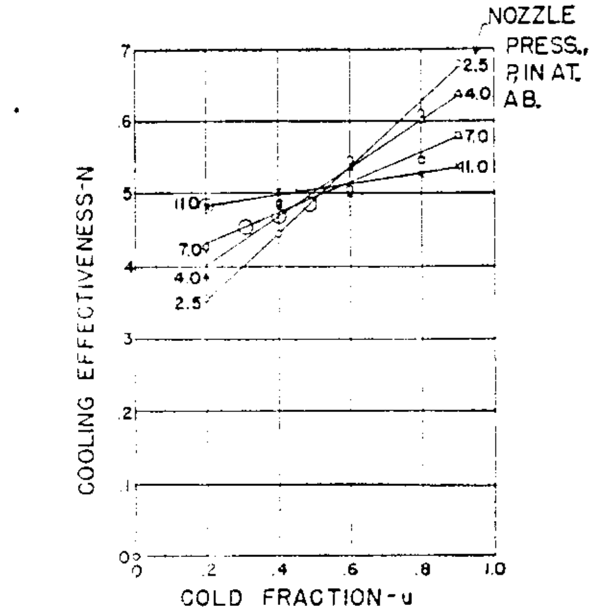


Fig. 6. Cooling effectiveness vs cold fraction for No. 2 Hilsch tube at four nozzle pressures.

the method of the cold temperature measurement. Hilsch chose to measure the temperature of the metal wall of the tube on the discharge side of the orifice as an indication of cold air temperature. In Hilsch's words, "These measurements yield temperature differences which are certainly not too favorable, but will probably be too small." This must certainly be so, and particularly so as the cold flow approaches zero, at which point the metal temperature must be very near room air temperature. However, even if a temperature measurement is taken in or near the center of the cold tube on the discharge side of the orifice (as was done in the writer's tests), three side effects can increase the reading above that which exists in the center of the orifice, namely:

1) Heat pickup from the "cold" tube
2) Orifice edge leakage (see Fig. 1, Cases B and C)
3) Recirculation in the "cold" tube which can mix room air with the cold air (see Fig. 1, Case B)

Referring again to Figures 6 and 7, it can be seen that since Hilsch's data show zero temperature drop at zero cold flow, there is a point for each curve at $N = 0$ and $u = 0$, but in no case does this point fall reasonably on any of the curves. The writer believes that this apparent discontinuity in the cooling effectiveness, N, between $u = 0.2$ and $u = 0$ substantiates the aforementioned contention that a temperature drop exists at $u = 0$, even though it is reduced by the three side effects mentioned.

The larger circles shown on the curves of Figures 6 and 7 represent the points at which the temperature drop is a maximum according to Hilsch's data. This occurs in general between $u = 0.2$ and $u = 0.5$, depending on nozzle pressure and orifice diameter. It can be noted that the value of N at maximum temperature drop varies only from 0.45 to 0.48 for Figure 6, and from 0.41 to 0.46 for Figure 7. The higher average value of N for Figure 6 is attributable to a larger size tube (about twice the diameter) than that of Figure 7.

Now by assuming a constant value of N of 0.46 for No. 2 Hilsch tube (Figs. 6 and 8), Equation 6 becomes

$$T_N - T_C = \frac{(1-u)\ \Delta t}{2.17 - u} \tag{7}$$

Equation 7 represents in simplified form the performance of a vortex tube of the size and design of the Hilsch No. 2 tube. This equation has been plotted for $P = 4$ and 11 in Figure 8 (dashed curves). The difference between the solid and dashed curves lies in the assumption of constant cooling effectiveness, N, in Equation 7.

The next step is to look into a theoretical derivation of N, and to see if the values of N obtained from the test

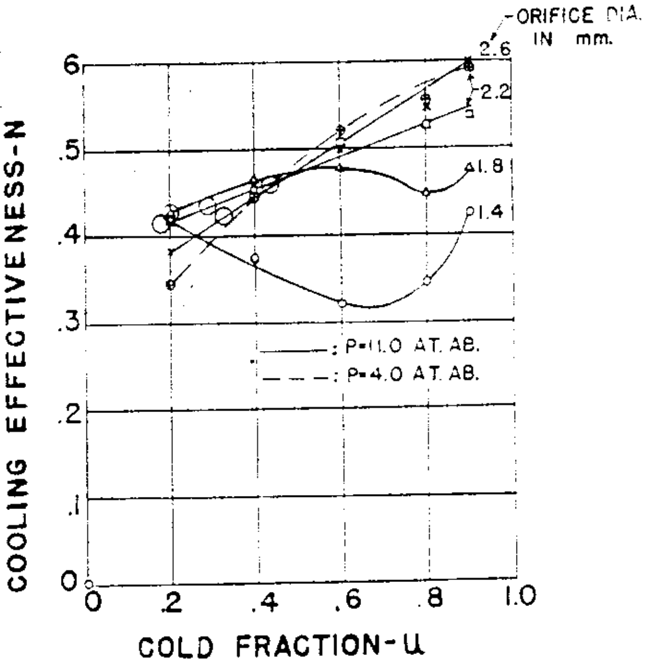


Fig. 7. Cooling effectiveness vs cold fraction for No. 1 Hilsch tube for four orifice diameters.

data could be reasonably attributed solely to a heat transfer process. In the following, an approximate derivation of N as a function of heat transfer coefficient, effective surface area, and flow is given.

Referring again to the diagram of Figure 5, the rate of heat transfer from the core flow within the imaginary tube surface is

$$Q = wC\ (T_H - T_C) \tag{8}$$

And also,

$$Q = AU\theta \tag{9}$$

Where A is the effective surface area of the imaginary tube, θ is the mean static temperature difference, and U is the overall heat transfer coefficient. Combining Equations 8 and 9 with Equation 3 gives

$$N = \frac{AU}{wC}\left(\frac{\theta}{T_H - t_N}\right) \tag{10}$$

It is notable that due to the unusual nature of the outer helix flow in which static temperature increases more than total temperature due to the large negative velocity gradient, the conventional log-mean temperature difference is not applicable here for Θ. However, by making the simplifying assumption that the temperature difference varies linearly with A, a very simple expression for N can be obtained which should be adequate since the effective area, A, is not known close enough to justify refinement in θ. Since the core flow is at a relatively low velocity, it is also assumed that the total and static temperatures are equal here. Then, referring to Figure 5,

$$\Theta = (T_C - t_N)/2 \tag{11}$$

Since $T_H - t_N = T_C - t_N + T_H - T_C$,

$T_H - t_N = T_C - t_N + (AU\theta)/(wC)$, using Equations 8 and 9.

And therefore, using Equation 11

$$T_H - t_N = 2\theta + \frac{AU}{wC}\theta \tag{12}$$

Substituting Equation 12 in Equation 10 gives

$$N = \left(\frac{AU}{wC}\right) \Big/ \left(2 + \frac{AU}{wC}\right) \tag{13}$$

Equation 13 is in the desired form. It is interesting to note the similarity between this equation and that for a conventional counterflow heat exchanger with equal heat capacity (wC) on both sides, for which

$$N = \left(\frac{AU}{wC}\right) \Big/ \left(1 + \frac{AU}{wC}\right) \tag{14}$$

From Equation 13 it can be seen that for N to be 0.50, the approximate value calculated for the Hilsch tubes, $AU/wC = 2$

If approximately representative values of A and w are used, the value of U required for this theory to match the actual performance is easily calculated. Taking for example, the No. 1 Hilsch tube (Fig. 7 data):

$A = 0.0113$ sq ft, based on an assumed effective length of 65 percent of the hot tube length and a diameter equal to the cold orifice (2.2 mm)

$C = 0.24$ Btu per lb, deg F

$w = 6.75$ lb per hr, which is the total nozzle flow at $P = 4.0$ atm abs.

Then

$$U = \frac{2 \times 6.75 \times 0.24}{0.0113} = 286 \text{ Btu per hr, sq ft, deg F}$$

As defined for the foregoing analysis, w is the flow inside the imaginary tube. Due to the radial outflow near the orifice which was shown to exist under most operating con-

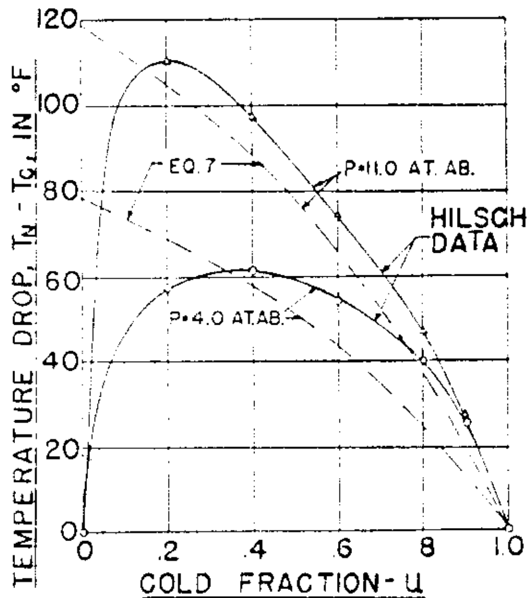


Fig. 8. Temperature drop vs cold fraction for No. 2 Hilsch tube at two nozzle pressures, by test and by theoretical equation.

ditions (see Fig. 1), the flow from the orifice is less than the value of w inside the tube. Thus even at $u = 0$ (no cold flow) w may be fairly large, but probably no larger than the nozzle flow. It is for this reason that the total nozzle flow is used in the foregoing evaluation of U. If the actual core flow, w, is less than the nozzle flow, then U required is proportionally lower. Also it can be seen that as P is raised or lowered, the U required will increase or decrease proportional to the flow, but this is compensated for by the available heat transfer coefficient varying almost proportional to the mass flow rate.

The mass flow rate in the preceding numerical example is about 46 lb per sec, sq ft in the core, which corresponds

(Continued on page 1018)

The Vortex Tube

(Continued from page 989)

in a tube of 2.2-mm diameter to a film coefficient of about 140 Btu per hr, sq ft by a conventional heat transfer equation for flow in a straight tube[7]. This is one half of the required overall coefficient. The following two factors may be sufficient to raise the above heat transfer coefficient up to the value of U required by the heat transfer theory.

1) A large degree of turbulence due largely to the helical nature of the flows, which would raise the heat transfer coefficients. It is known, for instance, that for flow inside helically coiled pipe, heat transfer coefficients may be several times higher than in straight pipe for the same Reynolds number[7].

2) The absence of a solid wall at the interface between the core and the outer flow would seem to make it logical for the film coefficient to be based on the total relative axial velocity of the inner and outer streams. Since these flow countercurrently, their velocities are additive. Due to the absence of a solid tube wall there are not two film resistances, but only one resistance evaluated from the relative axial velocity. The effect of using a relative velocity in the preceding numerical example would be to raise the calculated film coefficient by about 30 percent, from 140 to 182.

Conclusions

The heat transfer theory of vortex tube action as presented in this paper is not complete in itself and must rely on empirical results in the performance equation (Eq. 6). The cooling effectiveness, N, has been shown to be a useful correlating factor. Although the required overall heat transfer coefficient is greater than can be accounted for by conventional calculations, the two are sufficiently within range as to be encouraging. The theory does not preclude the possibility that other mechanisms may be simultaneously acting to produce cooling of the air. Other theories have been published recently[8,9] which are of an entirely different nature, but which do not appear to have been based on prior tests concerning the nature of the internal flow processes. Discussion of these theories is beyond the scope of this paper.

The writer suggests a test setup in which, if any cold air were produced, the heat transfer theory would be greatly enhanced, as follows:

Run a vortex tube in which the "imaginary tube surface" of the preceding theory is replaced with a thin metallic tube which is rotated mechanically at an angular velocity comparable to that of the air at the tube radius.

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