When a surface is submerged in a fluid, forces develop on the surface due to the fluid. The determination of these forces is important in the design of storage tanks, ships, dams, and other hydraulic structures. For fluids at rest we know that the force must be perpendicular to the surface since there are no shearing stresses present. We also know that the pressure will vary linearly with depth if the fluid is incompressible. For a horizontal surface, such as the bottom of a liquid-filled tank (Fig. 2.16), the magnitude of the resultant force is simply $F_R = \rho A$, where $\rho$ is the uniform pressure on the bottom and $A$ is the area of the bottom. For the open tank shown, $p = \gamma h$. Note that if atmospheric pressure acts on both sides of the bottom, as is illustrated, the resultant force on the bottom is simply due to the liquid in the tank. Since the pressure is constant and uniformly distributed over the bottom, the resultant force acts through the centroid of the area as shown in Fig. 2.16.

For the more general case in which a submerged plane surface is inclined, as is illustrated in Fig. 2.17, the determination of the resultant force acting on the surface is more involved. For the present we will assume that the fluid surface is open to the atmosphere. Let the plane in which the surface lies intersect the free surface at 0 and make an angle $\theta$ with
The resultant force of a static fluid on a plane surface is due to the hydrostatic pressure distribution on the surface.

This surface as in Fig. 2.17. The $x$-$y$ coordinate system is defined so that $0$ is the origin and $y$ is directed along the surface as shown. The area can have an arbitrary shape as shown. We wish to determine the direction, location, and magnitude of the resultant force acting on one side of this area due to the liquid in contact with the area. At any given depth, $h$, the force acting on $dA$ (the differential area of Fig. 2.17) is $dF = \gamma h \, dA$ and is perpendicular to the surface. Thus, the magnitude of the resultant force can be found by summing these differential forces over the entire surface. In equation form

$$ F_R = \int_A \gamma h \, dA = \int_A \gamma y \sin \theta \, dA $$

where $h = \gamma \sin \theta$. For constant $\gamma$ and $\theta$

$$ F_R = \gamma \sin \theta \int_A y \, dA \quad (2.17) $$

Location of resultant force (center of pressure, CP)
The integral appearing in Eq. 2.17 is the first moment of the area with respect to the x axis, so we can write

$$\int_A y \, dA = y_c A$$

where $y_c$ is the y coordinate of the centroid measured from the x axis which passes through 0. Equation 2.17 can thus be written as

$$F_R = \gamma A y_c \sin \theta$$

or more simply as

$$F_R = \gamma h_c A$$

(2.18)

where $h_c$ is the vertical distance from the fluid surface to the centroid of the area. Note that the magnitude of the force is independent of the angle $\theta$ and depends only on the specific weight of the fluid, the total area, and the depth of the centroid of the area below the surface. In effect, Eq. 2.18 indicates that the magnitude of the resultant force is equal to the pressure at the centroid of the area multiplied by the total area. Since all the differential forces that were summed to obtain $F_R$ are perpendicular to the surface, the resultant $F_R$ must also be perpendicular to the surface.

Although our intuition might suggest that the resultant force should pass through the centroid of the area, this is not actually the case. The y coordinate, $y_R$, of the resultant force can be determined by summation of moments around the x axis. That is, the moment of the resultant force must equal the moment of the distributed pressure force, or

$$F_R y_R = \int_A y \, dF = \int_A \gamma \sin \theta \, y^2 \, dA$$

and, therefore, since $F_R = \gamma A y_c \sin \theta$

$$y_R = \frac{\int_A y^2 \, dA}{y_c A}$$

The integral in the numerator is the second moment of the area (moment of inertia), $I_x$, with respect to an axis formed by the intersection of the plane containing the surface and the free surface (x axis). Thus, we can write

$$y_R = \frac{I_x}{y_c A}$$

Use can now be made of the parallel axis theorem to express $I_x$ as

$$I_x = I_{xc} + A y_c^2$$

where $I_{xc}$ is the second moment of the area with respect to an axis passing through its centroid and parallel to the x axis. Thus,

$$y_R = \frac{I_{xc}}{y_c A} + y_c$$

(2.19)

Equation 2.19 clearly shows that the resultant force does not pass through the centroid but is always below it, since $I_{xc}/y_c A > 0$. 

The x coordinate, $x_R$, for the resultant force can be determined in a similar manner by summing moments about the y axis. Thus,

$$F_R x_R = \int_A \gamma \sin \theta \, xy \, dA$$

and, therefore,

$$x_R = \frac{\int_A xy \, dA}{\gamma_c A} = \frac{I_{xy}}{\gamma_c A}$$

where $I_{xy}$ is the product of inertia with respect to the x and y axes. Again, using the parallel axis theorem, we can write

$$x_R = \frac{I_{xye}}{\gamma_c A} + x_c$$

(2.20)

![Geometric properties of some common shapes.](image)

Recall that the parallel axis theorem for the product of inertia of an area states that the product of inertia with respect to an orthogonal set of axes (x-y coordinate system) is equal to the product of inertia with respect to an orthogonal set of axes parallel to the original set and passing through the centroid of the area, plus the product of the area and the x and y coordinates of the centroid of the area. Thus, $I_{xy} = I_{xye} + A x_c y_c$. 

**Figure 2.18** Geometric properties of some common shapes.
where $I_{xye}$ is the product of inertia with respect to an orthogonal coordinate system passing through the centroid of the area and formed by a translation of the $x$-$y$ coordinate system. If the submerged area is symmetrical with respect to an axis passing through the centroid and parallel to either the $x$ or $y$ axes, the resultant force must lie along the line $x = x_c$, since $I_{xye}$ is identically zero in this case. The point through which the resultant force acts is called the center of pressure. It is to be noted from Eqs. 2.19 and 2.20 that as $y_c$ increases the center of pressure moves closer to the centroid of the area. Since $y_c = h_c / \sin \theta$, the distance $y_c$ will increase if the depth of submergence, $h_c$, increases, or, for a given depth, the area is rotated so that the angle, $\theta$, decreases. Centroidal coordinates and moments of inertia for some common areas are given in Fig. 2.18.

The 4-m-diameter circular gate of Fig E2.6a is located in the inclined wall of a large reservoir containing water ($\gamma = 9.80$ kN/m$^3$). The gate is mounted on a shaft along its horizontal diameter. For a water depth of 10 m above the shaft determine: (a) the magnitude and location of the resultant force exerted on the gate by the water, and (b) the moment that would have to be applied to the shaft to open the gate.

**Solution**

(a) To find the magnitude of the force of the water we can apply Eq. 2.18,

$$F_R = \gamma h_c A$$

and since the vertical distance from the fluid surface to the centroid of the area is 10 m it follows that

$$F_R = (9.80 \times 10^3 \text{ N/m}^3)(10 \text{ m})(4\pi \text{ m}^2)$$

$$= 1230 \times 10^3 \text{ N} = 1.23 \text{ MN} \quad \text{(Ans)}$$

To locate the point (center of pressure) through which $F_R$ acts, we use Eqs. 2.19 and 2.20,

$$x_R = \frac{I_{xye}}{y_c A} + x_c \quad y_R = \frac{I_{xye}}{y_c A} + y_c$$
For the coordinate system shown, \( x_R = 0 \) since the area is symmetrical, and the center of pressure must lie along the diameter \( A-A \). To obtain \( y_R \), we have from Fig. 2.18

\[
I_{sc} = \frac{\pi R^4}{4}
\]

and \( y_c \) is shown in Fig. E2.6b. Thus,

\[
y_R = \frac{(\pi/4)(2 \text{ m})^4}{(10 \text{ m}/\sin 60^\circ)(4 \pi \text{ m}^2)} + \frac{10 \text{ m}}{\sin 60^\circ}
\]

\[
= 0.0866 \text{ m} + 11.55 \text{ m} = 11.6 \text{ m}
\]

and the distance (along the gate) below the shaft to the center of pressure is

\[
y_R - y_c = 0.0866 \text{ m}
\]

(Ans)

We can conclude from this analysis that the force on the gate due to the water has a magnitude of 1.23 MN and acts through a point along its diameter \( A-A \) at a distance of 0.0866 m (along the gate) below the shaft. The force is perpendicular to the gate surface as shown.

(b) The moment required to open the gate can be obtained with the aid of the free-body diagram of Fig. E2.6c. In this diagram \( W \) is the weight of the gate and \( O_x \) and \( O_y \) are the horizontal and vertical reactions of the shaft on the gate. We can now sum moments about the shaft

\[
\sum M_e = 0
\]

and, therefore,

\[
M = F_R(y_R - y_c)
\]

\[
= (1230 \times 10^3 \text{ N})(0.0866 \text{ m})
\]

\[
= 1.07 \times 10^5 \text{ N} \cdot \text{m}
\]

(Ans)

---

**Example 2.7**

A large fish-holding tank contains seawater (\( \gamma = 64.0 \text{ lb/ft}^3 \)) to a depth of 10 ft as shown in Fig. E2.7a. To repair some damage to one corner of the tank, a triangular section is replaced with a new section as illustrated. Determine the magnitude and location of the force of the seawater on this triangular area.

**Solution**

The various distances needed to solve this problem are shown in Fig. E2.7b. Since the surface of interest lies in a vertical plane, \( y_c = h_c = 9 \) ft, and from Eq. 2.18 the magnitude of the force is

\[
F_R = \gamma h_c A
\]

\[
= (64.0 \text{ lb/ft}^3)(9 \text{ ft})(9/2 \text{ ft}^2) = 2590 \text{ lb}
\]

(Ans)
Note that this force is independent of the tank length. The result is the same if the tank is 0.25 ft, 25 ft, or 25 miles long. The y coordinate of the center of pressure (CP) is found from Eq. 2.19,

$$y_R = \frac{I_{xc}}{y_cA} + y_c$$

and from Fig. 2.18

$$I_{xc} = \frac{(3 \text{ ft})(3 \text{ ft})^3}{36} = \frac{81}{36} \text{ ft}^4$$

so that

$$y_R = \frac{81/36 \text{ ft}^4}{(9 \text{ ft})(9/2 \text{ ft}^2)} + 9 \text{ ft}$$

$$= 0.0556 \text{ ft} + 9 \text{ ft} = 9.06 \text{ ft} \quad \text{(Ans)}$$

Similarly, from Eq. 2.20

$$x_R = \frac{I_{yce}}{y_cA} + x_c$$

and from Fig. 2.18

$$I_{yce} = \frac{(3 \text{ ft})(3 \text{ ft})^2}{72} (3 \text{ ft}) = \frac{81}{72} \text{ ft}^4$$
so that

\[ x_R = \frac{81/72 \text{ ft}^4}{(9 \text{ ft})(9/2 \text{ ft}^2)} + 0 = 0.0278 \text{ ft} \] (Ans)

Thus, we conclude that the center of pressure is 0.0278 ft to the right of and 0.0556 ft below the centroid of the area. If this point is plotted, we find that it lies on the median line for the area as illustrated in Fig. E2.7c. Since we can think of the total area as consisting of a number of small rectangular strips of area \( \delta A \) (and the fluid force on each of these small areas acts through its center), it follows that the resultant of all these parallel forces must lie along the median.

### 2.9 Pressure Prism

An informative and useful graphical interpretation can be made for the force developed by a fluid acting on a plane area. Consider the pressure distribution along a vertical wall of a tank of width \( b \), which contains a liquid having a specific weight \( \gamma \). Since the pressure must vary linearly with depth, we can represent the variation as shown in Fig. 2.19a, where the pressure is equal to zero at the upper surface and equal to \( \gamma h \) at the bottom. It is apparent from this diagram that the average pressure occurs at the depth \( h/2 \), and therefore the resultant force acting on the rectangular area \( A = bh \) is

\[ F_R = p \cdot A = \gamma \left( \frac{h}{2} \right) A \]

which is the same result as obtained from Eq. 2.18. The pressure distribution shown in Fig. 2.19a applies across the vertical surface so we can draw the three-dimensional representation of the pressure distribution as shown in Fig. 2.19b. The base of this “volume” in pressure-area space is the plane surface of interest, and its altitude at each point is the pressure. This volume is called the pressure prism, and it is clear that the magnitude of the resultant force acting on the surface is equal to the volume of the pressure prism. Thus, for the prism of Fig. 2.19b the fluid force is

\[ F_R = \text{volume} = \frac{1}{2} (\gamma h)(bh) = \gamma \left( \frac{h}{2} \right) A \]

where \( bh \) is the area of the rectangular surface, \( A \).

---

The pressure prism is a geometric representation of the hydrostatic force on a plane surface.

---

**FIGURE 2.19**

Pressure prism for vertical rectangular area.
The resultant force must pass through the centroid of the pressure prism. For the volume under consideration the centroid is located along the vertical axis of symmetry of the surface, and at a distance of $h/3$ above the base (since the centroid of a triangle is located at $h/3$ above its base). This result can readily be shown to be consistent with that obtained from Eqs. 2.19 and 2.20.

This same graphical approach can be used for plane surfaces that do not extend up to the fluid surface as illustrated in Fig. 2.20a. In this instance, the cross section of the pressure prism is trapezoidal. However, the resultant force is still equal in magnitude to the volume of the pressure prism, and it passes through the centroid of the volume. Specific values can be obtained by decomposing the pressure prism into two parts, $ABDE$ and $BCD$, as shown in Fig. 2.20b. Thus,

$$F_R = F_1 + F_2$$

where the components can readily be determined by inspection for rectangular surfaces. The location of $F_R$ can be determined by summing moments about some convenient axis, such as one passing through $A$. In this instance

$$F_R y_A = F_1 y_1 + F_2 y_2$$

and $y_1$ and $y_2$ can be determined by inspection.

For inclined plane surfaces the pressure prism can still be developed, and the cross section of the prism will generally be trapezoidal as is shown in Fig. 2.21. Although it is usually convenient to measure distances along the inclined surface, the pressures developed depend on the vertical distances as illustrated.
The use of pressure prisms for determining the force on submerged plane areas is convenient if the area is rectangular so the volume and centroid can be easily determined. However, for other non-rectangular shapes, integration would generally be needed to determine the volume and centroid. In these circumstances it is more convenient to use the equations developed in the previous section, in which the necessary integrations have been made and the results presented in a convenient and compact form that is applicable to submerged plane areas of any shape.

The effect of atmospheric pressure on a submerged area has not yet been considered, and we may ask how this pressure will influence the resultant force. If we again consider the pressure distribution on a plane vertical wall, as is shown in Fig. 2.22a, the pressure varies from zero at the surface to \( yh \) at the bottom. Since we are setting the surface pressure equal to zero, we are using atmospheric pressure as our datum, and thus the pressure used in the determination of the fluid force is gage pressure. If we wish to include atmospheric pressure, the pressure distribution will be as is shown in Fig. 2.22b. We note that in this case the force on one side of the wall now consists of \( F_R \) as a result of the hydrostatic pressure distribution, plus the contribution of the atmospheric pressure, \( P_{atm} A \), where \( A \) is the area of the surface. However, if we are going to include the effect of atmospheric pressure on one side of the wall we must realize that this same pressure acts on the outside surface (assuming it is exposed to the atmosphere), so that an equal and opposite force will be developed as illustrated in the figure. Thus, we conclude that the resultant fluid force on the surface is that due only to the gage pressure contribution of the liquid in contact with the surface—the atmospheric pressure does not contribute to this resultant. Of course, if the surface pressure of the liquid is different from atmospheric pressure (such as might occur in a closed tank), the resultant force acting on a submerged area, \( A \), will be changed in magnitude from that caused simply by hydrostatic pressure by an amount \( P_s A \), where \( P_s \) is the gage pressure at the liquid surface (the outside surface is assumed to be exposed to atmospheric pressure).

Example 2.8

A pressurized tank contains oil (SG = 0.90) and has a square, 0.6-m by 0.6-m plate bolted to its side, as is illustrated in Fig. E2.8a. When the pressure gage on the top of the tank reads 50 kPa, what is the magnitude and location of the resultant force on the attached plate? The outside of the tank is at atmospheric pressure.

SOLUTION

The pressure distribution acting on the inside surface of the plate is shown in Fig. E2.8b. The pressure at a given point on the plate is due to the air pressure, \( p_a \), at the oil surface, and the
Pressure due to the oil, which varies linearly with depth as is shown in the figure. The resultant force on the plate (having an area $A$) is due to the components, $F_1$ and $F_2$, with

$$F_1 = (p_r + \gamma h_1)A$$

$$= (50 \times 10^3 \text{ N/m}^2 + (0.90)(9.81 \times 10^3 \text{ N/m}^3)(2 \text{ m})/(0.36 \text{ m}^2)$$

$$= 24.4 \times 10^3 \text{ N}$$

and

$$F_2 = \gamma \left( \frac{h_2 - h_1}{2} \right) A$$

$$= (0.90)(9.81 \times 10^3 \text{ N/m}^3) \left( \frac{0.6 \text{ m}}{2} \right) (0.36 \text{ m}^2)$$

$$= 0.954 \times 10^3 \text{ N}$$

The magnitude of the resultant force, $F_R$, is therefore

$$F_R = F_1 + F_2 = 25.4 \times 10^3 \text{ N} = 25.4 \text{ kN}$$

(Ans)

The vertical location of $F_R$ can be obtained by summing moments around an axis through point $O$ so that

$$F_R y_O = F_1(0.3 \text{ m}) + F_2(0.2 \text{ m})$$

or

$$(25.4 \times 10^3 \text{ N})y_O = (24.4 \times 10^3 \text{ N})(0.3 \text{ m}) + (0.954 \times 10^3 \text{ N})(0.2 \text{ m})$$

$$y_O = 0.296 \text{ m}$$

(Ans)

Thus, the force acts at a distance of 0.296 m above the bottom of the plate along the vertical axis of symmetry.

Note that the air pressure used in the calculation of the force was gage pressure. Atmospheric pressure does not affect the resultant force (magnitude or location), since it acts on both sides of the plate, thereby canceling its effect.
The development of a free-body diagram of a suitable volume of fluid can be used to determine the resultant fluid force acting on a curved surface.

The equations developed in Section 2.8 for the magnitude and location of the resultant force acting on a submerged surface only apply to plane surfaces. However, many surfaces of interest (such as those associated with dams, pipes, and tanks) are nonplanar. Although the resultant fluid force can be determined by integration, as was done for the plane surfaces, this is generally a rather tedious process and no simple, general formulas can be developed. As an alternative approach we will consider the equilibrium of the fluid volume enclosed by the curved surface of interest and the horizontal and vertical projections of this surface.

For example, consider the curved section $BC$ of the open tank of Fig. 2.23a. We wish to find the resultant fluid force acting on this section, which has a unit length perpendicular to the plane of the paper. We first isolate a volume of fluid that is bounded by the surface of interest, in this instance section $BC$, the horizontal plane surface $AB$, and the vertical plane surface $AC$. The free-body diagram for this volume is shown in Fig. 2.23b. The magnitude and location of forces $F_1$ and $F_2$ can be determined from the relationships for planar surfaces. The weight, $W$, is simply the specific weight of the fluid times the enclosed volume and acts through the center of gravity (CG) of the mass of fluid contained within the volume. The forces $F_H$ and $F_V$ represent the components of the force that the tank exerts on the fluid.

In order for this force system to be in equilibrium, the horizontal component $F_H$ must be equal in magnitude and collinear with $F_2$, and the vertical component $F_V$ equal in magnitude and collinear with the resultant of the vertical forces $F_1$ and $W$. This follows since the three forces acting on the fluid mass ($F_2$, the resultant of $F_1$ and $W$, and the resultant force that the tank exerts on the mass) must form a concurrent force system. That is, from the principles of statics, it is known that when a body is held in equilibrium by three nonparallel forces they must be concurrent (their lines of action intersect at a common point), and coplanar. Thus,

\[ F_H = F_2 \]
\[ F_V = F_1 + W \]

and the magnitude of the resultant is obtained from the equation

\[ F_R = \sqrt{(F_H)^2 + (F_V)^2} \]

\[ \text{(a)} \]
\[ \text{(b)} \]
\[ \text{(c)} \]

**FIGURE 2.23** Hydrostatic force on a curved surface.
The resultant force passes through the point $O$, which can be located by summing moments about an appropriate axis. The resultant force of the fluid acting on the curved surface $BC$ is equal and opposite in direction to that obtained from the free-body diagram of Fig. 2.23b. The desired fluid force is shown in Fig. 2.23c.

**Example 2.9**

The 6-ft-diameter drainage conduit of Fig. E2.9a is half full of water at rest. Determine the magnitude and line of action of the resultant force that the water exerts on a 1-ft length of the curved section $BC$ of the conduit wall.

![Diagram of the conduit and forces](image)

**Solution**

We first isolate a volume of fluid bounded by the curved section $BC$, the horizontal surface $AB$, and the vertical surface $AC$, as shown in Fig. E2.9b. The volume has a length of 1 ft. The forces acting on the volume are the horizontal force, $F_h$, which acts on the vertical surface $AC$, the weight, $W$, of the fluid contained within the volume, and the horizontal and vertical components of the force of the conduit wall on the fluid, $F_h$ and $F_v$, respectively.

The magnitude of $F_h$ is found from the equation

$$F_h = \gamma h \cdot A = (62.4 \text{ lb/ft}^3)(\frac{\pi}{2} \text{ ft})(3 \text{ ft}^2) = 281 \text{ lb}$$

and this force acts 1 ft above $C$ as shown. The weight, $W$, is

$$W = \gamma \cdot \text{vol} = (62.4 \text{ lb/ft}^3)(9\pi/4 \text{ ft}^3)(1 \text{ ft}) = 441 \text{ lb}$$

and acts through the center of gravity of the mass of fluid, which according to Fig. 2.18 is located 1.27 ft to the right of $AC$ as shown. Therefore, to satisfy equilibrium

$$F_h = F_1 = 281 \text{ lb} \quad F_v = W = 441 \text{ lb}$$

and the magnitude of the resultant force is

$$F_R = \sqrt{(F_h)^2 + (F_v)^2}$$

$$= \sqrt{(281 \text{ lb})^2 + (441 \text{ lb})^2} = 523 \text{ lb} \quad \text{(Ans)}$$

The force the water exerts on the conduit wall is equal, but opposite in direction, to the forces $F_h$ and $F_v$ shown in Fig. E2.9b. Thus, the resultant force on the conduit wall is shown in Fig. E2.9c. This force acts through the point $O$ at the angle shown.
An inspection of this result will show that the line of action of the resultant force passes through the center of the conduit. In retrospect, this is not a surprising result since at each point on the curved surface of the conduit the elemental force due to the pressure is normal to the surface, and each line of action must pass through the center of the conduit. It therefore follows that the resultant of this concurrent force system must also pass through the center of concurrence of the elemental forces that make up the system.

This same general approach can also be used for determining the force on curved surfaces of pressurized, closed tanks. If these tanks contain a gas, the weight of the gas is usually negligible in comparison with the forces developed by the pressure. Thus, the forces (such as $F_1$ and $F_2$ in Fig. 2.23b) on horizontal and vertical projections of the curved surface of interest can simply be expressed as the internal pressure times the appropriate projected area.

### 2.11 Buoyancy, Flotation, and Stability

#### 2.11.1 Archimedes' Principle

When a body is completely submerged in a fluid, or floating so that it is only partially submerged, the resultant fluid force acting on the body is called the *buoyant force*. A net upward vertical force results because pressure increases with depth and the pressure forces acting from below are larger than the pressure forces acting from above. This force can be determined through an approach similar to that used in the previous article for forces on curved surfaces. Consider a body of arbitrary shape, having a volume $V$, that is immersed in a fluid as illustrated in Fig. 2.24a. We enclose the body in a parallelepiped and draw a free-body diagram of the parallelepiped with the body removed as shown in Fig. 2.24b. Note that the forces $F_1, F_2, F_3,$ and $F_4$ are simply the forces exerted on the plane surfaces of the parallelepiped (for simplicity the forces in the $x$ direction are not shown), $W$ is the weight of the shaded fluid volume (parallelepiped minus body), and $F_B$ is the force the body is exerting on the fluid. The forces on the vertical surfaces, such as $F_3$ and $F_4$, are all equal and cancel, so the equilibrium equation of interest is in the $z$ direction and can be expressed as

$$F_B = F_2 - F_1 - W$$

(2.21)

If the specific weight of the fluid is constant, then

$$F_2 - F_1 = \gamma(h_2 - h_1)A$$

where $A$ is the horizontal area of the upper (or lower) surface of the parallelepiped, and Eq. 2.21 can be written as

$$F_B = \gamma(h_2 - h_1)A - \gamma(h_2 - h_1)A - V$$

Simplifying, we arrive at the desired expression for the buoyant force

$$F_B = \gamma V$$

(2.22)

where $\gamma$ is the specific weight of the fluid and $V$ is the volume of the body. The direction of the buoyant force, which is the force of the fluid on the body, is opposite to that shown on the free-body diagram. Therefore, the buoyant force has a magnitude equal to the weight of the displaced fluid.