RICE UNIVERSITY

FRICIONAL LOSSES IN CONICAL DIFFUSERS
HAVING WIDE ANGLES

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ABSTRACT

Systematic experimental results of pressure recovery efficiency and loss coefficient of conical diffusers are presented. The Reynolds number of water flow may be up to $3.5 \times 10^5$. Area ratios of the downstream pipes to the upstream pipe were 2.25:1, 4:1, 9:1, 16:1, and 25:1. The cone angles of the diffusers were 10°, 30°, 45°, 60°, 90°, 120° and 180°. Fully developed turbulent flow entered the diffusers.
ACKNOWLEDGEMENT

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SYMBOLS

$A_1$  upstream pipe cross sectional area
$A_2$  downstream pipe cross sectional area
$D$    inside diameter of pipe
$d_1$  inside diameter of upstream pipe
$d_2$  inside diameter of downstream pipe
$f$    friction factor for pipe flow
$g$    acceleration of gravity
$\Delta h$  loss of head of diffuser
$\Delta h'$ loss of head of $180^\circ$ diffuser
$l$    length of pipe
$l_1$  length of upstream pipe
$l_2$  length of downstream pipe
$l_3$  distance from diffuser outlet to downstream pressure measuring orifices
$l_4$  distance from diffuser inlet to upstream pressure measuring orifices
$N_R$  Reynolds number
$\Delta p$ pressure difference
$p'$   static pressure at the inlet section
$p_2$  static pressure at the outlet section
$V_1$  mean velocity of fluid in the inlet pipe
$V_2$  mean velocity of fluid in the outlet pipe
$\eta$  diffuser efficiency
$\theta$ cone angle of diffuser
$\mu$  loss coefficient
\( \nu \) kinetic viscosity
\( \rho \) density of fluid
\( \varphi \) coefficient of diffuser loss
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I. INTRODUCTION

The function of a diffuser is to transfer dynamic pressure of a flowing fluid into static pressure by decelerating the flow in an expanding passage. Flow diffusion is of considerable practical importance in turbines, pumps, fans, compressors and other rotodynamic machines, as well as in exit cones of wind tunnels, air intakes on aircraft, and some other duct systems.

The efficiency of this conversion process, which is called pressure recovery efficiency, or diffuser efficiency, is of immediate interest since it affects machine performance. Let us consider a typical diffuser section; $p_i$ and $V_i$ are the pressure and the mean velocity of the fluid at the initial section, and $p_2$ and $V_2$ are those at the final section of the diffuser, respectively. The ideal pressure at the section 2 would be $p_{2i}$ if no losses occurred in the diffuser. The cross-sectional areas at sections 1 and 2 are $A_1$ and $A_2$, respectively. (See Figure 1)

The diffuser efficiency may be defined as the ratio of actual pressure recovered to ideal pressure that could be recovered by a frictionless diffuser, i.e.,

$$\eta = \frac{p_2 - p_i}{p_{2i} - p_i},$$

where $\eta$ denotes the diffuser efficiency. Bernoulli's equation for ideal flow through a diffuser is

$$\frac{p_{2i}}{\rho g} = \frac{p_i}{\rho g} + \frac{V_i^2}{2 g} - \frac{V_2^2}{2 g}$$

(2)

where $\rho$ is the mass density of fluid, which can be assumed as constant for incompressible flow. For the actual case,
\[
\frac{p_2}{\rho g} = \frac{p_1}{\rho g} + \frac{V_f^2}{2 g} - \frac{V_s^2}{2 g} - \Delta h \tag{3}
\]

where \( \Delta h \) is the loss of head due to friction and turbulence.

According to the law of continuity

\[ A_1 V_1 = A_2 V_2 \tag{4} \]

or

\[ V_2 = \frac{A_1}{A_2} V_1 \tag{5} \]

Therefore,

\[ \frac{A_2 V_2 - p_1}{\rho g} = \frac{V_f^2 - V_s^2}{2 g} = \frac{V_f^2}{2 g} \left[ 1 - \left( \frac{A_1}{A_2} \right)^2 \right] \tag{6} \]

and

\[ \frac{p_2 - p_1}{\rho g} = \frac{V_f^2}{2 g} \left[ 1 - \left( \frac{A_1}{A_2} \right)^2 \right] - \Delta h \tag{7} \]

Substituting in (1), we finally get

\[ \eta = 1 - \frac{\Delta h}{\frac{V_f^2}{2 g} \left[ 1 - \left( \frac{A_1}{A_2} \right)^2 \right]} \tag{8} \]

Let us define a coefficient of diffuser loss \( \phi \) by the relation

\[ \phi = 1 - \eta \tag{9} \]

Then

\[ \phi = \frac{\Delta h}{\frac{V_f^2}{2 g} \left[ 1 - \left( \frac{A_1}{A_2} \right)^2 \right]} \tag{10} \]
This coefficient is more convenient to employ in some situations than is the diffuser efficiency.

For the sake of comparison the results with some other earlier investigators, another factor $\mu$ will be defined. This factor $\mu$ is the ratio of the head loss $\Delta h$ in a diffuser to the head loss $\Delta h'$ in an abrupt enlargement, or diffuser with a total included cone angle of 180° (see Figure 4). If there were no head loss for an abrupt enlargement, we would have

$$\frac{p_{2i}}{\rho g} = \frac{p_i}{\rho g} + \frac{V_i^2}{2g} - \frac{V_e^2}{2g}$$  \hspace{1cm} (11)

The momentum equation applied between sections 1 and 2 is

$$\rho_i A_i + \rho'(A_e - A_i) - \rho_e A_e = \rho (A_e V_e^2 - A_i V_i^2)$$  \hspace{1cm} (12)

where $\rho'$ is the average pressure on the annular ring.

The law of continuity is:

$$A_i V_i = A_e V_e$$  \hspace{1cm} (4)

The loss of head of 180° diffuser is:

$$\Delta h' = \frac{p_{2i} - p_e}{\rho g}$$  \hspace{1cm} (13)

Substituting $p_{2i}$ from (11), $p_e$ from (12) to (13), and applying relationship of (4), we finally get
\[ \Delta h' = \frac{(V_i - V_2)^2}{2g} + (1 - \frac{A_1}{A_2})(\frac{p_1}{\rho g} - \frac{p_1'}{\rho g}) \]  

(14)

It is usually assumed that \( p' = p_1 \), in which case the loss of head due to abrupt enlargement is:

\[ \Delta h' = \frac{(V_i - V_2)^2}{2g} \]  

(15)

or

\[ \Delta h' = \frac{V_i^2}{2g} (1 - \frac{A_1}{A_2})^2 \]  

(16)

Since the factor \( \mu \) is defined as:

\[ \mu = \frac{\Delta h}{\Delta h'} \]  

(17)

we get

\[ \mu = \frac{\Delta h}{\frac{V_i^2}{2g} (1 - \frac{A_1}{A_2})^2} \]  

(18)

Various types of diffusers have been developed for different applications. Cross sections are generally circular, square, or rectangular; and wall elements are straight or curved. The conical diffuser which has a circular cross section and straight wall elements, is the most popular type. The object of the investigation described herein was to find the pressure recovery efficiency of this kind of diffuser.
II. FACTORS AFFECTING DIFFUSER EFFICIENCY

There are various factors which can affect the pressure recovery efficiency. Among the most important factors are the length of upstream pipe, \( l_1 \) and the length of downstream pipe, \( l_2 \), see Figure 1. Peters\(^{(7)}\) has shown that the flow condition in a conical diffuser and the duct behind the diffuser depend on the velocity distribution at the inlet to the diffuser, i.e., on the state of the boundary layer in the flow entering the diffuser. Various velocity distributions were obtained by using different lengths of inlet pipe in Peters' work. Also, it has been shown that the rise in pressure during the transformation of velocity head to pressure head is not complete at the final section of the cone. Further results show that the position with respect to length along the downstream pipe of maximum pressure depends on the angle of expansion and on the flow conditions at the entry to the diffuser.

The previously derived formula (8) for the diffuser efficiency is:

\[
\eta = \frac{p_2 - p_1}{\frac{1}{2} \rho V_1^2 \left[ 1 - \left( \frac{A_1}{A_2} \right)^2 \right]} \tag{19}
\]

where \( \rho \) = density of the fluid
\( V_1 \) = mean velocity at the initial section,
\( A_1 \) = upstream pipe cross sectional area,
\( A_2 \) = downstream pipe cross sectional area,
\( p_1 \) = pressure at the initial section,
and \( p_2 \) = pressure at the final section.

If one then measures the \( p_2 \) at the maximum section of the cone and
the location of maximum pressure downstream from the exit of the diffuser separately, two different efficiencies can be found. These two efficiencies are shown in Fig. 2. It is clear that the efficiency is appreciably increased when the diffuser is followed by a length of duct. The effect is greatest when the inlet length, $l_1$, is long and the cone angle is large.

No extensive tests of the effect of the Reynolds number on the efficiency appear to have been made. Peters' experiments were made at $N_R = 2 \times 10^5$. Gibson (ref. 1,2) indicated that there was no appreciable scale effect on the efficiency in the range

$$0.5 \times 10^5 \leq N_R \leq 2.5 \times 10^5$$

in which his experiments were made.

In general the efficiency decreases as the area ratio $A_2/A_1$ increases.

The shape of diffuser cross section is another important factor. Square and rectangular cross sections are less efficient than is the circular cross section of a conical diffuser. The effect of curved, rather than straight, walls has been investigated by Warner (ref. 8), Ziembinske (ref. 16), Gibson (ref. 1,2) and Ackeret (ref. 17). The general conclusion was that for small cone angles, little benefit was to be expected from using curved wall. However, when the expansion is large, the efficiency is appreciably improved.

Cone angle seems to be another important factor to affect the efficiency of conical diffuser. The flow between two straight diverging walls has been investigated in detail by Vedernikov (ref. 18). His observations show that as the angle of expansion increase, the flow becomes unsymmetrical and the separation occurs at one wall near the en-
trance to the diffuser where the direction of the wall changes suddenly. Prandtl\textsuperscript{(ref.14)} also has confirmed this fact.

It is very interesting phenomenon that the loss in a diffuser with roughened walls is less than that of a diffuser with smooth walls when the cone angle is larger than 40°. This is true because although wall roughness increases the friction loss, it can delay the flow separation. The contribution to the loss coefficient due to wall friction is quite small, but the loss due to flow separation greatly increases as the cone angle increases from 10° to 60°.
III. COMPARISON OF EARLIER INVESTIGATIONS

Gibson's experiments were made in 1910. He studied flow in conical diffusers with the following diameters of inlet and outlet pipes and corresponding ratios of outlet to inlet cross sections as indicated:

- 1.5 in. to 3.0 in. \( \frac{4}{1} \) area ratio
- 2.0 in. to 3.0 in. \( \frac{2.25}{1} \) area ratio
- 0.5 in. to 1.5 in. \( \frac{9}{1} \) area ratio
- 1.0 in. to 3.0 in. \( \frac{9}{1} \) area ratio

The experiments also covered the following cone angles:

<table>
<thead>
<tr>
<th>Cone Angle</th>
<th>Length of Diffuser (in inches)</th>
</tr>
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<tbody>
<tr>
<td>3°</td>
<td>28.63</td>
</tr>
<tr>
<td>4°</td>
<td>21.50</td>
</tr>
<tr>
<td>5°</td>
<td>17.18</td>
</tr>
<tr>
<td>7 1/2°</td>
<td>11.40</td>
</tr>
<tr>
<td>10°</td>
<td>8.575</td>
</tr>
<tr>
<td>12 1/2°</td>
<td>6.85</td>
</tr>
<tr>
<td>15°</td>
<td>5.70</td>
</tr>
<tr>
<td>17 1/2°</td>
<td>4.88</td>
</tr>
<tr>
<td>20°</td>
<td>4.255</td>
</tr>
<tr>
<td>30°</td>
<td>2.802</td>
</tr>
<tr>
<td>40°</td>
<td>2.060</td>
</tr>
<tr>
<td>50°</td>
<td>1.609</td>
</tr>
<tr>
<td>60°</td>
<td>1.30</td>
</tr>
<tr>
<td>90°</td>
<td>0.75</td>
</tr>
<tr>
<td>180°</td>
<td>0.00</td>
</tr>
</tbody>
</table>
A deficiency in Gibson's experimental work is that the straight section of his inlet pipe was too short for the velocity profile to become fully developed. Data taken from a figure in his paper gives an inlet length of 4 inches, which corresponds to an \( l/D \) ratio of only 2.0 to 2.66. Since experiments of J. Nikuradse and H. Kirsten (3) found that minimum \( l/D \) values of 25 to 50 downstream from an entrance are necessary for the velocity profile in turbulent flow to become fully developed, it is doubtful that the velocity profiles in Gibson's inlet pipes were fully developed. His measured losses may therefore be somewhat different from those that would occur in diffusers with fully developed inlet profiles. In addition, in Gibson's experiments the distance from the diffuser to the downstream pressure measuring orifice, \( l_3 \) (see Fig. 1) is less than 13 inches compared with a 3 in. diameter. Peters has indicated that maximum pressure cannot be reached by a diameters, especially for those cone angles smaller than 40°. (Fig 2)

In 1959 Y. Furuya made an investigation of "Pressure Recovery Efficiency of Short Conical Diffusers and Roughened Diffusers" (ref.5). Cone angles of 10°, 15°, 20°, 30°, 40°, 50°, 60°, 80°, 120°, and 180° were used in his experiments. He considered the effect of both inlet and outlet pipe lengths; although he does not present definite data concerning these lengths. He also considered the effect of roughened walls on efficiency. Unfortunately, he did not systematically investigate effects of the variations of area ratio and Reynolds number. Also, the accuracy of the pressure measurements is somewhat unreliable.

The experiments of Warner and Ziembinski were carried out with small angles of expansion only.

The present study is an extension of the work described above. Its
contribution lines essentially in the effects of area ratio; cone angle, and Reynolds number are investigated systematically.

An inlet length ($l_1$) of $60 \, \text{d}_1$ (for 1 in. diameter, $l_1 = 60$ in.) was taken. This value fulfills both Kirsten's and Nikuradse's requirements for fully developed inlet flow. The location of the point of maximum pressure recovery in the downstream pipe is variable. It depends on inlet flow condition, velocity and cone angle; and is approximately equal to $6 \, \text{d}_2$. Therefore, the lengths of downstream pipe ($l_2$) were from 12 to 32 $\text{d}_2$ in the present experiments, and the locations of pressure measuring orifices ($l_3$) were 9.6 to 24 $\text{d}_2$ from the outlet of the diffuser. The pressure head loss due to pipe friction in both the inlet and outlet pipes has been subtracted after being calculated on the assumption that fully developed flow existed in both pipes. The data of $l_1$, $l_2$, $l_3$ are

<table>
<thead>
<tr>
<th>$d_1$</th>
<th>$d_2$</th>
<th>$l_1/d_1$</th>
<th>$l_2/d_2$</th>
<th>$l_3/d_2$</th>
<th>$l_3/d_1$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.0 in.</td>
<td>1.5 in.</td>
<td>60</td>
<td>32</td>
<td>24</td>
<td>6</td>
</tr>
<tr>
<td>1.0 in.</td>
<td>2.0 in.</td>
<td>60</td>
<td>30</td>
<td>24</td>
<td>6</td>
</tr>
<tr>
<td>1.0 in.</td>
<td>3.0 in.</td>
<td>60</td>
<td>20</td>
<td>16</td>
<td>6</td>
</tr>
<tr>
<td>1.0 in.</td>
<td>4.0 in.</td>
<td>60</td>
<td>15</td>
<td>12</td>
<td>6</td>
</tr>
<tr>
<td>1.0 in.</td>
<td>5.0 in.</td>
<td>60</td>
<td>12</td>
<td>9.6</td>
<td>6</td>
</tr>
</tbody>
</table>

The largest area ratio ($A_2/A_1$) used in Gibson's experiments was 9:1, in Peters', 2.34:1, and in Square's, 4:1. In the author's experiments, the area ratio range was 2.25:1, 4:1, 9:1, 16:1, and 25:1.

The cone angles used in the author's test diffusers were:
10°, 30°, 45°, 60°, 90°, 120°, and 180°.

A wide variety of velocity ranges were also be applied. The maximum inlet velocity was 45 ft/sec., and the maximum Reynolds number based
on inlet flow was $3.5 \times 10^5$. According to Gibson, on plotting the results of his entire series of experiments water flow in conical diffusers, it appears that for values of inlet velocity, $V_i$, greater than 5 feet per second and up to about 23 feet per second (the highest velocity attained in the experiments), the percentage loss of velocity head, i.e., the factor $\mu$, does not vary in any regular manner with the velocity, and it appears to be essentially constant. One of the principal purposes of the author's test is to confirm the independency of diffuser efficiency from Reynolds number within a certain range.

Because of the restriction of time only smooth, straight-wall, conical diffusers were tested. Delay of separation from the walls of diffusers with large cone angles was not attempted.

In order to assure that reasonable accuracy of pressure measurement was attained, a special design of pressure-measuring orifices was used. (Fig. 5) Along the periphery of the pipe section a total of eight small holes (1/16" diameter) was drilled with equal spacing. All holes were drilled normal to the interior surface of the wall, and all burrs and surface roughness near the holes were carefully removed. A vented plenum chamber was welded to the outside of the pipe, so that an average of the pressure at the eight holes was conveyed to the manometer.
IV. EQUIPMENT AND EXPERIMENTAL PROCEDURE

A. Equipment.

A sketch of equipment arrangement is shown in Fig. 4. Water was pumped by a 10 hp, 1750 rpm, centrifugal pump, manufactured by Mission Co., through 2-inch galvanized iron pipe and a reducer into a section of drawn brass tubing which formed the inlet to the diffuser. This tubing had an inside diameter of 1 inch, a wall thickness of 1/8 inch, and a length of 60 inches.

The area ratio of the diffuser determined the diameter of the downstream or outlet pipe, which was aluminum having a wall thickness of 1/4 inch. The size of the large end of diffuser was made exactly the same as the inside diameter of the downstream pipe. The length of the diffuser varied from about 1 inch to 23 inches depending on area ratio and cone angle. Because this diffuser length was variable, a special variable-length coupling was used. The valve next to this expansion joint was used to control the fluid flow rate.

The plenum for the pressure-measurement orifices were welded on upstream and downstream pipes, respectively. The upstream orifices were located 6 inches upstream from the inlet of the diffuser, and the downstream orifices were located 36 inches downstream from the 1 1/2 inches outlet or 48 inches for 2.0 inch, 3.0 inch, 4.0 inch, and 5.0 inch outlets. Pet cocks on the upper side of plenums were used to vent air. Gauge cocks on the bottom side were connected to 1/4-inch plastic tubing leading to a differential manometer having a 50-inch range. Two kinds of gauge liquids were used in the manometer. Mercury was used in measuring larger pressure difference, and a blue oil with a specific gravity of 1.75
was used for small differences.

In order to seal the diffuser section from water leakage, special O-rings were used. The diffuser could be assembled from several pieces with the same cone angle, so that the area ratio could be changed. A total of 35 sections of diffuser were used in these experiments. With different combinations the pressure losses of conical diffusers ranging from $10^\circ$ to $180^\circ$ in cone angle and from $1.5 : 1$ to $25 : 1$ in area ratio at different inlet velocities were obtained.

The construction of typical diffuser sections is shown in Fig. 8.

B. Experimental Procedure.

The path of water flow in these experiments is sketched in Fig. 7. Calibrated tanks were used to measure volume of water. The following steps were followed for each test:

1. Valves A, B, E were opened and valves C, F were closed. The guide D was turned to the direction of Tank I.
2. The pump was then started.
3. After the water flow became stable, the directions of guide D were suddenly changed to Tank II, and the stop watch started simultaneously.
4. After the water in Tank II reached a certain level, the direction of guide D was returned back to Tank I.
5. The time interval as measured by the stop watch was recorded.
6. The water column of Tank II was then measured.
7. The pressure difference from the manometer was recorded.

Since the diameters of upstream and downstream pipes of the diffuser, $d_1$ and $d_2$, are both known, the velocities $V_1$ and $V_2$ in these pipes can
be found. The efficiency and loss of head may be found as follows:

\[ \eta = \frac{\rho_2 - \rho_1}{\rho_2 i - \rho_i} = \frac{\Delta \rho / \rho_2}{V_i^2 / \rho_2 \left[ 1 - \left( \frac{A_1}{A_2} \right)^2 \right]} \]  

(20)

where \( \Delta \rho = \rho_2 - \rho_1 \) = pressure difference between entrance and exit of diffuser. Therefore, the difference of pressure head is equal to the pressure head difference calculated from manometer reading plus head of friction loss in upstream and downstream pipes with length \( l_4 \) and \( l_3 \), respectively.

From Eq. (3)

\[ \frac{\rho_i}{\rho_2} + \frac{V_i^2}{2g} = \frac{\rho_2}{\rho_2} + \frac{V_2^2}{2g} + \Delta h \]

:. \( \Delta h = \left( \frac{V_2^2}{2g} - \frac{V_i^2}{2g} \right) - \left( \frac{\rho_2}{\rho_2} - \frac{\rho_i}{\rho_2} \right) \)

(21)

\[ = \frac{V_i^2}{2g} \left[ 1 - \left( \frac{A_1}{A_2} \right)^2 \right] - \frac{\Delta \rho}{\rho_2} \]

Eq. (17):

\[ \mu = \frac{\Delta h'}{\Delta h} = \frac{\frac{V_i^2}{2g} \left[ 1 - \left( \frac{A_1}{A_2} \right)^2 \right] - \frac{\Delta \rho}{\rho_2}}{\frac{V_i^2}{2g} \left[ 1 - \left( \frac{A_1}{A_2} \right)^2 \right]^2} \]  

(22)

The Reynolds number at upstream flow can be found as follows:

\[ N_R = \frac{V_i d_i}{\nu} \]

(\text{where} \( \nu \) is kinematic viscosity in pound-seconds per square foot).
V. RESULTS AND DISCUSSION OF RESULTS

On plotting the results of the whole series of experiments on conical diffusers, it appears that for Reynolds numbers, within the range of $1.0 \times 10^5$ and $3.5 \times 10^5$ (or from 15 to 45 fps), both the pressure recovery efficiency and loss coefficient do not vary in any regular manner with velocity, or Reynolds number, and appears nearly constant. This phenomenon has been discovered by Gibson. According to his conclusion, the total loss increases very rapidly with cone angle to a maximum of 121 per cent, with an angle of about 63°, afterwards diminishing to 101.7 per cent, with an angle of 180°, i.e., with a sudden change of section. However, after carefully rearranging the experimental setup, and considering the effects of inlet and outlet pipe length, the author found that this maximum point of loss disappeared. The coefficient increases rapidly from 10° to 60° cone angles, then changes gradually thereafter. The level of 100 per cent loss appears as an asymptote, and no loss is higher than that value. Therefore, the 180° cone angle, or an abrupt enlargement is still, in the author's judgment, the least efficient diffuser.

The fact that the pressure drops seriously with cone angle for an angle smaller than 60° shows that the loss coefficient due to friction or boundary layer thicknesses is small but the loss due to flow separation greatly increases as the diverging angle increases from 10° to 60°. It is more interesting that when carefully looking into those curves which are depicted in Fig. 19, we may find out that all curves display the same tendency and almost coincide with one another. Therefore, it would be no serious error if we use a single typical curve to represent
all those curves in Fig. 19; or it still would be not very difficult to
determine out a certain empirical formula to represent them. The curve
is shown in Fig. 20. A formula is herewith proposed to closely repre-
sent this curve:

$$\mu = 1.1 \left[ \theta - \sqrt{(\theta - 44.5)^2 + 136} \right] + 50$$

where $\mu$ is loss coefficient, %;

$\theta$ is cone angle, degree.

$10^\circ \leq \theta \leq 180^\circ$
VI. SUMMARY

1. Both the efficiency and the loss coefficient are nearly independent of Reynolds number, within the range of $1.0 \times 10^5$ to $3.5 \times 10^5$.

2. After comparison with the results of earlier investigations, the deviations due to the neglect of effects of inlet and outlet pipe length and friction losses are evidently important.

3. The loss coefficient of the $180^\circ$ diffuser is nearly 100 per cent. This proves that the assumption in Eq. (15), which states that $p'$ is equal to $p_1$, is valid.

4. The coefficient of loss increases with cone angle. No loss is higher than 100 per cent, with any cone angle.

5. The efficiency drops seriously from a cone angle $10^\circ$ to $60^\circ$, and there is little change after $90^\circ$.

6. The loss coefficients are almost coincident with one another from an area ratio of $2.25 : 1$ to $25 : 1$.

7. For large area ratios, i.e., greater than $9 : 1$, the downstream pressure becomes extremely unstable at about a $90^\circ$ cone angle.

8. An empirical formula of loss coefficient is proposed as follows:

$$
\mu = 1.1 \left[ \theta - \sqrt{(\theta - 44.5)^2 + 136} \right] + 50
$$

where $\mu$ is loss coefficient, in $\%$; $10^\circ \leq \theta \leq 180^\circ$

$\theta$ is cone angle, in degree.
VII. REFERENCES


6. ASME research publication, "Fluid Meters, their Selection and Installation", 1933.


1. Efficiency at exit of diffuser, without outlet length
2. Efficiency at exit of diffuser, with outlet length
3. Efficiency at the maximum pressure section

**FIGURE 2. EFFECT OF INLET AND OUTLET CONDITIONS**

**FIGURE 3. POSITION OF MAXIMUM PRESSURE**
FIGURE 6. DIFFUSER SECTIONS
TO RESERVOIR

H

OS*

FIGURE 7.
SKELETON OF WATER FLOW

MANOMETER

DIFFUSER

EXPANSION JOINT

TANK I

TANK II

TO RESERVOIR

FROM RESERVOIR

PUMP
FIGURE 8. DIFFUSER EFFICIENCY OF AREA RATIO 2.25:1
FIGURE 9. DIFFUSER EFFICIENCY OF AREA RATIO 4:1
FIGURE 11. DIFFUSER EFFICIENCY OF AREA RATIO 16:1
Figure 12. Diffuser Efficiency of Area Ratio 25:1
FIGURE 13. VARIATION OF THE DIFFUSER EFFICIENCY WITH CONE ANGLE
FIGURE 14. LOSS COEFFICIENT OF AREA RATIO 2.25:1
FIGURE 15. LOSS COEFFICIENT OF AREA RATIO 4:1

Reynolds Number (x10^5)

Loss Coefficient (%)
FIGURE 16. LOSS COEFFICIENT OF AREA RATIO 9:1
FIGURE 17  LOSS COEFFICIENT OF AREA RATIO 16:1

REYNOLDS NUMBER \( \times 10^5 \)

\( \eta' \) (\%)

LOSS COEFFICIENT
FIGURE 28. LOSS COEFFICIENT OF AREA RATIO 25:1
FIGURE 19 VARIATION OF THE LOSS COEFFICIENT WITH CONE ANGLE
\[ \mu = 11 \left( \theta - \left[ (\theta - 4.43)^2 + 136 \right] + 50 \right) \]

\[ 10^\circ \leq \theta \leq 180^\circ \]

**Figure 20. Modified Curve of Loss Coefficient**