Seventh Edition

MECHANICAL ENGINEERING

(OBJECTIVE TYPE)

[with Multiple Choice Questions and Answers]

(Including Brief Theory)

Dr. R.K. Bansal

MECHANICAL ENGINEERING
(OBJECTIVE TYPE)

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[*For Gate Test, U.P.S.C. Engineering Services Examinations (I.E.S.), I.A.S. Exams, (Engg. Group), Objective Type Tests of Public Undertaking and Private Limited Companies, Interviews, Degree Examinations, Quick Review of the Subject and other Companies Tests*]

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to

the loving memory of

my daughter, Babli

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PREFACE TO THE SEVENTH EDITION

The seventh edition has been thoroughly revised and enlarged. A new chapter on General Engineering that consists of subjects Physics, Chemistry, Mathematics and Electrical sciences has been included. The questions of 'Fill in the blanks' and 'Mark the true and false statements' have been added. These types of questions are generally asked in the latest test papers of most of the organisations. The text of each chapter has been thoroughly revised and enlarged.

Two typical objective type test papers, each containing 224 objective type questions from all chapters are given in the last chapter (chapter-15) for the sake of confidence and revision.

The book contains following 15 chapters:

1. Fluid Mechanics and Hydraulic Machines
2. Engineering Mechanics
3. Thermodynamics
4. Internal Combustion Engines and Nuclear Power Plants
5. Steam Boilers, Engines, Nozzle and Turbines
6. Compressor, Gas Turbines and Jet Engines
7. Heat Transfer, Refrigeration and Air Conditioning
8. Strength of Materials
9. Theory of Machines
10. Machine Design
11. Engineering Materials
12. Production Engineering
13. Industrial Engineering and Production Management
14. General Engineering and General Aptitude
15. Typical Objective Type Test Papers.

At the end of each chapter additional objective type questions from competitive examinations such as Gate, I.E.S. (Indian Engineering Service) Examination in Mech. Engg., I.A.S. (Indian Administrative Service) Examination and objective type tests of public undertakings and private limited companies have been included. To make the book more useful to the students and for quick review of the subject brief theory is given at the beginning of each chapter.

Though every care has been taken in checking the manuscripts and proofreading, yet claiming perfection is very difficult. I shall be very tankful to the readers and users of this book for pointing out any mistakes that might have crept in. Suggestions for improvement are most welcome and would be incorporated in the next edition with a view to make the book more useful.

—Author

Preface to the First Edition

I am glad to present the book entitled, 'Mechanical Engineering (Objective Type)' to the Engineering students preparing for the Graduate Aptitude Test in Engineering (GATE), U.P.S.C. Indian Engineering Service (I.E.S.) Examination in Mechanical Engineering, U.P.S.C. Indian Administrative Service (I.A.S.). Examination in Mechanical Engineering, objective type tests of public undertakings and private limited companies and interviews. The course-contents have been planned in such a way that the general requirements of all the above examinations are fulfilled.

The trend for objective type examination is on the increase. Since 1978, U.P.S.C. has started conducting objective type examinations for Indian Engineering Service examination on and Indian Administrative Service examination. Also the students, seeking admission to all Post Graduate degree course in Engineering/Technology in the country, will have to qualify the Graduate Aptitude Test in Engineering (GATE) with effect from July/August 1984 admission with Scholarship. These requirements have been kept in mind, while writing this book.

The book is written in a simple and easy-to-follow language, so that even an average student can grasp the subject by self study. To make the book more useful to the students and for quick review of this subject, brief conventional type theory is given at the beginning of each chapter. The book serves the purpose for both the papers I and II of Preliminary and Main examination of I.E.S. (Indian Engg. Service). At the end of each chapter, answers to all questions have been given. The typical objective type test papers each consisting 140 objective type questions from all chapters, are given in the end for the sake of revision.

To write a book of this type, a large number of standard books have been consulted. I am thankful to these authorities whose works have been consulted and give me a great help in preparing the book.

I express my appreciation and gratefulness to my publisher Shri R.K. Gupta (a Mechanical Engineer) for his most co-operative, painstaking attitude and untiring efforts for bringing out the book in a short period.

Mrs. Nirmal Bansal deserve special credit as she not only provided an ideal atmosphere at home for book writing but also gave inspiration and valuable suggestions.

Though every care has been taken in checking the manuscripts and proofreading, yet claiming perfection is very difficult. I shall be very thankful to the readers and users of this book for pointing out any mistakes that might have crept in. Suggestions for improvement are most welcome and would be incorporated in the next edition with a view to make the book more useful.

—Author

Chapter 1

FLUID MECHANICS AND HYDRAULIC MACHINES

I. THEORY

1.1. DEFINITIONS AND FLUID PROPERTIES

Fluid mechanics is that branch of science which deals with the behaviour of the fluid (*i.e.,* liquids or gases) when they are at rest or in motion. When the fluids are at rest, there will be no relative motion between adjacent fluid layers and hence velocity gradient $\left(\frac{du}{dy}\right)$, which is defined as the change of velocity between two adjacent fluid layers divided by the distance between the layers, will be zero. Also the shear stress $\tau = \mu \frac{du}{dy}$ will be zero in which $\frac{du}{dy}$ is the velocity gradient or **rate of shear strain.**

The law, which states that the shear stress (τ) is directly proportional to the rate of shear strain $\left(\frac{du}{dy}\right)$, is called **Newton's Law of viscosity.** Fluids which obey Newton's law of viscosity are known as **Newtonian fluids** and the fluids which do not obey this law are called **Non-Newtonian fluids.**

(*i*) **Density** or **mass density.** It is defined as the mass per unit volume of a fluid and is denoted by the symbol ρ (rho).

(*ii*) **Weight density** or **specific weight**. It is defined as the weight per unit volume of a fluid and is denoted by the symbol w.

Mathematically, $\rho = \dfrac{\text{Mass of fluid}}{\text{Volume of fluid}}$

and $$w = \frac{\text{Weight of fluid}}{\text{Volume of fluid}} = \frac{\text{Mass of fluid} \times g}{\text{Volume}} = \rho \times g.$$

The value of density (ρ) for water is 1000 kg/m^3 and of specific weight or weight density (w) is 1000 × 9.81 N/m^3 or 9810 N/m^3 in S.I. units.

(*iii*) **Specific volume.** It is defined as volume per unit mass and hence it is the reciprocal of mass density. **Specific gravity** is the ratio of weight density or mass density of the fluid to the weight density or mass density of a standard fluid at a standard temperature. For liquids, water is taken as a standard fluid at 4°C and for gases, air is taken as standard fluid.

(*iv*) **Viscosity.** It is defined as the property of a fluid which offers resistance to the movement of one layer of fluid over another adjacent layer of the fluid. Unit of viscosity in MKS is expressed as $\dfrac{\text{kgf-sec}}{\text{m}^2}$, in SI system as $\dfrac{\text{Ns}}{\text{m}^2}$ and in CGS as $\dfrac{\text{dyne-sec}}{\text{m}^2}$. The unit of viscosity in CGS is also called Poise.

The equivalent numerical value of one poise in MKS units is obtained by dividing 98.1 and in SI units is obtained by dividing 10.

Kinematic viscosity is defined as the ratio of dynamic viscosity to density of fluid. It is denoted by the Greek symbol (ν) called 'nu'. Unit of kinematic viscosity in MKS is m^2/sec and in CGS is cm^2/sec which is also called stoke. The viscosity of a liquid decreases with the increase of temperature while the viscosity of the gas increases.

(*v*) **Compressibility.** It is the reciprocal of the bulk modulus of elasticity, which is defined as the ratio of compressive stress to volumetric strain. Mathematically,

$$\text{Bulk Modulus} = \frac{\text{Increase of pressure}}{\text{Volumetric strain}} = \frac{dp}{-\left(\dfrac{dV}{V}\right)}$$

$$\therefore \quad \text{Compressibility} = \frac{1}{\text{Bulk modulus}} = \frac{-\left(\dfrac{dV}{V}\right)}{dp}$$

(*vi*) **Surface tension.** It is defined as the tensile force acting on the surface of a liquid in contact with a gas such that the contact surface behaves like a membrance under tension. It is expressed as force per unit length and is denoted by σ (called sigma). Hence unit of surface tension in MKS is kgf/m while in SI is N/m.

The relation between surface tension (σ) and difference of pressure (p) between inside and outside of a liquid drop is given by $p = \dfrac{4\sigma}{d}$

For a soap bubble, $p = \dfrac{8\sigma}{d}$

For a liquid jet, $p = \frac{2\sigma}{d}$.

(*vii*) **Capillarity.** It is defined as a phenomenon of rise or fall of a liquid surface in a small vertical tube held in a liquid relative to general level of the liquid. The rise or fall of liquid is given by

$$h = \frac{4\sigma \cos \theta}{wd}$$

where d = Dia. of tube

θ = Angle of contact between liquid and glass tube.

(*viii*) **Ideal fluid** is a fluid which offers no resistance to flow and is incompressible. Hence for ideal fluid viscosity (μ) is zero and density (ρ) is constant.

(*ix*) **Real fluid** is a fluid which offers resistance to flow. Hence viscosity for real fluid is not zero.

1.2. PRESSURE AND ITS MEASUREMENT

Pressure at a point is defined as the force per unit area. The Pascal's law states that intensity of pressure for a fluid at rest is equal in all directions. The pressure at any point in a incompressible fluid (*i.e.,* liquid) at rest is equal to the product of weight density of fluid and vertical height from free surface of the liquid.

Mathematically, $p = wz = \rho g z$.

(*i*) **Hydrostatic law** states that the rate of increase of pressure in the vertically downward direction is equal to the specific weight of the fluid *i.e.,* $\frac{dp}{dz} = w = \rho g$.

(*ii*) **Absolute pressure** is the pressure measured with reference to absolute zero pressure while gauge pressure is the pressure measured with reference to atmospheric pressure. Thus the pressure above the atmospheric pressure is called gauge pressure. Vacuum pressure is the pressure below the atmospheric pressure.

Mathematically,

Gauge pressure = Absolute pressure – Atmospheric pressure

Vacuum pressure = Atmospheric pressure – Absolute pressure.

(*iii*) **Manometers** are defined as the devices used for measuring the pressure at a point in a fluid. They are classified as:

1. Simple Manometers, and
2. Differential Manometers.

Simple manometers are used for measuring pressure at a point while differential manometers are used for measuring the difference of pressures between the two points in a pipe or two different pipes.

(*iv*) The pressure at a point in a static compressible fluid is obtained by combining two equations *i.e.,* equation of state for a gas $\left(\frac{p}{\rho} = RT\right)$ and the equation given by hydrostatic law

$\left(\frac{dp}{dz} = -\rho g\right)$. For isothermal process, the pressure at a height Z in a static compressible fluid is given as $p = p_o e^{-gZ/RT}$

(v) For adiabatic process the pressure and temperature at a height Z are

$$p = p_o\left[1 - \frac{\gamma - 1}{\gamma}\frac{gZ}{RT_o}\right]^{\frac{\gamma-1}{\gamma}} \text{ and } T = T_o\left[1 - \frac{\gamma - 1}{\gamma}\frac{gZ}{RT_o}\right]$$

where p_o = Absolute pressure at ground or sea-level

R = Gas constant, γ = Ratio of specific heats

T_o = Temperature at ground or sea-level.

1.3. HYDROSTATIC FORCES ON PLANE SURFACES

The **force** exerted by a static liquid on a vertical, horizontal and inclined surface immersed in the liquid is given by

$$F = \rho g A\bar{h}$$

where ρ = Density of the liquid

A = Area of the immersed surface

$\bar{h}$ = Depth of the centre of gravity of the immersed surface from free surface of the liquid.

(i) **Centre of pressure** is defined as the point of application of the resultant pressure on the surface. The depth of centre of pressure (h*) from free surface of the liquid is given by

$$h^* = \frac{I_G \sin^2\theta}{A\bar{h}} + \bar{h} \quad \text{for inclined surface}$$

$$= \frac{I_G}{A\bar{h}} + \bar{h} \quad \text{for vertical surface}$$

The centre of pressure for a plane vertical surface lies at a depth of two-third the total height of the immersed surface from free surface.

(ii) **The total force on a curved surface** is given by $F = \sqrt{F_x^{\,2} + F_y^{\,2}}$

where F_x = Horizontal force on a curved surface and is equal to total pressure force on the projected area of the curved surface on the vertical plane

and F_y = Vertical force on the curved surface and is equal to the weight of the liquid actually or virtually supported by the curved surface.

The inclination of the resultant force on curved surface with horizontal is given by $\tan\theta = \frac{F_y}{F_x}$.

(*iii*) **The resultant force on a sluice gate** is given by

$$F = F_1 - F_2$$

where F_1 = Pressure force on the upstream side of the sluice gate

F_2 = Pressure force on the downstream side of the sluice gate.

(*iv*) **Lock-gates.** For a lock-gate, the reaction between the two gates (P) is equal to the reaction at the hinge (R), *i.e.*, $R = P$ and the reaction between the two gates (P) is given by $P = \dfrac{F}{2 \sin \theta}$

where F = Resultant water pressure on the lock-gate = $F_1 - F_2$

and θ = Inclination of the gate with the normal to the side of the lock.

1.4. BUOYANCY AND FLOATATION

Buoyant force is the upward force or thrust exerted by a liquid on body when the body is immersed in the liquid. The point through which the buoyant force is supposed to act is called **centre of buoyancy.** It is denoted by B. The point, about which a floating body starts oscillating when the body is given a small angular displacement, is known as **Metacentre.** It is denoted by M. The distance between the meta-centre (M) and centre of gravity (G) of a floating body is known as **metacentric height**. This is denoted by GM and mathematically it is given as

$$GM = \frac{I}{V} - BG$$

where I = Moment of Inertia of the plan of the floating body at the water surface

V = Volume of the body submerged in water

BG = Distance between the centre of gravity (G) and centre of buoyancy (B).

(*i*) **Conditions of equilibrium** of a floating and submerged body are:

Equilibrium	*Floating body*	*Submerged body*
(*i*) Stable	M should be above G	B should be above G
(*ii*) Unstable	M should be below G	B should be below G
(*iii*) Neutral	M and G coincide	B and G coincide

(*ii*) **The metacentric height (GM)** experimentally is given by

$$GM = \frac{wx}{W \tan \theta}$$

where w = Movable weight

x = Distance through which w is moved

W = Weight of floating body including w

θ = Angle through which floating body is tilted

(*iii*) **The time period of oscillation** of a floating body is given by $T = 2\pi \sqrt{\dfrac{k^2}{GM \times g}}$

where k = Radius of gyration, GM = Metacentric height.

1.5. KINEMATICS OF FLUID

Kinematics is defined as that branch of science which deals with the study of fluid in motion without considering the forces causing the motion. The fluids flow may be compressible or incompressible; steady or unsteady; uniform or non-uniform; laminar or turbulent; rotational or irrotational; one, two or three dimensional.

(*i*) If the density (ρ) changes from point to point during fluid flow, it is known **compressible flow.** But if density (ρ) is constant during fluid flow, it is called **incompressible flow.** Mathematically,

$$\rho \neq \text{Constant for compressible flow}$$

$$\rho = \text{Constant for incompressible flow.}$$

(*ii*) If the fluid characteristic like velocity, pressure, density, etc. do not change at a point with respect to time, the fluid flow is known as **steady flow.** If these fluid characteristic change with respect to time, the fluid flow is known as unsteady flow. Mathematically,

$$\left(\frac{\partial v}{\partial t}\right) = 0, \left(\frac{\partial p}{\partial t}\right) = 0 \text{ or } \left(\frac{\partial \rho}{\partial t}\right) = 0 \text{ for steady flow, and}$$

$$\left(\frac{\partial v}{\partial t}\right) \neq 0, \left(\frac{\partial p}{\partial t}\right) \neq 0 \text{ or } \left(\frac{\partial \rho}{\partial t}\right) \neq 0 \text{ for unsteady flow.}$$

(*iii*) If the velocity in a fluid flow does not change with respect to the length of direction of flow, the flow is said **uniform** and if the velocity change it is known **non-uniform** flow. Mathematically,

$$\left(\frac{\partial v}{\partial s}\right) = 0 \text{ for uniform, and } \left(\frac{\partial v}{\partial s}\right) \neq 0 \text{ for non-uniform flow.}$$

(*iv*) If the Reynold number (R_e) in a pipe is less than 2000, the flow is said to be **laminar** and if the Reynold number is more than 4000, the flow is said to be **turbulent.**

Reynolds number (R_e) is given by $R_e = \dfrac{\rho VD}{\mu}$ or $\dfrac{VD}{\nu}$

where V = Velocity of fluid, D = Dia. of pipe

μ = Viscosity of fluid, ν = Kinematic viscosity of fluid.

(*v*) If the fluid particles while flowing along stream lines also rotate about their own axis, that flow is known as **rotational flow** and if the fluid particles, while flowing along stream lines, do not rotate about their own axis, that type of flow is called **irrotational flow.**

(*vi*) The rate of discharge for incompressible fluid is given by

$$Q = A \times V$$

(*vii*) **Continuity equation** is written is general form as

$$\rho AV = \text{constant}$$

and in differential form as $\dfrac{\partial u}{\partial x} + \dfrac{\partial v}{\partial y} + \dfrac{\partial w}{\partial z} = 0$ for three-dimensional flow and $$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \qquad \text{for two-dimensional flow}$$

(*viii*) The components of acceleration in x, y and z direction are

$$a_x = u\frac{\partial u}{\partial x} + v\frac{\partial v}{\partial y} + w\frac{\partial u}{\partial z} + \frac{\partial u}{\partial t}$$

$$a_y = u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y} + w\frac{\partial v}{\partial z} + \frac{\partial v}{\partial t}$$

$$a_z = u\frac{\partial w}{\partial x} + v\frac{\partial w}{\partial y} + w\frac{\partial w}{\partial z} + \frac{\partial w}{\partial t}$$

(*ix*) **Local acceleration** is defined as the rate of change of velocity at a given point. In the above components of acceleration the expressions $\frac{\partial u}{\partial t}$, $\frac{\partial v}{\partial x}$ and $\frac{\partial w}{\partial t}$ are called local acceleration.

(*x*) **Convective acceleration** is defined as the rate of change of velocity due to change of position of fluid particles in a fluid flow.

(*xi*) **Velocity potential function** (ϕ) is defined as the scalar function of space and time such that its negative derivative with respect to any direction gives the fluid velocity in that direction. Hence the components of velocity in x, y and z direction in terms of velocity potential are

$$u = -\frac{\partial \phi}{\partial x},\ v = -\frac{\partial \phi}{\partial y} \text{ and } w = -\frac{\partial \phi}{\partial z}.$$

(*xii*) **Stream function** (ψ) is defined as the scalar function of space and time, such that its partial derivative with respect to any direction gives the velocity component at right angles to that direction. It is defined only for two-dimensional flow. The velocity components in x and y directions in terms of stream function are

$$u = -\frac{\partial \psi}{\partial y} \quad \text{and} \quad v = \frac{\partial \psi}{\partial x}.$$

(*xiii*) **Equipotential line** is a line along which the velocity potential (ϕ) is constant. A grid obtained by drawing a series of equipotential lines and stream lines is called a flow net.

(*xiv*) **Angular deformation** or shear deformation is defined as the average change in the angle contained by two adjacent sides. It is also called shear strain rate and is given by

$$\text{Shear strain rate} = \frac{1}{2}\left[\frac{\partial v}{\partial x} + \frac{\partial u}{\partial y}\right]$$

Rotational components of a fluid particle are given as

$$\omega_z = \frac{1}{2}\left[\frac{\partial v}{\partial x} - \frac{\partial u}{\partial y}\right],\ \omega_x = \frac{1}{2}\left[\frac{\partial w}{\partial y} - \frac{\partial v}{\partial z}\right],\ \omega_y = \frac{1}{2}\left[\frac{\partial u}{\partial z} - \frac{\partial w}{\partial x}\right]$$

Vorticity is equal to two times the value of rotation.

(*xv*) **Vortex flow** is defined as the flow of a fluid along a curved path. It is of two types namely (*i*) Forced vortex flow and (*ii*) Free vortex flow. If the fluid particles are moving round a curved path with the help of some external torque the flow is called **forced vortex flow.** And if no external torque is acquired to rotate the fluid particles, the flow is called **free-vortex flow.** The relation between tangential velocity and radius for vortex flow is given by

$$r = \omega \times r \qquad \text{for forced vortex}$$

$$v \times r = \text{constant} \qquad \text{for free vortex.}$$

The pressure variation along the radial direction for vortex flow along a horizontal plane,

$$\frac{\partial p}{\partial r} = \frac{\rho v^2}{r}$$

For forced vortex flow, $z = \dfrac{v^2}{2g}$

For free vortex flow the equation is $\dfrac{p_1}{\rho g} + \dfrac{v_1^2}{2g} + z_1 = \dfrac{p_2}{\rho g} + \dfrac{v_2^2}{2g} + z_2$.

1.6. DYNAMICS OF FLUID

Dynamics of fluid flow is defined that branch of science which deals with the study of fluids in motion considering the forces which cause the flow.

(*i*) **Euler's equation** of motion is obtained by considering forces due to pressure and gravity. **Navier-Strokes equations** are obtained by considering pressure force, gravity force and viscous force. **Reynold's equation of motion** are obtained by considering pressure force, gravity force, viscous force and force due to turbulence.

(*ii*) **Bernoulli's equation** is obtained by integrating the Euler's equation of motion. It states that, "For a steady, ideal flow of an incompressible fluid, the total energy which consists of pressure energy $\left(\dfrac{p}{\rho g}\right)$, kinetic energy $\left(\dfrac{v^2}{2g}\right)$ and datum energy (z) at any point of the fluid is constant."

Mathematically, it is written as

$$\frac{p}{\rho g} + \frac{v^2}{2g} + z = \text{constant}$$

or

$$\frac{p_1}{\rho g} + \frac{v_1^2}{2g} + z_1 = \frac{p_2}{\rho g} + \frac{v_2^2}{2g} + z_2$$

Bernoulli's equation for **real fluids** is written as

$$\frac{p_1}{\rho g} + \frac{v_1^2}{2g} + z_1 = \frac{p_2}{\rho g} + \frac{v_2^2}{2g} + z_2 + h_L$$

where h_L = Loss of energy between section 1 and 2.

(*iii*) **Applications of Bernoulli's equation are:**

(*a*) Venturimeter, (*b*) Orificemeter, and (*c*) Pitot-tube.

The discharge through a venturimeter is given by

$$Q = C_d \frac{A_1 A_2}{\sqrt{A_1^2 - A_2^2}} \times \sqrt{2gh}$$

where h = Difference of pressure head in terms of fluid head flowing through venturimeter for a horizontal venturimeter

C_d = Coefficient of venturimeter

A_1 = Area at the inlet of venturimeter

A_2 = Area at the throat of the venturimeter.

The value of 'h' is given by the differential U-tube manometer. For a horizontal venturimeter or inclined venturimeter

$$h = x\left[\frac{S_h}{S_f} - 1\right] \quad \text{for manometer with heavier liquid}$$

$$= x\left[1 - \frac{S_l}{S_f}\right] \quad \text{for manometer with lighter liquid}$$

where x = Difference in the readings of the differential manometer

S_h = Specific gravity of heavier liquid in manometer

S_f = Specific gravity of liquid flowing through venturimeter

S_l = Specific gravity of lighter liquid in manometer.

(*iv*) **Pitot tube** is used to find the velocity of a flowing fluid at any point in a pipe or a channel. The velocity is given by the relation,

$$V = C_v \sqrt{2gh}$$

where C_v = Coefficient of pitot tube,

and h = Difference of the pressure head.

(*v*) **Momentum equation** states that the net force acting on a fluid mass is equal to the change in momentum per second (or rate of change of momentum) in that direction. Mathematically, it is written as

$$F = \frac{d}{dt}(mv)$$

where mv = Momentum.

(*vi*) **The impulse-momentum equation** is given by

$$F \times dt = d\,(mv)$$

and it states that the impulse of a force (F) acting on a fluid mass (m) in a short interval of time (dt) is equal to the change of momentum $d(mv)$ in the direction of force.

(*vii*) **Force on a bend,** exerted by a flowing fluid in the direction of x and y, are given by

$$F_x = \rho Q[v_{1x} - v_{2x}] + (p_1A_1)_x + (p_2A_2)_x$$
$$F_y = \rho Q[v_{1y} - v_{2y}] + (p_1A_1)_y + (p_2A_2)_y$$

where v_{1x} = Initial velocity in the x-direction

v_{2x} = Final velocity in the x-direction

$(p_1A_1)_x$ = Initial pressure force in x-direction

$(p_2A_2)_x$ = Final pressure force in x-direction and so on

Resultant force on the bend is $F_R = \sqrt{F_x^2 + F_y^2}$.

1.7. ORIFICE AND MOUTHPIECE

Orifice is a small opening of any cross-section on the side or at the bottom of a tank, through which a fluid is flowing. **A mouthpiece** is a short length of a pipe which is two or three times its diameter in length, fitted in a tank or vessel containing the fluid.

Hydraulic Coefficients

(*i*) There are three *hydraulic coefficients* namely,

(*a*) Coefficient of velocity, C_v (*b*) Coefficient of contraction, C_c

(*c*) Coefficient of discharge, C_d.

(*ii*) The expression for coefficient of velocity in terms of x, y coordinates from vena-contracta is

$$C_v = \frac{x}{\sqrt{4yH}}$$

where H = Height of water from the centre of orifice.

(*iii*) Coefficient of discharge for different types of mouthpieces are

(*a*) $C_d = 0.855$ for external mouthpiece

(*b*) $C_d = 0.707$ for internal mouthpiece running full

(*c*) $C_d = 0.50$ for internal mouthpiece running free

(*d*) $C_d = 1.0$ for convergent or convergent divergent.

1.8. NOTCH AND WEIR

Notch is a device used for measuring the rate of flow of a liquid through a small channel. A **weir** is a concrete or masonry structure placed in the open channel over which the flow occurs.

(*i*) The **discharge** through the following notches or weirs is given by

$$Q = \tfrac{2}{3} C_d \times L \times H^{3/2} \quad \text{for rectangular notch or weir}$$

$$= \frac{8}{15} C_d \times \tan\frac{\theta}{2} \times \sqrt{2g} \times H^{5/2} \quad \text{for a triangle notch or weir.}$$

(*ii*) **The discharge** through a trapezoidal notch or weir is equal to the sum of discharge through a rectangular notch and the discharge through a triangular notch.

(*iii*) **The error** in discharge due to error in measurement of head over a notch is given by

$$\frac{dQ}{Q} = \frac{3}{2}\frac{dH}{H} \qquad \text{for a rectangular notch}$$

$$= \frac{5}{2}\frac{dH}{H} \qquad \text{for a triangular notch}$$

where Q = Discharge through notch, and H = Head over the notch.

An error of 1% in measuring H will produce 1.5% error in discharge over a rectangular notch and 2.5% error in discharge over a triangular notch.

(*iv*) **Velocity of approach** (V_a) is defined as the velocity with which the water approaches the notch or weir. This is given by

$$V_a = \frac{\text{Discharge over the notch}}{\text{Cross-sectional area of channel}}$$

The head due to velocity of approach is given by $h_a = \dfrac{V_a^2}{2g}$

Discharge over a rectangular weir, considering velocity of approach is given by

$$Q = \tfrac{2}{3} C_d L \sqrt{2g}\,[(H_1 + h_a)^{3/2} - h_a{}^{3/2}]$$

1.9. VISCOUS FLOW

Viscous flow is the flow for which Reynold number is less than 2000 or the fluid flows in layers.

(*i*) For the viscous flow through **circular pipes,** the shear stress distribution, velocity distribution, ratio of maximum velocity to average velocity and difference of pressure head are given by

(*a*) $\tau = -\dfrac{\partial p}{\partial x}\dfrac{r}{2}$ (Linear)

(*b*) $u = -\dfrac{1}{4\mu}\dfrac{\partial p}{\partial x}[R^2 - r^2] = U_{\max}\left[1 - \left(\dfrac{r}{R}\right)^2\right]$ (Parabolic)

(*c*) $\dfrac{U_{\max}}{\overline{u}} = 2.0$

(*d*) $h_f = \dfrac{32\mu\overline{u}L}{\rho g D^2}$

where $\dfrac{\partial p}{\partial x}$ = Pressure gradient, τ = Shear stress

r = Radius at any point, R = Radius of the pipe

$U_{\max}$ = Maximum velocity, h_f = Loss of pressure head

$\overline{u}$ = Average velocity, D = Diameter of pipe.

(*ii*) For viscous flow between **two parallel plates,** the shear stress distribution, velocity distribution, ratio of maximum velocity to average velocity and difference of pressure head are given by

(*a*) $\tau = -\frac{1}{2}\frac{\partial p}{\partial x}[t - 2y]$ (Linear) (*b*) $u = -\frac{1}{2\mu}\frac{\partial p}{\partial x}[ty - y^2]$ (Parabolic)

(*c*) $\frac{U_{max}}{\bar{u}} = 1.5$ (*d*) $h_f = \frac{12\mu\bar{u}L}{\rho g t^2}$.

where t = Distance between two plates,

y = Distance from the plates.

(*iii*) **Kinetic energy correction factor** (α) is defined as the ratio of kinetic energy per second based on actual velocity to kinetic energy per second based on average velocity. For a circular pipe through which viscous flow is taking place, $\alpha = 2.0$.

(*iv*) **Momentum correction factor** (β) is defined as the ratio of momentum of a fluid based on actual velocity to the momentum of the fluid based on average velocity. For a circular pipe, having viscous flow, $\beta = \frac{4}{3}$.

(*v*) The **coefficient of friction** (f) which is a function of Reynold number is given by

$$f = \frac{16}{R_e} \quad \text{for viscous flow or for } R_e < 2000$$

$$= \frac{0.079}{R_e^{1/4}} \quad \text{for } R_e \text{ varying from 4000 to } 10^5$$

$$= 0.0008 + \frac{0.05525}{R_e^{0.237}} \quad \text{for } R_e \geq 10^5 \text{ but } \leq 4 \times 10^7.$$

1.10. TURBULENT FLOW

Smooth and rough boundaries. If the average height (k) of the irregularities projecting from the surface of the boundary is small compared with the thickness of the laminar sub-layer (δ'), the boundary is known as **smooth.** But if k is large in comparison to δ', the boundary is known as **rough.** Mathematically,

$$\frac{k}{\delta'} < 0.25 \quad \text{for smooth boundary}$$

$$> 6.0 \quad \text{for rough boundary.}$$

And if $\frac{k}{\delta'}$ lies between 0.25 to 6.0, the boundary is in transition.

Darcy formula is given by $h_f = \frac{4fLV^2}{d \times 2g}$

where h_f = Head loss due to friction and is known as **Major,** Head loss.

Chezy's formula is given by $V = C\sqrt{m \times i}$

where C = Chezy's constant,

m = Hydraulic mean depth = $\frac{d}{4}$ for circular pipe (running full),

i = Loss of head per unit length = $\frac{h_f}{L}$.

1.11. FLOW THROUGH PIPES

(*i*) **Minor losses**

(*a*) Loss of head due to *sudden expansion* (h_e) is given by $h_e = \frac{(V_1 - V_2)^2}{2g}$.

(*b*) Loss of head due to *sudden contraction* (h_c) is given by

$$h_c = \left(\frac{1}{C_c} - 1\right)^2 \frac{V_2^2}{2g},$$ where C_c = Coefficient of contraction

(*c*) Loss of head at the inlet of a pipe, $h_i = 0.5\,\frac{V^2}{2g}$.

(*d*) Loss of head at the outlet of a pipe, $h_o = \frac{V^2}{2g}$.

(*ii*) **Hydraulic gradient and total energy lines.** The line representing the sum of pressure head and datum head with respect to some reference line is called hydraulic gradient line (H.G.L.) while the line representing the sum of pressure head, datum head and velocity head with respect to some reference line is known as total energy line (T.E.L.).

(*iii*) The **equivalent size** of the pipes connected in series is given by

$$\frac{L}{d^5} = \frac{L_1}{d_1^5} + \frac{L_2}{d_2^5} + \frac{L_3}{d_3^5} + \ldots\ldots$$

where L = Equivalent length of pipe = $L_1 + L_2 + L_3$

d = Equivalent size of the pipe

d_1, d_2, d_3 = Diameters of pipes connected in series.

(*iv*) For **parallel pipes,** the loss of head in each pipe is same and rate of flow in main pipe is equal to the sum of the rate of flow in each pipe, connected in parallel *i.e.,*

(*a*) $Q = Q_1 + Q_2 + \ldots\ldots$ (*b*) $h_{f_1} = h_{f_2} = \ldots\ldots$

(*v*) **Power transmitted** through a pipe is given by,

$$\text{H.P.} = \frac{w \times Q \times [H - h_f]}{75}$$

where H = Total head at the inlet of pipe

h_f = Head lost due to friction

Efficiency of power transmission through pipes, $\eta = \dfrac{[H - h_f]}{H}$

This efficiency will be maximum when $h_f = \dfrac{H}{3}$.

Diameter of **nozzle** for maximum power transmission through nozzle is

$$d = \left(\frac{D^5}{8fL}\right)^{1/4}$$

where d = Diameter of nozzle at outlet, D = Diameter of pipe

L = Length of pipe, f = Coefficient of friction.

(*vi*) **Water hammer.** When a liquid is flowing through a long pipe fitted with a valve at the end of the pipe and the valve is closed suddenly, a pressure wave of high intensity is produced behind the valve. This pressure wave of high intensity is having the effect of hammering action of the walls of the pipe. This phenomenon is known as water hammer. The intensity of pressure rise (p_i) due to water hammer is given by

$$p_i = \frac{\rho L V}{t} \qquad \text{If valve is closed gradually}$$

$$= V\sqrt{K\rho} \qquad \text{If valve is closed suddenly}$$

$$= V\sqrt{\frac{\rho}{\dfrac{1}{K} + \dfrac{D}{Et}}} \qquad \text{If valve is closed suddenly and pipe is elastic.}$$

where L = Length of pipe, V = Velocity of flow

K = Bulk modulus of fluid, D = Diameter of pipe

t = Time for closing valve

E = Modulus of elasticity for pipe material

The valve of closure is said to be gradual if $t > \dfrac{2L}{C}$

The valve closure is said to be sudden if $t < \dfrac{2L}{C}$

where C = Velocity of pressure wave produced due to water hammer = $\sqrt{\dfrac{K}{\rho}}$.

1.12. DIMENSIONAL AND MODEL ANALYSIS

(*i*) **Hydraulic similarities.** There are three types of similarities that must exist between the model and prototype. They are: (*a*) Geometric similarity, (*b*) Kinematic similarity, and (*c*) Dynamic similarity.

Geometric similarity means the similarity of all linear dimensions of model and prototype. **Kinematic** similarity means the similarity of motion between model and prototype. **Dynamic similarity** means the similarity of forces between the model and prototype.

(*ii*) **Dimensionless parameters.** They are five dimensionless parameters, namely: (*a*) Reynold's number, (*b*) Froude number, (*c*) Euler number, (*d*) Weber numbers, and (*e*) Mach number.

(*a*) **Reynold's number** is the ratio of inertia force to viscous force and is given by

$$R_e = \frac{\rho VD}{\mu} \text{ or } \frac{V \times D}{\nu} \text{ for pipe flow.}$$

(*b*) **Froude number** is the ratio of square root of the inertia force to gravity force and is given by

$$F_e = \sqrt{\frac{F_i}{F_g}} = \frac{V}{\sqrt{Lg}}.$$

(*c*) **Euler's number** is the ratio of square root of inertia force to pressure force and is given by

$$E = \sqrt{\frac{F_e}{F_p}} = \frac{V}{\sqrt{p/\rho}}$$

(*d*) **Weber number** is the ratio of square root of inertia force to surface tension force and is given by

$$W = \sqrt{\frac{F_i}{F_s}} = \frac{W}{\sqrt{\sigma/L\rho}}$$

(*e*) **Mach number** (M) is the ratio of square root of inertia force to elastic force and is given by

$$M = \sqrt{\frac{F_i}{F_e}} = \frac{V}{\sqrt{K/\rho}} = \frac{V}{C}.$$

where C = Velocity of sound wave in air.

(*iii*) **Models** are of two types namely (*a*) Undistorted and (*b*) Distorted model. If the models are geometrically similar to its proto-type the models are known as **undistorted model.** And if the models are having different scale ratios for horizontal and vertical dimensions, the models are known as **distorted model.**

1.13. BOUNDARY LAYERS

(*i*) **Boundary layer.** When a solid body is immersed in a flowing fluid, there is a narrow region to the fluid in the neighbourhood of the solid body, where the velocity of the fluid varies from zero to free-stream velocity. This narrow region of fluid is called boundary layer.

(*ii*) Boundary layer may be laminar or turbulent. If the Reynold number of the flow defined as

$$R_e = \frac{U \times x}{\nu}$$

is less than 5×10^5, the boundary layer is called laminary boundary layer. And if R_e is more than 5×10^5, the boundary layer is called turbulent boundary layer.

In the above expression, U = Free stream velocity,
x = Distance from leading edge,
ν = Kinematic viscosity.

(*iii*) **Displacement thickness (δ^*)** is given by $\delta^* = \int_0^\delta \left(1 - \frac{u}{U}\right) dy$

(*iv*) **Momentum thickness (θ)** is given by, $\theta = \int_0^\delta \frac{u}{U}\left(1 - \frac{u}{U}\right) dy$

(*v*) **Energy thickness** $= \int_0^\delta \frac{u}{U}\left[1 - \frac{u^2}{U^2}\right] dy$

(*vi*) **Von Karman momentum** integral equation is given

$$\frac{\tau_0}{\rho U^2} = \frac{\partial \theta}{\partial x}, \text{ where } \theta = \int_0^\delta \frac{u}{U}\left(1 - \frac{u}{U}\right) dy$$

(*vii*) Velocity profile for turbulent boundary is given by $\frac{u}{U} = \left(\frac{y}{\delta}\right)^{1/7}$

where u = Velocity within boundary layer
U = Free stream velocity
δ = Boundary layer thickness.

1.14. DRAG AND LIFT FORCES

(*i*) **Drag and lift forces.** The force, exerted by a flowing fluid on a solid body in the direction of motion, is called drag force while the force perpendicular to the direction of motion is called lift force. Mathematically, they are given as

$$F_D = C_D A \frac{\rho U^2}{2}, F_L = C_L A \frac{\rho U^2}{2}$$

where F_D = Drag force, F_L = Lift force
A = Projected area of body, U = Free-stream velocity
C_D = Coefficient of drag, C_L = Coefficient of lift.

(*ii*) The drag force on a sphere for $R_e \angle 0.2$ is given by

$$F_D = 3\pi\mu DU.$$

1.15. COMPRESSIBLE FLOW

(*i*) Bernoulli's equation for **compressible fluid** is given by

$$\frac{p}{\rho g} \log_e p + \frac{V^2}{2g} + Z = \text{constant for iso-thermal process}$$

$$\frac{\gamma}{\gamma-1}\frac{p}{\rho}+\frac{V^2}{2g}+Z = \text{constant for adiabatic process.}$$

(*ii*) **Velocity of sound wave** is given by

$$C = \sqrt{\frac{dp}{d\rho}} = \sqrt{\frac{k}{\rho}} \qquad \text{In terms of Bulk modulus } k$$

$$= \sqrt{\frac{p}{\rho}} = \sqrt{RT} \qquad \text{For isothermal process}$$

$$= \sqrt{\frac{\gamma p}{\rho}} = \sqrt{\gamma RT} \qquad \text{For adiabatic process}$$

(*iii*) If $M < 1$ flow is called **sub-sonic flow**

$M > 1$ flow is called **super-sonic flow**

$M = 1$ flow is called **sonic flow.**

In sub-sonic flow, the disturbance always moves ahead of the projectile. In sonic flow, the disturbance moves along the projectile while in super-sonic flow that disturbance lags behind the projectile.

(*iv*) **The mach angle** (α) is given by $\sin\alpha = \frac{C}{V} = \frac{1}{M}$

(*v*) **Area velocity relationship** for compressible fluid is given as $\frac{dA}{A} = \frac{dV}{V}\ [M^2 - 1]$

For $M < 1$, the velocity decreases with increase of area.

For $M > 1$, the velocity decreases with the decrease of area.

(*vi*) For maximum flow through an orifice of nozzle fitted to the tank the pressure ratio $\frac{p_2}{p_1}$ $= n = 0.528$.

(*vii*) The **compressibility correction factor** is given by

$$\text{C.C.F.} = \left[1+\frac{M_1^{\,2}}{4}+\frac{2-\gamma}{4}M_1^{\,4}+\ldots\ldots\right].$$

1.16. CHANNEL FLOW

(*i*) **Channel flow.** Channel flow means the flow of a liquid through a passage at atmospheric pressure. The flow of liquid in a channel takes place under the force of gravity which means the flow takes place due to the slope of the bed of the channel. The flow in a channel is classified as:

(*a*) Steady and unsteady flow

(*b*) Uniform and non-uniform flow

(*c*) Laminar and turbulent flow

(*d*) Sub-critical, critical and super-critical flow.

For the *steady flow*, the velocity at a point in a channel with respect to time should be constant. And for *unsteady flow*, the velocity at a point with respect to time should be variable.

The flow in a channel will be *uniform* if the velocity with respect to direction of flow is constant. The velocity will be constant if area of flow is constant. Area of flow will be constant if depth of flow is constant. Hence the flow in a channel will be uniform if the depth of flow is constant. But if the depth of flow is variable then the flow in a channel is known as *non-uniform*.

The non-uniform flow is divided into gradually varied flow (G.V.F.) and rapidly varied flow (R.V.E.). If the depth of flow varies gradually, then the non-uniform flow is known as gradually varied flow. And if the depth of flow varies rapidly, then non-uniform flow is called rapidly varied flow.

The flow in a channel is said to be *laminar* if the Reynolds number (R_e) is less than 500 or 600. Reynolds number is case of open channel flow is given by

$$R_e = \frac{\rho V R}{\mu}$$

where ρ = Density of liquid, V = Velocity of liquid, μ = Viscosity

R = Hydraulic radius or hydraulic mean depth

$$= \frac{A}{P} = \frac{\text{Area of flow}}{\text{Wetted perimeter}}.$$

If the Reynolds number is more than 2000, then the flow is open channel is known as *turbulent*.

(*ii*) The flow in open channel is said to be **sub-critical** if the Froude Number (F_e) is less than 1.0.

The Froude number is given by, $F_e = \dfrac{V}{\sqrt{gD}}$

where V = Velocity of flow

D = Hydraulic depth of channel

$$= \frac{A}{T}$$

where A = Wetted area, T = Top width of the channel.

(*iii*) The flow in a channel is known as **critical flow** if the Froude number is equal to 1.0. But if the Froude number is more than 1.0, then flow in the channel is known as **super-critical flow.**

(*iv*) **The velocity through a channel** is given by

(*a*) Chezy's Formula

(*b*) Manning's Formula.

Chezy's Formula

The velocity by Chezy's formula is given by, $V = C\sqrt{mi}$

where C = Chezy's constant and depends upon the surface of the channel

m = Hydraulic mean depth

$$= \frac{A}{P} = \frac{\text{Area of flow}}{\text{Wetted perimeter}}$$

i = Slope of the bed of the channel.

Manning's Formula

The velocity through a channel according to Manning's Formula is given by, $V = \frac{1}{N} m^{2/3} i^{1/2}$

where N = Manning's constant and depends upon the surface of the channel

m = Hydraulic mean depth

i = Slope of the bed of the channel.

(v) **The relation between Chezy's constant** (C) and Manning's constant (N) is given by

$$C = \frac{1}{N} m^{1/6}.$$

The Chezy's constant is not a dimensionless coefficients. The dimension of C is given by $L^{1/2} T^{-1}$.

(vi) **Different types of channels**

The different types of channels are

(a) Rectangular channel

(b) Trapezoidal channel

(c) Circular channel

(d) Triangular channel.

(vii) **Values of A (area of flow) and P (Wetted perimeter) for different channels.**

(a) *Rectangular channel* [See Fig. 1.1 (a)]

Let b = Width of channel, d = Depth of flow

then $A = b \times d$

$P = b + 2d$

(b) *Trapezoidal channel* [See Fig. 1.1 (b)]

Let b = Width of flow, d = Depth of flow

n = Side slope which is expressed as 1 vertical to n horizontal

then $A = (b + nd) \times d$

$P = b + 2d\sqrt{n^2 + 1}$

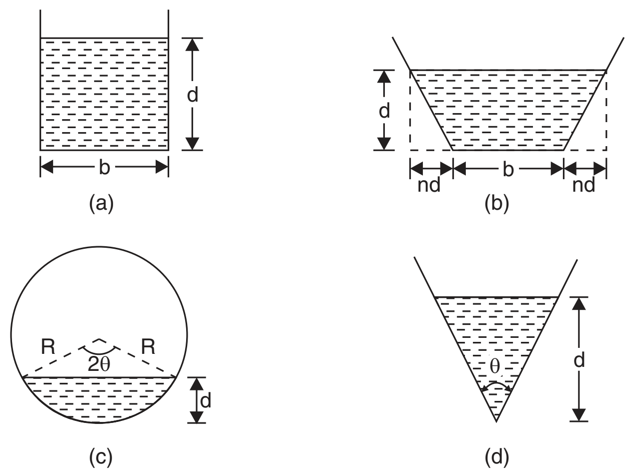

FIGURE 1.1

(*c*) *Circular channel* [Refer Fig. 1.1 (*c*)]

Let R = Radius of the channel, d = Depth of water in circular channel

2θ = Angle subtended at the centre of the channel by the free surface of water

then $$A = R^2\left(\theta - \frac{\sin 2\theta}{2}\right)$$

and $$P = 2R\theta$$

(*d*) *Triangular channel* [Refer Fig. 1.1 (*d*)]

Let d = Depth of flow, θ = Angle of triangular channel

then $$A = d^2 \tan\left(\frac{\theta}{2}\right)$$

and $$P = 2d \sec\left(\frac{\theta}{2}\right)$$

(*viii*) **Most efficient section of a channel.** The section of a channel is said to be most efficient if the discharge through the channel is maximum for a given area of flow, given surface resistance and given slope of the bed of the channel. The most efficient section is also known as most economical section of the channel.

(*a*) **A rectangular channel** will be most efficient, if

1. Depth of flow = Half of width of channel *i.e.*, $d = \frac{1}{2} \times b$

2. Hydraulic mean depth = Half of depth of flow *i.e.*, $m = \frac{1}{2} \times d$.

(*b*) **A trapezoidal channel** will be most efficient if

1. Length of sloping side = Half of top width

i.e., $$d\sqrt{n^2+1} = \frac{b+2nd}{2}.$$

2. Hydraulic mean depth = Half of depth of flow

i.e., $$m = \frac{d}{2}$$

3. The three sides of the trapezoidal section are the tangential to the semi-circle described on the water-line.

(*c*) **A circular channel** will be most efficient for maximum velocity, if

1. Depth of flow = 0.81 times the dia. of the circular channel.

2. Hydraulic mean depth = 0.3 times the dia. of the channel.

The flow through a circular channel will be maximum if

1. Depth of flow = 0.95 D.

(*ix*) **Specific energy.** The total energy of a flowing liquid in a channel with respect to the bed of channel is known as specific energy. It is given by

$$E = y + \frac{V^2}{2g}$$

where y = Depth of flow, V = Velocity of flow.

The curve which represents the variation of specific energy with the depth of flow is known as specific energy curve.

(*x*) **Critical depth.** The depth of flow of water at which the specific energy is minimum, is known as critical depth. For a rectangular channel, the critical depth (y_c) is given by

$$y_c = \left(\frac{q^2}{g}\right)^{1/3}$$

where q = Rate of flow per unit width of channel.

The velocity of flow, corresponding to critical depth is known as critical velocity.

(*xi*) **Hydraulic jump.** The sudden rise of water level which takes place due to the transformation of super-critical flow to the subcritical flow, is known as hydraulic jump. And when hydraulic jump takes place, a loss of energy due to eddy formation and turbulence occurs. The depth of water after hydraulic jump is given by

$$y_2 = -\frac{y_1}{2} + \sqrt{\left(\frac{y_1}{2}\right)^2 + \frac{2q^2}{gy_1}}$$

where y_2 = Depth of water after hydraulic jump
y_1 = Depth of water before hydraulic jump
q = Discharge per unit width.

The loss of energy (h_L) during hydraulic jump is given by, $h_L = \dfrac{(y_2 - y_1)^3}{4y_1y_2}$.

1.17. HYDRAULIC TURBINES

(*i*) **Hydraulic turbines.** The hydraulic machines which convert the hydraulic energy into the mechanical energy, are called turbines.

(*ii*) **The force exerted by a jet of water** on a stationary plate in the direction of jet is given by

$$\begin{aligned} F_x &= \rho A V^2 && \text{for a vertical plate} \\ &= \rho A V^2 \sin^2 \theta && \text{for an inclined plate} \\ &= \rho A V^2 [1 + \cos \theta] && \text{for a curved plate} \end{aligned}$$

(*iii*) For force exerted by the jet of water having velocity V on a plate moving with a velocity u is given by

$$\begin{aligned} F_x &= \rho A (V - u)^2 && \text{for a vertical plate} \\ &= \rho A (V - u)^2 \sin^2 \theta && \text{for an inclined plate} \\ &= \rho A (V - u)^2 [1 + \cos \theta] && \text{for a curved plate} \end{aligned}$$

(*iv*) **Efficiency of a series** of vanes is given as $\eta = \dfrac{2u(V-u)}{V^2}$

Efficiency will be maximum, when $u = \dfrac{V}{2}$. Maximum efficiency = 50%.

(*v*) **For a series of curved** radial vanes, the work done per unit weight per second

$$= \frac{1}{g}[V_{w_1 u_1} \pm V_{w_2 u_2}]$$

where V_{w_1}, V_{w_2} = Velocity of whirl at inlet and outlet.

(*vi*) **The net head** on the turbine is given by $H = H_g - h_f$

where H = Net head, H_g = Gross head, h_f = Head loss due to friction.

(*vii*) **The efficiencies** of a turbine are: (*a*) Hydraulic efficiency (η_h), (*b*) Mechanical efficiency (η_m), and (*c*) Overall efficiency (η_0).

(*a*) Hydraulic efficiency (η_h) is given by

$$\eta_h = \frac{\text{Power given by water of the runner}}{\text{Power supplied at inlet}} = \frac{\text{R.P.}}{\text{W.P.}}$$

$$= \rho \times Q \frac{[V_{w_1 u_1} \pm V_{w_2 u_2}]}{75} \bigg/ \left(\frac{\rho \times Q \times g \times H}{75} \right) = \frac{(V_{w_1 u_1} \pm V_{w_2 u_2})}{gH}$$

(*b*) Mechanical efficiency (η_m) is given by $\eta_n = \dfrac{\text{S.P.}}{\text{R.P.}}$

(*c*) Overall efficiency is given by $\eta_0 = \dfrac{\text{S.P.}}{\text{W.P.}} = \eta_n \times \eta_h$.

(*viii*) **Impulse turbine.** If at the inlet of a turbine, total energy is only kinetic energy, the turbine is called impulse turbine. Pelton wheel is an impulse turbine.

(*ix*) **Reaction turbine.** If at the inlet of a turbine, the total energy is kinetic energy as well as pressure energy, the turbine is called reaction turbine. Francis and Kaplan turbines are reaction turbines.

(*x*) **Jet ratio (m)** is defined as the ratio of diameter (D) of Pelton wheel to the diameter (d) of the jet

or Jet ratio $m = \frac{D}{d}$

(*xi*) Pelton wheel is a tangential flow impulse turbine, Francis is an inward flow reaction turbine and Kaplan is an axial flow reaction turbine. The rate of flow (Q) through the turbine is given by

$$Q = \frac{\pi}{4} d^2 \times \sqrt{2gH} \qquad \text{for Pelton Turbine}$$

$$= \pi D_1 B_1 V_{f1} \qquad \text{for Francis Turbine}$$

$$= \frac{\pi}{4} [D_1^2 - D_0^2] \times V_{f1} \qquad \text{for Kaplan Turbine}$$

where H = Net head, V_{f1} = Velocity of flow at inlet

D_0 = Dia. of Kaplan turbine, D_1 = Hub diameter.

(*xii*) **Draft-tube.** It is a pipe of gradually increasing area used for discharging water from the exit of a reaction turbine.

(*xiii*) **Specific speed** of a turbine is defined as the speed at which a turbine runs when it is working under a unit head and develops unit (*i.e.*, 1 kW power). It is given by $N_s = \frac{N\sqrt{P}}{H^{5/4}}$, where P = Shaft power, H = Net head on turbine.

(*a*) **Unit speed (N_u)** is the speed of a turbine, when the head on the turbine is one metre. It is given by $N_u = \frac{N}{\sqrt{H}}$.

(*b*) **Unit discharge (Q_u)** is the discharge through a turbine when the head (H) on the turbine is unity. It is given by $Q_u = \frac{Q}{\sqrt{H}}$.

(*c*) **Unit power (P_u)** is the power developed by a turbine when the head on the turbine is unity. It is given by $P_u = \frac{P}{H^{3/2}}$.

(*xiv*) **Characteristic curves.** The following three are the important characteristic curves of a turbine:

(*a*) Main characteristic curve, (*b*) Operating characteristic curves, and (*c*) Constant efficiency curves.

(*xv*) **Governing of a turbine** is defined as the operation by which the speed of the turbine is kept constant under all conditions of working.

1.18. CENTRIFUGAL PUMPS

(*i*) The hydraulic machine which converts the mechanical energy into pressure energy by means of centrifugal force is called **centrifugal pump**. The work done by impeller on water per second per unit weight of water

$$= \frac{1}{g} V_{w_2} \times u_2$$

where V_{w_2} = Velocity of whirl at outlet, u_2 = Tangential velocity of wheel at outlet.

(*ii*) **The manometric head (H_m)** is the head against which a centrifugal pump has to work. It is given as

$$H_m = \frac{V_{w_2} \times u_2}{g} - \text{Loss of head in impeller and casing}$$

$$= \frac{V_{w_2} \times u_2}{g} \quad \text{If losses in pump are zero}$$

$$= \text{Total head at outlet} - \text{Total head at inlet}$$

$$= \left(\frac{p_0}{w} + \frac{V_0^2}{2g} + Z_0\right) - \left(\frac{p_i}{w} + \frac{V_i^2}{w} + Z_i\right)$$

$$= h_s + h_d + h_{fd} + \frac{V_d^2}{2g}.$$

(*iii*) **The efficiencies of a pump** are: (*a*) Manometric efficiency (η_{man}), (*b*) Mechanical (η_m), and (*c*) overall efficiency (η_0). They are expressed as

$$\eta_{man} = \frac{gH_m}{V_{w_2} \times u_2}$$

$$\eta_m = \frac{\text{I.P.}}{\text{S.P.}} = \left(\rho \times Q \times \frac{V_{w_2} \times u_2}{75}\right) \Big/ \text{S.P.}$$

$$\eta_0 = \eta_{man} \times \eta_m$$

where I.P. = Power on impeller, S.P. = Shaft power.

(*iv*) **Multistage centrifugal** pumps are used to produce a high head or to discharge a large quantity of water. The produce a high head, the impellers are connected in series while to discharge a large quantity of liquid, the impellers are connected in parallel.

(*v*) **Specific speed of a pump** is defined as the speed at which a pump runs when the head developed is one metre and discharge is one cubic metre. It is given as $N_s = \frac{N\sqrt{Q}}{H_m^{3/4}}$, where H_m = Manometric head.

(*vi*) **Cavitation** is defined as the phenomenon of formation of vapour bubbles and sudden collapsing of the vapour bubbles.

1.19. RECIPROCATING PUMP

(*i*) The discharge through a reciprocating pump per second is given by

$$Q = \frac{ALN}{60} \qquad \text{for a single acting}$$

$$= \frac{2ALN}{60} \qquad \text{for a double acting}$$

where A = Area of position, L = Length of stroke.

(*ii*) **Work done by reciprocating** pump per second

$$= \frac{wALN}{60}(h_s + h_d) \qquad \text{for a single acting}$$

$$= \frac{2wALN}{60}(h_s + h_d) \qquad \text{for a double acting.}$$

(*iii*) **The pressure head due** to acceleration is given by

$$h_a = \frac{l}{g} \times \frac{A}{a} \times \omega^2 r \cos\theta$$

where l = Length of suction or delivery pipe

a = Area of suction or delivery pipe

r = Radius of crank = $L/2$

ω = Angular speed = $\frac{2\pi N}{60}$.

(*iv*) **Indicator diagram** is a graph between the pressure head in the cylinder and the distance travelled by the piston from inner dead centre for one complete revolution of the crank. Work done by the pump is proportional to the area of indicator diagram. Area of ideal indicator diagram is the same as the area of indicator diagram due to acceleration in suction and delivery pipes.

(*v*) **Air vessel** is a device used: (*a*) to obtain a continuous supply of water at uniform rate, (*b*) to save a considerable amount of work, and (*c*) to run the reciprocating pump at a high speed without separation.

The mean velocity ($\overline{V}$) for a single acting pump is $\overline{V} = \frac{A}{a}\frac{\omega r}{\pi}$.

The work saved by fitting air vessels in a single acting reciprocating pump is 84.8% while in a double acting, the work saved is 39.2%.

1.20. MISCELLANEOUS HYDRAULIC DEVICES

The miscellaneous hydraulic devices are hydraulic press, hydraulic accumulator, hydraulic intensifier, hydraulic ram, hydraulic lift, hydraulic cranes, hydraulic coupling and hydraulic torque converter.

(*i*) **Hydraulic press** is a device used for lifting heavy weights by the application of a much smaller force. **Hydraulic accumulator** is a device used for storing the energy of a fluid in the form of pressure energy. **Hydraulic intensifier** is a device used for increasing the pressure intensity of a liquid. **Hydraulic ram** is a pump which raises water without any external power (such as electricity) for its operation. **Hydraulic lift** is a device used for carrying persons or goods from one floor to another floor in a multi-storeyed building. **Hydraulic crane** is a device used for raising or transferring heavy weights. **Hydraulic coupling** is a device, in which power is transmitted from driving shaft to driven shaft without any change of torque while **torque convertor** is a device in which arrangement is provided for getting increased or decreased torque at the driven shaft.

(*ii*) **Capacity of a hydraulic accumulator** = $p \times A \times L$

where p = Liquid pressure supplied by pump

A = Area of sliding ram, and L = Stroke of the ram.

(*iii*) **Hydraulic ram** has two efficiencies namely D' Aubuisson's efficiency and Rankine efficiency. They are given by

$$\text{D' Aubuisson's, } \eta = \frac{wH}{Wh} \text{ and Rankine's, } \eta = \frac{w(H-h)}{(W-w)\times h}$$

where w = Weight of water raised/sec, W = Weight of water supplied raised/sec,

h = Height of water in supply tank, and H = Height of water raised.

II. OBJECTIVE TYPE QUESTIONS

Tick mark the most appropriate statement of the multiple choice answers:

Fluid Properties

1. An ideal fluid is defined as the fluid which

(*a*) is compressible ☐

(*b*) is incompressible ☐

(*c*) is incompressible and non-viscous (inviscid) ☐

(*d*) has negligible surface tension. ☐

2. Newton's law of viscosity states that

(*a*) shear stress is directly proportional to the velocity ☐

(*b*) shear stress is directly proportional to velocity gradient ☐

(*c*) shear stress is directly proportional to shear strain ☐

(*d*) shear stress is directly proportional to the viscosity. ☐

3. A Newtonian fluid is defined as the fluid which

(*a*) is incompressible and non-viscous ☐ (*b*) obeys Newton's law of viscosity ☐

(*c*) is highly viscous ☐ (*d*) is compressible and non-viscous. ☐

4. Kinematic viscosity is defined as equal to

(*a*) dynamic viscosity × density ☐ (*b*) dynamic viscosity/density ☐

(*c*) dynamic viscosity × pressure ☐ (*d*) pressure × density. ☐

5. Dynamic viscosity (μ) has the dimensions as

(*a*) MLT^{-2} ☐ (*b*) $ML^{-1}T^{-1}$ ☐

(*c*) $ML^{-1}T^{-2}$ ☐ (*d*) $M^{-1}L^{-1}T^{-1}$. ☐

6. Poise is the unit of

(*a*) mass density ☐ (*b*) kinematic viscosity ☐

(*c*) viscosity ☐ (*d*) velocity gradient. ☐

7. The increase of temperature

(*a*) increases the viscosity of a liquid ☐ (*b*) decreases the viscosity of a liquid ☐

(*c*) decreases the viscosity of a gas ☐ (*d*) increases the viscosity of a gas. ☐

8. Stoke is the unit of

(*a*) surface tension ☐ (*b*) viscosity ☐

(*c*) kinematic viscosity ☐ (*d*) none of the above. ☐

9. The multiplying factor for converting one poise into MKS unit of dynamic viscosity is

(*a*) 9.81 ☐ (*b*) 98.1 ☐

(*c*) 981 ☐ (*d*) 0.981. ☐

10. Surface tension has the units of

(*a*) force per unit area ☐ (*b*) force per unit length ☐

(*c*) force per unit volume ☐ (*d*) none of the above. ☐

11. The gases are considered incompressible when Mach number is

(*a*) equal to 1.0 ☐ (*b*) equal to 0.50 ☐

(*c*) more than 0.3 ☐ (*d*) less than 0.2. ☐

12. Kinematic viscosity (ν) is equal to

(*a*) $\mu \times \rho$ ☐ (*b*) $\frac{\mu}{\rho}$ ☐

(*c*) $\frac{\rho}{\mu}$ ☐ (*d*) none of the above. ☐

13. Compressibility is equal to

(*a*) $\dfrac{-\left(\dfrac{dV}{V}\right)}{dp}$ ☐ (*b*) $\dfrac{dp}{-\left(\dfrac{dV}{V}\right)}$ ☐

(*c*) $\dfrac{dp}{d\rho}$ ☐ (*d*) $\sqrt{\dfrac{dp}{d\rho}}$. ☐

14. Hydrostatic law of pressure is given as

(*a*) $\frac{\partial p}{\partial z} = \rho g$ ☐ (*b*) $\frac{\partial p}{\partial z} = 0$ ☐

(*c*) $\frac{\partial p}{\partial z} = z$ ☐ (*d*) $\frac{\partial p}{\partial z}$ = constant. ☐

15. Four curves are shown in Fig. 1.2 with velocity gradient $\left(\frac{\partial u}{\partial y}\right)$ along *x*-axis and viscous shear stress (τ) along *y*-axis. Curve *A* corresponds to

(*a*) ideal fluid ☐
(*b*) newtonian fluid ☐
(*c*) non-newtonian fluid ☐
(*d*) ideal solid. ☐

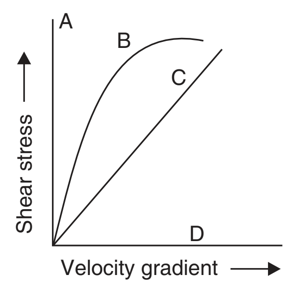

FIGURE 1.2

16. Curve *B* in Fig. 1.2 corresponds to

(*a*) ideal fluid ☐
(*b*) newtonian fluid ☐
(*c*) non-newtonian fluid ☐
(*d*) idealsolid. ☐

17. Curve *C* in Fig. 1.2 corresponds to

(*a*) ideal fluid ☐ (*b*) newtonian fluid ☐
(*c*) non-newtonian fluid ☐ (*d*) ideal solid. ☐

18. Curve *D* in Fig. 1.2 corresponds to

(*a*) ideal fluid ☐ (*b*) newtonian fluid ☐
(*c*) non-newtonian fluid ☐ (*d*) ideal solid. ☐

19. The relation between surface tension (σ) and difference of pressure (Δp) between the inside and outside of a liquid droplet is given as

(*a*) $\Delta p = \frac{\sigma}{4d}$ ☐ (*b*) $\Delta p = \frac{\sigma}{2d}$ ☐

(*c*) $\Delta p = \frac{4\sigma}{d}$ ☐ (*d*) $\Delta p = \frac{\sigma}{d}$. ☐

20. For a soap bubble, the surface tension (σ) and difference of pressure (Δp) are related as

(*a*) $\Delta p = \frac{\sigma}{4d}$ ☐ (*b*) $\Delta p = \frac{\sigma}{2d}$ ☐

(*c*) $\Delta p = \frac{4\sigma}{d}$ ☐ (*d*) $\Delta p = \frac{8\sigma}{d}$. ☐

21. For a liquid jet, the surface tension (σ) and difference of pressure (Δp) are related as

(*a*) $\Delta p = \frac{\sigma}{4d}$ ☐ (*b*) $\Delta p = \frac{\sigma}{2d}$ ☐

(*c*) $\Delta p = \frac{4\sigma}{d}$ ☐ (*d*) $\Delta p = \frac{2\sigma}{d}$. ☐

22. The capillary rise or fall of a liquid is given by

(*a*) $h = \dfrac{\sigma \cos \theta}{4\rho g d}$ ☐ (*b*) $h = \dfrac{4\sigma \cos \theta}{\rho g d}$ ☐

(*c*) $h = \dfrac{8\sigma \cos \theta}{\rho g d}$ ☐ (*d*) none of the above. ☐

Pressure and Hydrostatic Forces on Surfaces

23. Pascal's law states that pressure at a point is equal in all direction in

(*a*) a liquid at rest ☐ (*b*) a fluid at rest ☐

(*c*) a laminar flow ☐ (*d*) a turbulent flow. ☐

24. The hydrostatic law states that rate of increase of pressure in a vertical direction

(*a*) is equal to density of the fluid ☐ (*b*) is equal to specific weight of the fluid ☐

(*c*) is equal to weight of the fluid ☐ (*d*) none of the above. ☐

25. Fluid statics deals with the following forces

(*a*) viscous and gravity forces ☐ (*b*) viscous and gravity forces ☐

(*c*) gravity and pressure forces ☐ (*d*) surface tension and gravity forces. ☐

26. Gauge pressure at a point is equal to

(*a*) absolute pressure plus atmospheric pressure ☐

(*b*) absolute pressure minus atmospheric pressure ☐

(*c*) vacuum pressure plus absolute pressure ☐

(*d*) none of the above. ☐

27. Atmospheric pressure head in terms of water column is

(*a*) 7.5 m ☐ (*b*) 8.5 m ☐

(*c*) 9.81 m ☐ (*d*) 10.30 m. ☐

28. The hydrostatic pressure on a plane surface is equal to

(*a*) $\rho g A\bar{h}$ ☐ (*b*) $\rho g A\bar{h} \sin^2 \theta$ ☐

(*c*) $\frac{1}{4}\rho g A\bar{h}$ ☐ (*d*) $\rho g A\bar{h} \sin \theta$ ☐

where A = Area of plane surface and $\bar{h}$ = Depth of centroid of the plane area below the liquid free surface.

29. Centre of pressure of a plane surface immersed in a liquid is

(*a*) above the centre of gravity of the plane surface ☐

(*b*) at the centre of gravity of the plane surface ☐

(*c*) below the centre of gravity of the plane surface ☐

(*d*) none of the above. ☐

30. The resultant hydrostatic force acts through a point known as

(*a*) centre of gravity ☐ (*b*) centre of buoyancy ☐

(*c*) centre of pressure ☐ (*d*) none of the above. ☐

31. For submerged curved surface, the vertical component of the hydrostatic force is

(*a*) mass of the liquid supported by the curved surface ☐

(*b*) weight of the liquid supported by the curved surface ☐

(*c*) the force of the projected area of the curved surface on vertical plane ☐

(*d*) none of the above. ☐

32. Manometer is a device used for measuring

(*a*) velocity at a point in a fluid ☐ (*b*) pressure at a point in a fluid ☐

(*c*) discharge of a fluid ☐ (*d*) none of the above. ☐

33. Differential manometers are used for measuring

(*a*) velocity at a point in a fluid ☐

(*b*) pressure at a point in a fluid ☐

(*c*) difference of pressure between two points ☐

(*d*) none of the above. ☐

34. The pressure at a height Z in a static compressible fluid undergoing isothermal compression is given as

(*a*) $p = p_0\, e^{-\frac{gR}{ZT}}$ ☐ (*b*) $p = p_0\, e^{-\frac{gT}{RZ}}$ ☐

(*c*) $p = p_0 e^{-\frac{RT}{gZ}}$ ☐ (*d*) $p = p_0 e^{-\frac{gZ}{RT}}$ ☐

where p_0 = Pressure at ground level, R = Gas constant, T = Absolute temperature.

35. The pressure at a height Z in a static compressible fluid undergoing adiabatic compression is given by

(*a*) $p = p_0 \left[1 - \frac{\gamma-1}{\gamma}\frac{RT_0}{gZ}\right]^{\frac{\gamma}{\gamma-1}}$ ☐ (*b*) $p = p_0 \left[1 - \frac{\gamma}{\gamma-1}\frac{RT_0}{gZ}\right]^{\frac{\gamma}{\gamma-1}}$ ☐

(*c*) $p = p_0 \left[1 - \frac{\gamma-1}{\gamma}\frac{gZ}{RT_0}\right]^{\frac{\gamma}{\gamma-1}}$ ☐ (*d*) none of the above. ☐

36. The temperature at a height Z in a static compressible fluid undergoing adiabatic compression is given as

(*a*) $T = T_0 \left[1 - \frac{\gamma-1}{\gamma}\frac{RT_0}{gZ}\right]$ ☐ (*b*) $T = T_0 \left[1 - \frac{\gamma-1}{\gamma}\frac{gZ}{RT_0}\right]$ ☐

(*c*) $T = T_0 \left[1 - \frac{\gamma}{\gamma-1}\frac{RT_0}{gZ}\right]$ ☐ (*d*) none of the above. ☐

37. Temperature lapse-rate is given by

(*a*) $L = -\frac{R}{g}\left[\frac{\gamma-1}{\gamma}\right]$ ☐ (*b*) $L = -\frac{R}{g}\left[\frac{\gamma}{\gamma-1}\right]$ ☐

(*c*) $L = -\frac{g}{R}\left[\frac{\gamma-1}{\gamma}\right]$ ☐ (*d*) none of the above. ☐

38. When the fluid is at rest, the shear stress is

(*a*) maximum ☐ (*b*) zero ☐

(*c*) unpredictable ☐ (*d*) none of the above. ☐

39. The depth of centre of pressure of an inclined immersed surface from free surface of liquid is equal to

(*a*) $\frac{I_G}{A\bar{h}} + \bar{h}$ ☐ (*b*) $\frac{I_G A \sin^2 \theta}{\bar{h}} + \bar{h}$ ☐

(*c*) $\frac{I_G \sin^2 \theta}{A\bar{h}} + \bar{h}$ ☐ (*d*) $\frac{I_G \bar{h}}{A \sin^2 \theta} + \bar{h}$. ☐

40. The depth of centre of pressure of a vertical immersed surface from free surface of liquid is equal to

(*a*) $\frac{I_G}{A\bar{h}} + \bar{h}$ ☐ (*b*) $\frac{I_G A}{\bar{h}} + \bar{h}$ ☐

(*c*) $\frac{I_G \bar{h}}{A} + \bar{h}$ ☐ (*d*) $\frac{A\bar{h}}{I_G} + \bar{h}$. ☐

41. The centre of pressure for a plane vertical surface lies at a depth of

(*a*) half the height of the immersed surface ☐

(*b*) one-third the height of the immersed surface ☐

(*c*) two-third the height of the immersed surface ☐

(*d*) none of the above. ☐

42. The inlet length of a venturimeter

(*a*) is equal to the outlet length ☐ (*b*) is more than the outlet length ☐

(*c*) is less than the outlet length ☐ (*d*) none of the above. ☐

43. Flow of a fluid in a pipe takes place from

(*a*) higher level to lower level ☐ (*b*) higher pressure to lower pressure ☐

(*c*) higher energy to lower energy ☐ (*d*) none of the above. ☐

Buoyancy and Floatation

44. For a floating body, the buoyant force passes through the

(*a*) centre of gravity of the body ☐

(*b*) centre of gravity of the submerged part of the body ☐

(*c*) metacentre of the body ☐

(*d*) centroid of the liquid displaced by the body. ☐

45. The condition of stable equilibrium for a floating body is

(*a*) the metacentre M coincides with the centre of gravity G ☐

(*b*) the metacentre M is below centre of gravity G ☐

(*c*) the metacentre M is above centre of gravity G ☐

(*d*) the centre of buoyancy B is above centre of gravity G. ☐

46. A submerged body will be in stable equilibrium if the

(*a*) centre of buoyancy *B* is below the centre of gravity *G* ☐
(*b*) centre of buoyancy *B* coincides with *G* ☐
(*c*) centre of buoyancy *B* is above the metacentre *M* ☐
(*d*) centre of buoyancy *B* is above *G*. ☐

47. The metacentric height of a floating body is

(*a*) the distance between metacentre and centre of buoyancy ☐
(*b*) the distance between the centre of buoyancy and centre of gravity ☐
(*c*) the distance between metacentre and centre of gravity ☐
(*d*) none of the above. ☐

48. The point, through which the buoyant force is acting, is called

(*a*) centre of pressure ☐ (*b*) centre of gravity ☐
(*c*) centre of buoyancy ☐ (*d*) none of the above. ☐

49. The point, through which the weight is acting, is called

(*a*) centre of pressure ☐ (*b*) centre of gravity ☐
(*c*) centre of buoyancy ☐ (*d*) none of the above. ☐

50. The point, about which a floating body, starts oscillating when the body is tilted is called

(*a*) centre of pressure ☐ (*b*) centre of buoyancy ☐
(*c*) centre of gravity ☐ (*d*) metacentre. ☐

51. The metacentric height (*GM*) is given by

(*a*) $GM = BG - \frac{I}{V}$ ☐ (*b*) $GM = \frac{V}{I} - BG$ ☐
(*c*) $GM = \frac{I}{V} - BG$ ☐ (*d*) none of the above. ☐

52. For floating body, if the metacentre is above the centre of gravity, the equilibrium is called

(*a*) stable ☐ (*b*) unstable ☐
(*c*) neutral ☐ (*d*) none of the above. ☐

53. For a floating body, if the metacentre is below the centre of gravity, the equilibrium is called

(*a*) stable ☐ (*b*) unstable ☐
(*c*) neutral ☐ (*d*) none of the above. ☐

54. For a floating body, if the metacentre coincides with the centre of gravity, the equilibrium is called

(*a*) stable ☐ (*b*) unstable ☐
(*c*) neutral ☐ (*d*) none of the above. ☐

55. For a floating body, if centre of buoyancy is above the centre of gravity, the equilibrium is called

(*a*) stable ☐ (*b*) unstable ☐
(*c*) neutral ☐ (*d*) none of the above. ☐

56. For a submerged body, if the centre of buoyancy is above the centre of gravity, the equilibrium is called

(*a*) stable ☐ (*b*) unstable ☐
(*c*) neutral ☐ (*d*) none of the above. ☐

57. For a submerged body, if the centre of buoyancy is below the centre of gravity, the equilibrium is called

(*a*) stable ☐ (*b*) unstable ☐
(*c*) neutral ☐ (*d*) none of the above. ☐

58. For a submerged body, if the centre of buoyancy coincides with the centre of gravity, the equilibrium is called

(*a*) stable ☐ (*b*) unstable ☐
(*c*) neutral ☐ (*d*) none of the above. ☐

59. For a submerged body, if the metacentre is below the centre of gravity, the equilibrium is called

(*a*) stable ☐ (*b*) unstable ☐
(*c*) neutral ☐ (*d*) none of the above. ☐

60. The metacentric height (GM) experimentally is given as

(*a*) $GM = \dfrac{W \tan \theta}{wx}$ ☐ (*b*) $GM = \dfrac{w \tan \theta}{W \times x}$ ☐

(*c*) $GM = \dfrac{wx}{W \tan \theta}$ ☐ (*d*) $GM = \dfrac{Wx}{w \tan \theta}$ ☐

where w = Movable weight, W = Weight of floating body including w, θ = Angle of tilt.

61. The time period of oscillation of a floating body is given by

(*a*) $T = 2\pi \sqrt{\dfrac{GM \times g}{k^2}}$ ☐ (*b*) $T = 2\pi \sqrt{\dfrac{k^2}{GM \times g}}$ ☐

(*c*) $T = 2\pi \sqrt{\dfrac{GM}{gk^2}}$ ☐ (*d*) $T = 2\pi \sqrt{\dfrac{gk^2}{GM}}$ ☐

where k = Radius of gyration, GM = Metacentric height and T = Time period.

Kinematics and Dynamics of Flow

62. The necessary condition for the flow to be steady is that

(*a*) the velocity does not change from place to place ☐
(*b*) the velocity is constant at a point with respect to time ☐
(*c*) the velocity changes at a point with respect to time ☐
(*d*) none of the above. ☐

63. The necessary condition for the flow to be uniform is that

(*a*) the velocity is constant at a point with respect to time ☐
(*b*) the velocity is constant in the flow field with respect to space ☐

(*c*) the velocity changes at a point with respect to time ☐
(*d*) none of the above. ☐

64. The flow in the pipe is laminar if

(*a*) Reynold number is equal to 2500 ☐ (*b*) Reynold number is equal to 4000 ☐
(*c*) Reynold number is more than 2500 ☐ (*d*) None of the above. ☐

65. A stream line is a line

(*a*) which is along the path of a particle ☐
(*b*) which is always parallel to the main direction of flow ☐
(*c*) across which there is no flow ☐
(*d*) on which tangent drawn at any point gives the direction of velocity. ☐

66. Continuity equation can take the form

(*a*) $A_1V_1 = A_2V_2$ ☐ (*b*) $\rho_1A_1 = \rho_2A_2$ ☐
(*c*) $\rho_1A_1V_1 = \rho_2A_2V_2$ ☐ (*d*) $p_1A_1V_1 = p_2A_2V_2$. ☐

67. Pitot-tube is used for measurement of

(*a*) pressure ☐ (*b*) flow ☐
(*c*) velocity at a point ☐ (*d*) discharge. ☐

68. Bernoulli's theorem deals with the law of conservation of

(*a*) mass ☐ (*b*) momentum ☐
(*c*) energy ☐ (*d*) none of the above. ☐

69. Continuity equation deals with the law of conservation of

(*a*) mass ☐ (*b*) momentum ☐
(*c*) energy ☐ (*d*) none of the above. ☐

70. Irrotational flow means

(*a*) the fluid does not rotate while moving ☐
(*b*) the fluid moves in straight lines ☐
(*c*) the net rotation of fluid-particles about their mass centres is zero ☐
(*d*) none of the above. ☐

71. The velocity components in x and y-directions in terms of velocity potential (ϕ) are

(*a*) $u = -\frac{\partial\phi}{\partial x}, v = \frac{\partial\phi}{\partial y}$ ☐ (*b*) $u = \frac{\partial\phi}{\partial y}, v = \frac{\partial\phi}{\partial x}$ ☐

(*c*) $u = -\frac{\partial\phi}{\partial y}, v = -\frac{\partial\phi}{\partial x}$ ☐ (*d*) $u = -\frac{\partial\phi}{\partial x}, v = -\frac{\partial\phi}{\partial y}$. ☐

72. The velocity components in x and y-directions in terms of stream function (ψ) are

(*a*) $u = \frac{\partial\psi}{\partial x}, v = \frac{\partial\psi}{\partial y}$ ☐ (*b*) $u = -\frac{\partial\psi}{\partial x}, v = \frac{\partial\psi}{\partial y}$ ☐

(*c*) $u = \frac{\partial\psi}{\partial y}, v = \frac{\partial\psi}{\partial x}$ ☐ (*d*) $u = -\frac{\partial\psi}{\partial y}, v = \frac{\partial\psi}{\partial x}$. ☐

73. The relation between tangential velocity (v) and radius (r) is given by

(a) $V \times r$ = constant for forced vortex ☐ (b) V/r = constant for forced vortex ☐

(c) $V \times r$ = constant for free vortex ☐ (d) V/r = constant for free vortex. ☐

74. The pressure variation along the radial direction for vortex flow along a horizontal plane is given as

(a) $\dfrac{\partial p}{\partial r} = -\rho \dfrac{V^2}{r}$ ☐ (b) $\dfrac{\partial p}{\partial r} = \rho \dfrac{V}{r^2}$ ☐

(c) $\dfrac{\partial p}{\partial r} = \rho \dfrac{V^2}{r}$ ☐ (d) none of the above. ☐

75. For a forced vortex flow the height of paraboloid formed is equal to

(a) $\dfrac{p}{w} + \dfrac{V^2}{2g}$ ☐ (b) $\dfrac{V^2}{2g}$ ☐

(c) $\dfrac{V^2}{r^2 \times 2g}$ ☐ (d) $\dfrac{\omega r^2}{2g}$. ☐

76. Bernoulli's equation is derived making assumptions that

(a) the flow is uniform, steady and incompressible ☐

(b) the flow is non-viscous, uniform and steady ☐

(c) the flow is steady, non-viscous, incompressible and irrotational ☐

(d) none of the above. ☐

77. The Bernoulli's equation can take the form

(a) $\dfrac{p_1}{\rho_1} + \dfrac{V_1^2}{2g} + Z_1 = \dfrac{p_2}{\rho_2} + \dfrac{V_2^2}{2g} + Z_2$ ☐ (b) $\dfrac{p_1}{\rho_2 q} + \dfrac{V_1^2}{2} + Z_1 = \dfrac{p_2}{\rho_2 q} + \dfrac{V_2^2}{2} + Z_2$ ☐

(c) $\dfrac{p_1}{\rho_1 q} + \dfrac{V_1^2}{2g} + Z_1 = \dfrac{p_2}{\rho_2 q} + \dfrac{V_2^2}{2g} + gZ_2$ ☐ (d) $\dfrac{p_1}{\rho_1 g} + \dfrac{V_1^2}{2g} + Z_1 = \dfrac{p_2}{\rho_2 g} + \dfrac{V_2^2}{2g} + Z_2$. ☐

78. The flow rate through a circular pipe is measured by

(a) Pitot-tube ☐ (b) Venturi-meter ☐

(c) Orifice-meter ☐ (d) None of the above. ☐

79. If the velocity, pressure, density etc., do not change at a point with respect to time, the flow is called

(a) uniform ☐ (b) incompressible ☐

(c) non-uniform ☐ (d) steady. ☐

80. If the velocity, pressure, density etc., change at a point with respect to time, the flow is called

(a) uniform ☐ (b) compressible ☐

(c) unsteady ☐ (d) incompressible. ☐

81. If the velocity in a fluid flow does not change with respect to length of direction of flow, it is called

(*a*) steady flow ☐ (*b*) uniform flow ☐

(*c*) incompressible flow ☐ (*d*) rotational flow. ☐

82. If the velocity in a fluid flow changes with respect to length of direction of flow, it is called

(*a*) unsteady flow ☐ (*b*) compressible flow ☐

(*c*) irrotational flow ☐ (*d*) none of the above. ☐

83. If the density of a fluid is constant from point to point in a flow region, it is called

(*a*) steady flow ☐ (*b*) incompressible flow ☐

(*c*) uniform flow ☐ (*d*) rotational flow. ☐

84. If the density of a fluid changes from point to point in a flow region, it is called

(*a*) steady flow ☐ (*b*) unsteady flow ☐

(*c*) non-uniform flow ☐ (*d*) compressible flow. ☐

85. If the fluid particles move in straight lines and all the lines are parallel to the surface, the flow is called

(*a*) steady ☐ (*b*) uniform ☐

(*c*) compressible ☐ (*d*) laminar. ☐

86. If the fluid particles move in a zigzag way, the flow is called

(*a*) unsteady ☐ (*b*) non-uniform ☐

(*c*) turbulent ☐ (*d*) incompressible. ☐

87. The acceleration of a fluid particle in the direction of *x* is given by

(*a*) $A_x = u\,\dfrac{\partial}{\partial x} + v\,\dfrac{\partial v}{\partial y} + w\,\dfrac{\partial w}{\partial z} + \dfrac{\partial u}{\partial t}$ ☐ (*b*) $A_x = u\,\dfrac{\partial u}{\partial x} + u\,\dfrac{\partial v}{\partial y} + v\,\dfrac{\partial w}{\partial z} + \dfrac{\partial u}{\partial t}$ ☐

(*c*) $A_x = u\,\dfrac{\partial u}{\partial x} + v\,\dfrac{\partial u}{\partial y} + w\,\dfrac{\partial u}{\partial z} + \dfrac{\partial u}{\partial t}$ ☐ (*d*) none of the above. ☐

88. The local acceleration in the direction of *x* is given by

(*a*) $u\,\dfrac{\partial u}{\partial x} + \dfrac{\partial u}{\partial t}$ ☐ (*b*) $\dfrac{\partial u}{\partial t}$ ☐

(*c*) $u\,\dfrac{\partial u}{\partial x}$ ☐ (*d*) none of the above. ☐

89. The convective acceleration in the direction of *x* is given by

(*a*) $u\,\dfrac{\partial u}{\partial x} + v\,\dfrac{\partial v}{\partial y} + w\,\dfrac{\partial w}{\partial z}$ ☐ (*b*) $u\,\dfrac{\partial u}{\partial x} + u\,\dfrac{\partial u}{\partial y} + u\,\dfrac{\partial u}{\partial z}$ ☐

(*c*) $u\,\dfrac{\partial u}{\partial x} + u\,\dfrac{\partial v}{\partial y} + u\,\dfrac{\partial w}{\partial z}$ ☐ (*d*) $u\,\dfrac{\partial u}{\partial x} + v\,\dfrac{\partial u}{\partial y} + w\,\dfrac{\partial u}{\partial z}$. ☐

90. Shear strain rate is given by

(*a*) $\frac{1}{2}\left(\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}\right)$ ☐ (*b*) $\frac{1}{2}\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}$ ☐

(*c*) $\frac{1}{2}\left(\frac{\partial v}{\partial x}+\frac{\partial u}{\partial y}\right)$ ☐ (*d*) $\frac{1}{2}\frac{\partial v}{\partial x}+\frac{\partial u}{\partial y}$. ☐

91. For a two-dimensional fluid element in *x-y* plane, the rotational component is given as

(*a*) $\omega_z=\frac{1}{2}\left(\frac{\partial v}{\partial x}+\frac{\partial u}{\partial y}\right)$ ☐ (*b*) $\omega_z=\frac{1}{2}\left(\frac{\partial u}{\partial x}-\frac{\partial v}{\partial y}\right)$ ☐

(*c*) $\omega_z=\frac{1}{2}\left(\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}\right)$ ☐ (*d*) $\omega_z=\frac{1}{2}\left(\frac{\partial v}{\partial x}-\frac{\partial u}{\partial y}\right)$. ☐

92. Vorticity is given by

(*a*) two times the rotation ☐ (*b*) 1.5 times the rotation ☐

(*c*) three times the rotation ☐ (*d*) equal to the rotation. ☐

93. Study of fluid motion with the forces causing the flow is known as

(*a*) kinematics of fluid flow ☐ (*b*) dynamics of fluid flow ☐

(*c*) statics of fluid flow ☐ (*d*) none of the above. ☐

94. Study of fluid motion without considering the forces causing the flow is known as

(*a*) kinematics of fluid flow ☐ (*b*) dynamics of fluid flow ☐

(*c*) statics of fluid flow ☐ (*d*) none of the above. ☐

95. Study of fluid at rest, is known as

(*a*) kinematics ☐ (*b*) dynamics ☐

(*c*) statics ☐ (*d*) none of the above. ☐

96. The term $V^2/2g$ is known as

(*a*) kinetic energy ☐ (*b*) pressure energy ☐

(*c*) kinetic energy per unit weight ☐ (*d*) none of the above. ☐

97. The term $p/\rho g$ is known as

(*a*) kinetic energy per unit weight ☐ (*b*) pressure energy ☐

(*c*) pressure energy per unit weight ☐ (*d*) none of the above. ☐

98. The term Z is known as

(*a*) potential energy ☐ (*b*) pressure energy ☐

(*c*) potential energy per unit weight ☐ (*d*) none of the above. ☐

99. The discharge through a venturimeter is given as

(*a*) $Q=\frac{A_1^2A_2^2}{\sqrt{A_1^2-A_2^2}}\times\sqrt{2gh}$ ☐ (*b*) $Q=\frac{A_1A_2}{\sqrt{2A_1^2-A_2^2}}\times\sqrt{2gh}$ ☐

(*c*) $Q=\frac{A_1A_2}{\sqrt{A_1^2-A_2^2}}\times\sqrt{2gh}$ ☐ (*d*) none of the above. ☐

100. The difference of pressure head (h) measured by a mercury-oil differential manometer is given as

(a) $h = x\left[1 - \frac{S_g}{S_0}\right]$ ☐ (b) $h = x\,[S_g - S_0]$ ☐

(c) $h = x\,[S_0 - S_g]$ ☐ (d) $h = x\left[\frac{S_g}{S_0} - 1\right]$ ☐

where x = Difference of mercury level, S_g = Specific gravity of mercury, and S_0 = Specific gravity of oil.

101. The difference of pressure head (h) measured by a differential manometer containing lighter liquid is

(a) $h = x\left[1 - \frac{S_l}{S_0}\right]$ ☐ (b) $h = x\left[\frac{S_l}{S_0} - 1\right]$ ☐

(c) $h = x\,[S_0 - S_l]$ ☐ (d) none of the above ☐

where S_l = Specific gravity of lighter liquid in manometer

S_0 = Specific gravity of fluid flowing

x = Difference of lighter liquid levels in differential manometer.

102. Pitot-tube is used to measure

(a) discharge ☐ (b) average velocity ☐

(c) velocity at a point ☐ (d) pressure at a point. ☐

103. Venturimeter is used to measure

(a) discharge ☐ (b) average velocity ☐

(c) velocity at a point ☐ (d) pressure at a point. ☐

104. Orifice-meter is used to measure

(a) discharge ☐ (b) average velocity ☐

(c) velocity at a point ☐ (d) pressure at a point. ☐

105. For a sub-merged curved surface, the horizontal component of force due to static liquid is equal to

(a) weight of liquid supported by the curved surface ☐

(b) force on a projection of the curved surface on a vertical plane ☐

(c) area of curved surface × pressure at the centroid of the submerged area ☐

(d) none of the above. ☐

106. For a sub-merged curved surface, the component of force due to static liquid is equal to

(a) weight of the liquid supported by curved surface ☐

(b) force on a projection of the curved surface on a vertical plane ☐

(c) area of curved surface × pressure at the centroid of the sub-merged area ☐

(d) none of the above. ☐

107. An oil of specific gravity 0.7 and pressure 0.14 kgf/cm^2 will have the height of oil as

(a) 70 cm of oil ☐ (b) 2 m of oil ☐

(c) 20 cm of oil ☐ (d) 10 cm of oil. ☐

108. The difference in pressure head, measured by a mercury water differential manometer for a 20 m difference of mercury level will be

(*a*) 2.72 m ☐ (*b*) 2.52 m ☐
(*c*) 2.0 m ☐ (*d*) 0.2 m. ☐

109. The difference in pressure head, measured by a mercury-oil differential manometer for a 20 cm difference of mercury level will be (sp. gr. of oil = 0.8)

(*a*) 2.72 m of oil ☐ (*b*) 2.52 m of oil ☐
(*c*) 3.20 m of oil ☐ (*d*) 2.0 m of oil. ☐

110. The rate of flow through a venturimeter varies as

(*a*) H ☐ (*b*) $\sqrt{H}$ ☐
(*c*) $H^{3/2}$ ☐ (*d*) $H^{5/2}$. ☐

111. The rate of flow through a V-notch varies as

(*a*) H ☐ (*b*) $\sqrt{H}$ ☐
(*c*) $H^{3/2}$ ☐ (*d*) $H^{5/2}$. ☐

Orifices and Mouthpieces

112. The range for coefficient of discharge (C_d) for a venturimeter is

(*a*) 0.6 to 0.7 ☐ (*b*) 0.7 to 0.8 ☐
(*c*) 0.8 to 0.9 ☐ (*d*) 0.95 to 0.99. ☐

113. The coefficient of velocity (C_v) for an orifice is

(*a*) $C_v = \sqrt{\frac{4x^2}{yH}}$ ☐ (*b*) $C_v = \sqrt{\frac{2x}{4yH}}$ ☐
(*c*) $C_v = \sqrt{\frac{x^2}{4yH}}$ ☐ (*d*) none of the above. ☐

114. The coefficient of discharge (C_d) in terms of C_v and C_c is

(*a*) $C_d = \frac{C_v}{C_c}$ ☐ (*b*) $C_d = C_v \times C_c$ ☐
(*c*) $C_d = \frac{C_c}{C_v}$ ☐ (*d*) none of the above. ☐

115. An orifice is known as large orifice when the head of liquid from the centre of orifice is

(*a*) more than 10 times the depth of orifice ☐ (*b*) less than 10 times the depth of orifice ☐
(*c*) less than 5 times the depth of orifice ☐ (*d*) none of the above. ☐

116. Which mouthpiece is having maximum coefficient of discharge?

(*a*) external mouthpiece ☐ (*b*) convergent divergent mouthpiece ☐
(*c*) internal mouthpiece ☐ (*d*) none of the above. ☐

117. The coefficient of discharge (C_d)

(*a*) for an orifice is more than that for a mouthpiece ☐
(*b*) for internal mouthpiece is more than that external mouthpiece ☐

(*c*) for a mouthpiece is more than that for an orifice ☐
(*d*) none of the above. ☐

118. Orifices are used to measure
(*a*) velocity ☐ (*b*) pressure ☐
(*c*) rate of flow ☐ (*d*) none of the above. ☐

119. Mouthpieces are used to measure
(*a*) velocity ☐ (*b*) pressure ☐
(*c*) viscosity ☐ (*d*) rate of flow. ☐

120. The ratio of actual velocity of a jet of water at veena-contracta to the theoretical velocity is known as
(*a*) coefficient of discharge ☐ (*b*) coefficient of velocity ☐
(*c*) coefficient of contraction ☐ (*d*) coefficient of viscosity. ☐

121. The ratio of actual discharge of a jet of water to its theoretical discharge is known as
(*a*) coefficient of discharge ☐ (*b*) coefficient of velocity ☐
(*c*) coefficient of contraction ☐ (*d*) coefficient of viscosity. ☐

122. The ratio of the area of the jet of water at veena-contracta to the area of orifice is known as
(*a*) coefficient of discharge ☐ (*b*) coefficient of velocity ☐
(*c*) coefficient of contraction ☐ (*d*) coefficient of viscosity. ☐

123. The discharge through a large rectangular orifice is
(*a*) $\frac{2}{3} C_d \times b \times \sqrt{2g}(\sqrt{H_2} - \sqrt{H_1})$ ☐ (*b*) $\frac{8}{15} C_d \times b \times \sqrt{2g}\,(H_2^{3/2} - H_1^{3/2})$ ☐
(*c*) $\frac{2}{3} C_d \times b \times \sqrt{2g}\,(H_2^{3/2} - H_1^{3/2})$ ☐ (*d*) none of the above ☐

where b = Width of orifice, H_1 = Height of liquid above top edge of the orifice, H_2 = Height of liquid above bottom edge of orifice.

124. The discharge through fully submerged orifice is
(*a*) $C_d \times b \times (H_2 - H_1) \times \sqrt{2g} \times H^{3/2}$ ☐ (*b*) $C_d \times b \times (H_2 - H_1) \times \sqrt{2gH}$ ☐
(*c*) $C_d \times b \times (H_2^{3/2} - H_1^{3/2}) \times \sqrt{2gH}$ ☐ (*d*) none of the above ☐

where H = Difference of liquid levels on both sides of the orifice
H_1 = Height of liquid above top edge orifice of upstream side
H_2 = Height of liquid above bottom edge of orifice on upstream side.

Notches and Weirs

125. Notch is a device used for measuring
(*a*) rate of flow through pipes ☐ (*b*) rate of flow through a small channel ☐
(*c*) velocity through a pipe ☐ (*d*) velocity through a small channel. ☐

126. The discharge through a rectangular notch is given by

(*a*) $Q = \frac{2}{3} C_d \times L \times H^{5/2}$ ☐ (*b*) $Q = 2/3\ C_d \times L \times H^{3/2}$ ☐

(*c*) $Q = \frac{2}{3} C_d \times L \times H^{5/2}$ ☐ (*d*) $Q = 8/15\ C_d \times L \times H^{3/2}$. ☐

127. The discharge through a triangular notch is given by

(*a*) $Q = 2/3\ C_d \times \tan \frac{\theta}{2} \times \sqrt{2gH}$ ☐ (*b*) $Q = 2/3\ C_d \times \tan \frac{\theta}{2} \times \sqrt{2g} \times H^{3/2}$ ☐

(*c*) $Q = 8/15\ C_d \times \tan \frac{\theta}{2} \times \sqrt{2g}\ H^{5/2}$ ☐ (*d*) none of the above. ☐

where θ = Total angle of triangular notch, H = Head over notch.

128. The discharge through a trapezoidal notch is given as

(*a*) $Q = 2/3\ C_{d1} \times L \times H^{3/2} + 8/15\ C_{d2} \times \tan \theta/2 \times \sqrt{2g} \times H^{3/2}$ ☐

(*b*) $Q = 2/3\ C_{d1} \times L \times H^{5/2} + 8/15\ C_{d2} \times \tan \theta/2 \times \sqrt{2g}\ H^{3/2}$ ☐

(*c*) $Q = 2/3\ C_{d1} \times L \times H^{3/2} + 8/15\ C_{d2} \times \tan \theta/2 \times \sqrt{2g}\ H^{5/2}$ ☐

(*d*) none of the above ☐

where $\theta/2$ = Slope of the side of the trapezoidal notch.

129. The error in discharge due to the error in the measurement of head over a rectangular notch is given by

(*a*) $\frac{dQ}{Q} = \frac{5}{2}\frac{dH}{H}$ ☐ (*b*) $\frac{dQ}{Q} = \frac{3}{2}\frac{dH}{H}$ ☐

(*c*) $\frac{dQ}{Q} = \frac{7}{2}\frac{dH}{H}$ ☐ (*d*) $\frac{dQ}{Q} = \frac{1}{2}\frac{dH}{H}$. ☐

130. The error in discharge due to the error in the measurement of head over a triangular notch is given by

(*a*) $\frac{dQ}{Q} = \frac{5}{2}\frac{dH}{H}$ ☐ (*b*) $\frac{dQ}{Q} = \frac{3}{2}\frac{dH}{H}$ ☐

(*c*) $\frac{dQ}{Q} = \frac{7}{2}\frac{dH}{H}$ ☐ (*d*) $\frac{dQ}{Q} = \frac{1}{2}\frac{dH}{H}$. ☐

131. The velocity with which the water approaches a notch is called

(*a*) velocity of flow ☐ (*b*) velocity of approach ☐

(*c*) velocity of whirl ☐ (*d*) none of the above. ☐

132. The discharge over a rectangular notch considering velocity of approach is given as

(*a*) $Q = \frac{3}{2} C_d L\sqrt{2g}\ (H^{3/2} - h_d^{3/2})$ ☐ (*b*) $Q = \frac{2}{3} C_d L\sqrt{2g}\ (H - h_a)^{3/2}$ ☐

(*c*) $Q = \frac{2}{3} C_d L\sqrt{2g}\ [(H + h_a)^{3/2} - h_a^{3/2}]$ ☐ (*d*) none of the above ☐

where H = Head over notch, and h_a = Head due to velocity of approach.

133. The velocity of approach (V_a) is given by

(*a*) $V_a = \dfrac{\text{Discharged over notch}}{\text{Area of notch}}$ ☐ (*b*) $V_a = \dfrac{\text{Discharged over notch}}{\text{Area of channel}}$ ☐

(*c*) $V_a = \dfrac{\text{Discharged over notch}}{\text{Heat over notch} \times \text{Width of channel}}$ ☐ (*d*) none of the above. ☐

134. Francis's formula for a rectangular weir with end contraction suppressed is given as

(*a*) $Q = 1.84\ LH^{5/2}$ ☐ (*b*) $Q = 2/3\ L \times H^{3/2}$ ☐

(*c*) $Q = 1.84\ LH^{3/2}$ ☐ (*d*) $Q = 2/3\ L \times H^{5/2}$. ☐

135. Francis's formula for a rectangular weir for two end contractions is given by

(*a*) $Q = 1.84[L - 0.2 \times 2H]H^{5/2}$ ☐ (*b*) $Q = 1.84[L - 0.2H]H^{3/2}$ ☐

(*c*) $Q = 1.84[L - 0.2H]H^{5/2}$ ☐ (*d*) none of the above. ☐

136. Bazin's formula for discharge over a rectangular weir without velocity of approach is given by

(*a*) $Q = mL \times \sqrt{2gH^{5/2}}$ ☐ (*b*) $Q = mL \times \sqrt{2g} \times H^{3/2}$ ☐

(*c*) $Q = m \times L \times \sqrt{2gH}$ ☐ (*d*) none of the above ☐

where $m = 0.405 + \dfrac{0.003}{H}$ and H = Head over weir.

137. Cipolletti weir is a trapezoidal weir having side slope of

(*a*) 1 horizontal to 2 vertical ☐ (*b*) 4 horizontal to 1 vertical ☐

(*c*) 1 horizontal to 4 vertical ☐ (*d*) 1 horizontal to 3 vertical. ☐

Laminar and Turbulent Flow Through Pipes

138. A flow is said to be laminar when

(*a*) the fluid particles move in a zigzag way ☐

(*b*) the Reynold number is high ☐

(*c*) the fluid particles move in layers parallel to the boundary ☐

(*d*) none of the above. ☐

139. For the laminar flow through a circular pipe

(*a*) the maximum velocity = 1.5 times the average velocity ☐

(*b*) the maximum velocity = 2.0 times the average velocity ☐

(*c*) the maximum velocity = 2.5 times the average velocity ☐

(*d*) none of the above. ☐

140. The loss of pressure head for the laminar flow through pipes varies

(*a*) as the square of velocity ☐ (*b*) directly as the velocity ☐

(*c*) as the inverse of the velocity ☐ (*d*) none of the above. ☐

141. For the laminar flow through a pipe, the shear stress over the cross-section

(*a*) varies inversely as the distance from the centre of the pipe ☐

(*b*) varies directly as the distance from the surface of the pipe ☐

(*c*) varies directly as the distance from the centre of the pipe ☐
(*d*) remains constant over the cross-section. ☐

142. For the laminar flow between two parallel plates
(*a*) the maximum velocity = 2.0 times the average velocity ☐
(*b*) the maximum velocity = 2.5 times of the average velocity ☐
(*c*) the maximum velocity = 1.33 times the average velocity ☐
(*d*) none of the above. ☐

143. The valve of the kinetic energy correction factor (α) for the viscous flow through a circular pipe is
(*a*) 1.33 ☐ (*b*) 1.50 ☐
(*c*) 2.0 ☐ (*d*) 1.25. ☐

144. The valve of the momentum correction factor (β) for the viscous flow through a circular pipe is
(*a*) 1.33 ☐ (*b*) 1.50 ☐
(*c*) 2.0 ☐ (*d*) 1.25. ☐

145. The pressure drop per unit length of a pipe for laminar flow is
(*a*) equal to $\frac{12\mu\overline{UL}}{\rho g D^2}$ ☐ (*b*) equal to $\frac{12\mu\overline{U}}{\rho g D^2}$ ☐
(*c*) equal to $\frac{32\mu\overline{UL}}{\rho g D^2}$ ☐ (*d*) none of the above. ☐

146. For viscous flow between two parallel plates, the pressure drop per unit length is equal to
(*a*) $\frac{12\mu\overline{U}L}{\rho g D^2}$ ☐ (*b*) $\frac{12\mu\overline{U}L}{D^2}$ ☐
(*c*) $\frac{32\mu\overline{U}L}{D^2}$ ☐ (*d*) $\frac{12\mu\overline{U}}{D^2}$. ☐

147. The velocity distribution in laminar flow through a circular pipe follows the
(*a*) parabolic law ☐ (*b*) linear law ☐
(*c*) logarithmic law ☐ (*d*) none of the above. ☐

148. A boundary is known as hydrodynamically smooth, if
(*a*) $\frac{k}{\delta'} = 0.3$ ☐ (*b*) $\frac{k}{\delta'} > 0.3$ ☐
(*c*) $\frac{k}{\delta'} < 0.25$ ☐ (*d*) $\frac{k}{\delta'} = 6.0$ ☐
where k = Average height of the irregularities from the boundary and δ' = Thickness of laminar sub-layer.

149. The coefficient of friction for laminar flow through a circular pipe is given by
(*a*) $f = \frac{0.0791}{(R_e)^{1/4}}$ ☐ (*b*) $f = \frac{16}{R_e}$ ☐
(*c*) $f = \frac{64}{R_e}$ ☐ (*d*) none of the above. ☐

150. The loss of head due to sudden expansion of a pipe is given by

(*a*) $h_L = \dfrac{V_1^2 - V_2^2}{2g}$ ☐ (*b*) $h_L = \dfrac{0.5\,V_1^2}{2g}$ ☐

(*c*) $h_L = \dfrac{(V_1 - V_2)^2}{2g}$ ☐ (*d*) none of the above. ☐

151. The loss of head due to sudden contraction of a pipe is equal to

(*a*) $\left(\dfrac{1}{C_c} - 1\right)^2 \dfrac{V_2}{2g}$ ☐ (*b*) $\left(1 - \dfrac{1}{C_c}\right)^2 \dfrac{V_2}{2g}$ ☐

(*c*) $\dfrac{1}{C_c}\left(1 - \dfrac{V_2^{\,2}}{2g}\right)$ ☐ (*d*) none of the above. ☐

152. Hydraulic gradient line (H.G.L.) represents the sum of

(*a*) pressure head and kinetic head ☐
(*b*) kinetic head and datum head ☐
(*c*) pressure head, kinetic head and datum head ☐
(*d*) pressure head and datum head. ☐

153. Total Energy Line (T.E.L.) represents the sum of

(*a*) pressure head and kinetic head ☐
(*b*) kinetic head and datum head ☐
(*c*) pressure head and datum head ☐
(*d*) pressure head, kinetic head and datum head. ☐

154. When the pipes are connected in series, the total rate of flow

(*a*) is equal to the sum of the rate of flow in each pipe ☐
(*b*) is equal to the reciprocal of the sum of rate of flow in each pipe ☐
(*c*) is the same as flowing through each pipe ☐
(*d*) none of the above. ☐

155. Power transmitted through pipes will be maximum when

(*a*) head lost due to friction = $\frac{1}{2}$ total head at inlet of the pipe ☐
(*b*) head lost due to friction = $\frac{1}{4}$ total head at inlet of the pipe ☐
(*c*) head lost due to friction = total head at the inlet of the pipe ☐
(*d*) head lost due to friction = $\frac{1}{3}$ total head at the inlet of the pipe. ☐

156. The valve closure is said to be gradual if the time required to close the valve

(*a*) $t = \dfrac{2L}{C}$ ☐ (*b*) $t \le \dfrac{2L}{C}$ ☐

(*c*) $t < \dfrac{4L}{C}$ ☐ (*d*) $t > \dfrac{2L}{C}$. ☐

where L = Length of pipe, C = Velocity of pressure wave.

157. The velocity of pressure wave in terms of bulk modulus (K) and density (ρ) is given by

(a) $C = \sqrt{\frac{\rho}{K}}$ ☐ (b) $C = \sqrt{K\rho}$ ☐

(c) $C = \sqrt{\frac{K}{\rho}}$ ☐ (d) none of the above. ☐

158. The coefficient of friction in terms of shear stress is given by

(a) $f = \frac{2\tau V^2}{\tau_0}$ ☐ (b) $f = \frac{2\tau_0}{\rho V^2}$ ☐

(c) $f = \frac{\tau_0}{2\rho V^2}$ ☐ (d) $f = \frac{\rho V^2}{2\tau_0}$. ☐

159. Reynold shear stress for turbulent flow is given by

(a) $\tau = \overline{\rho u v'}$ ☐ (b) $\bar{\tau} = \mu \frac{\partial u}{\partial y}$ ☐

(c) $\bar{\tau} = \eta \frac{\overline{du}}{\partial y}$ ☐ (d) none of the above. ☐

where u', v' = Fluctuating component of velocity in the direction x and y and η = Eddy viscosity.

160. The shear stress in turbulent flow due to Prandtl is given by

(a) $\bar{\tau} = \rho l^2 \left(\frac{du}{dy}\right)^2$ ☐ (b) $\bar{\tau} = \rho^2 l \left(\frac{du}{dy}\right)^2$ ☐

(c) $\bar{\tau} = \rho^2 l^2 \left(\frac{du}{dy}\right)$ ☐ (d) none of the above. ☐

where l = Mixing length.

161. Shear velocity (u_*) is equal to

(a) $\sqrt{\rho\tau_0}$ ☐ (b) $\sqrt{\frac{\tau_0}{\rho}}$ ☐

(c) $\sqrt{\frac{\rho}{\tau_0}}$ ☐ (d) $\frac{1}{\sqrt{\rho\tau_0}}$ ☐

where τ_0 = Shear stress at the surface.

162. The velocity distribution in turbulent flow for pipes is given by

(a) $u = U_{max} + 5.5\, u_* \log_e (y/R)$ ☐ (b) $u = 2.5\, u_* \log_e (y/R)$ ☐

(c) $u = U_{max} + 2.5\, u_* \log_e (y/R)$ ☐ (d) none of the above. ☐

where u_* = Shear velocity, R = Radius of pipe, y = Distance from pipe wall, U_{max} = Centre-line velocity.

163. When the pipes are connected in parallel, the total loss of head

(a) is equal to the sum of the loss of head in each pipe ☐

(b) is same as in each pipe ☐

(*c*) is equal to the reciprocal of the sum of loss of head in each pipe ☐
(*d*) none of the above. ☐

164. L_1, L_2, L_3 are the length of three pipes, connected in series. If d_1, d_2 and d_3 are their diameters, then the equivalent size of the pipe is given by

(*a*) $\frac{L}{d^5} = \frac{L_1}{d_1^5} + \frac{L_2}{d_2^5} + \frac{L_3}{d_3^5}$ ☐ (*b*) $\frac{d^5}{L} = \frac{d_1^5}{L_1} + \frac{d_2^5}{L_2} + \frac{d_3^5}{L_3}$ ☐

(*c*) $Ld^5 = L_1 d_1^5 + L_2 d_2^5 + L_3 d_3^5$ ☐ (*d*) none of the above. ☐

where $L = L_1 + L_2 + L_3$.

165. The power transmitted in kW through pipe is given by

(*a*) $\frac{\rho \times g \times Q \times H}{75}$ ☐ (*b*) $\frac{\rho \times g \times Q \times h_f}{1000}$ ☐

(*c*) $\frac{\rho \times g \times Q \times (H - h_f)}{4500}$ ☐ (*d*) $\frac{\rho \times g \times Q \times (H - h_f)}{1000}$ ☐

where H = Total head at the inlet of pipe, h_f = Head lost due to friction in pipe and Q = Discharge per second.

166. Efficiency of power transmission through pipe is given by

(*a*) $\frac{H - h_f}{H}$ ☐ (*b*) $\frac{H}{H + h_f}$ ☐

(*c*) $\frac{H - h_f}{H + h_f}$ ☐ (*d*) none of the above. ☐

where H = Total head at inlet, h_f = Head lost due to friction.

167. Maximum efficiency of power transmission through pipe is

(*a*) 50% ☐ (*b*) 66.67% ☐
(*c*) 75% ☐ (*d*) 100%. ☐

168. Diameter of nozzle (d) for maximum power transmission is given by

(*a*) $d = \left(\frac{D^4}{8fL}\right)^{1/5}$ ☐ (*b*) $d = \left(\frac{D^5}{8fL}\right)^{1/5}$ ☐

(*c*) $d = \left(\frac{D^5}{8fL}\right)^{1/4}$ ☐ (*d*) none of the above. ☐

where D = Dia. of pipe, L = Length of pipe.

169. Water-hammer in pipes takes place when

(*a*) fluid is flowing with high velocity ☐
(*b*) fluid is flowing with high pressure ☐
(*c*) flowing fluid is suddenly brought to rest by closing the valve ☐
(*d*) flowing fluid is gradually brought to rest. ☐

170. The pressure rise (p_i) due to water hammer, when the valve is closed suddenly and pipe is assumed rigid, is equal to

(*a*) $V\sqrt{\frac{k}{\rho}}$ ☐ (*b*) $V\sqrt{k\rho}$ ☐

(*c*) $V\sqrt{\frac{\rho}{k}}$ ☐ (*d*) $Vk\rho$ ☐

where V = Velocity of flow, k = Bulk modulus of water, and ρ = Density of fluid.

171. The pressure rise (p_i) due to water hammer, when valve is closed gradually is equal to

(*a*) ρLV ☐ (*b*) $\frac{\rho LV}{t}$ ☐

(*c*) $\frac{\rho t}{VL}$ ☐ (*d*) $\frac{\rho}{LVt}$ ☐

where t = Time required to close the valve.

172. The pressure rise (p_i) due to water hammer, when valve is closed suddenly and pipe is elastic, is equal to

(*a*) $V \times \sqrt{\frac{kEt}{\rho D}}$ ☐ (*b*) $V \times \sqrt{\frac{\frac{1}{k}+\frac{D}{Et}}{\rho}}$ ☐

(*c*) $V \times \sqrt{\frac{\rho}{\frac{1}{k}+\frac{D}{Et}}}$ ☐ (*d*) none of the above. ☐

where E = Modulus of elasticity for pipe material, D = Diameter of pipe, t = Time required to close valve, and k = Bulk modulus of water.

173. The pressure rise (p_i) due to water hammer depends on

(*a*) the diameter of pipe only ☐
(*b*) the length of pipe only ☐
(*c*) the required to close the valve only ☐
(*d*) elastic properties of the pipe material only ☐
(*e*) elastic properties of liquid flowing through pipe only ☐
(*f*) all of the above. ☐

174. The valve closure is said to be sudden if the time required to close the valve

(*a*) $t = \frac{2L}{C}$ ☐ (*b*) $t < \frac{2L}{C}$ ☐

(*c*) $t > \frac{2L}{C}$ ☐ (*d*) none of the above ☐

where C = Velocity of pressure wave produced, and L = Length of pipe.

175. For a viscous flow through circular pipes, certain curves are shown in Fig. 1.3. Curve A is for

(*a*) shear stress distribution ☐
(*b*) velocity distribution ☐
(*c*) pressure distribution ☐
(*d*) none of the above. ☐

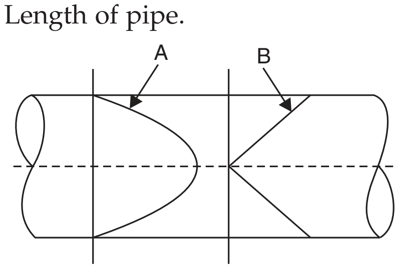

FIGURE 1.3

176. Curve B in Fig. 1.3 is for

(*a*) shear stress distribution ☐ (*b*) velocity distribution ☐
(*c*) pressure distribution ☐ (*d*) none of the above. ☐

177. Figure 1.4 shows four curves for velocity distribution across a section for Reynolds number equal to 1000, 4000, 6000 and 10000. Curve A corresponds to Reynold number equal to

(*a*) 1000 ☐
(*b*) 4000 ☐
(*c*) 6000 ☐
(*d*) 10000. ☐

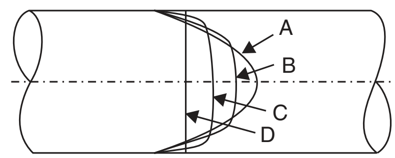

FIGURE 1.4

178. Curve B in Fig. 1.4 corresponds to Reynolds number

(*a*) 1000 ☐ (*b*) 4000 ☐
(*c*) 6000 ☐ (*d*) 10000. ☐

179. Curve C in Fig. 1.4 corresponds to the Reynold number

(*a*) 1000 ☐ (*b*) 4000 ☐
(*c*) 6000 ☐ (*d*) 10000. ☐

180. Curve D in Fig. 1.4 corresponds to the Reynold number

(*a*) 1000 ☐ (*b*) 4000 ☐
(*c*) 6000 ☐ (*d*) 10000. ☐

181. The shear stress distribution across a section of a circular pipe having viscous flow is given by

(*a*) $\tau = \frac{\partial p}{\partial x} r^2$ ☐ (*b*) $\tau = \frac{\partial p}{\partial x} \frac{r}{2}$ ☐

(*c*) $\tau = -\frac{\partial p}{\partial x} \frac{r}{2}$ ☐ (*d*) $\tau = -\frac{\partial p}{\partial x} \times 2r$. ☐

182. The velocity distribution across a section of a circular pipe having viscous flow is given by

(*a*) $u = U_{max}\left[1-\left(\frac{r}{R}\right)^2\right]$ ☐ (*b*) $u = U_{max}\,[R^2 - r^2]$ ☐

(*c*) $u = U_{max}\left[1-\frac{r}{R}\right]^2$ ☐ (*d*) none of the above. ☐

183. The velocity distribution across a section of two fixed parallel plates having viscous flow is given by

(*a*) $u = \frac{1}{2\mu}\left(-\frac{\partial p}{\partial x}\right)(t^2 - y^2)$ ☐ (*b*) $u = -\frac{1}{2\mu}\frac{\partial p}{\partial x}\,[ty - y^2]$ ☐

(*c*) $u = \frac{1}{2\mu}\frac{\partial p}{\partial x}\,[y - ty]$ ☐ (*d*) $u = -\frac{1}{2\mu}\frac{\partial p}{\partial x}\,[t - y^2]$ ☐

where t = Distance between two plates and y is measured from the lower plate.

184. The shear stress distribution across a section of two fixed parallel plates having viscous flow is given by

(*a*) $\tau = -\frac{1}{2}\frac{\partial p}{\partial x}[t^2 - y^2]$ ☐ (*b*) $\tau = -\frac{1}{2}\frac{\partial p}{\partial x}[t - 2y]$ ☐

(*c*) $\tau = -\frac{1}{2}\frac{\partial p}{\partial x}[ty - y^2]$ ☐ (*d*) $\tau = \frac{1}{2}\frac{\partial p}{\partial x}[y - ty]$ ☐

where t = Distance between two parallel plates and y is measured from the plate.

Dimensional and Model Analysis

185. Reynold's number is defined as the

(*a*) ratio of inertia force to gravity force ☐ (*b*) ratio of viscous force to gravity force ☐

(*c*) ratio of viscous force to elastic force ☐ (*d*) ratio of inertia force to viscous force. ☐

186. Froude's number is defined as the ratio of

(*a*) inertia force to viscous force ☐ (*b*) inertia force to gravity force ☐

(*c*) inertia force to elastic force ☐ (*d*) inertia force to pressure force. ☐

187. Mach number is defined as the ratio of

(*a*) inertia force to viscous force ☐ (*b*) viscous force to surface tension force ☐

(*c*) viscous force to elastic force ☐ (*d*) inertia force to elastic force. ☐

188. Euler's number is the ratio of

(*a*) inertia force to pressure force ☐ (*b*) inertia force to elastic force ☐

(*c*) inertia force to gravity force ☐ (*d*) none of the above. ☐

189. Models are known undistorted model, if

(*a*) the prototype and model are having different scale ratios ☐

(*b*) the prototype and model are having same scale ratio ☐

(*c*) model and prototype are kinematically similar ☐

(*d*) none of the above. ☐

190. Geometric similarity between model and prototype means

(*a*) the similarity of discharge ☐ (*b*) the similarity of linear dimensions ☐

(*c*) the similarity of motion ☐ (*d*) the similarity of forces. ☐

191. Kinematic similarity between model and prototype means

(*a*) the similarity of forces ☐ (*b*) the similarity of shape ☐

(*c*) the similarity of motion ☐ (*d*) the similarity of discharge. ☐

192. Dynamic similarity between model and prototype means

(*a*) the similarity of forces ☐ (*b*) the similarity of motion ☐

(*c*) the similarity of shape ☐ (*d*) none of the above. ☐

193. Reynold number is expressed as

(*a*) $R_e = \frac{\rho\mu L}{V}$ ☐ (*b*) $R_e = \frac{V\mu L}{\rho}$ ☐

(*c*) $R_e = \frac{\rho V L}{\mu}$ ☐ (*d*) $R_e = \frac{V \times L}{\nu}$. ☐

194. Froude's number (F_e) is given by

(*a*) $F_e = V\sqrt{\frac{L}{g}}$ ☐ (*b*) $F_e = V\sqrt{\frac{g}{L}}$ ☐

(*c*) $F_e = \frac{V}{\sqrt{L \cdot g}}$ ☐ (*d*) none of the above. ☐

195. Mach number (M) is given by

(*a*) $M = \frac{C}{V}$ ☐ (*b*) $M = V \times C$ ☐

(*c*) $M = \frac{V}{C}$ ☐ (*d*) none of the above. ☐

196. The ratio of inertia force to viscous force is known as

(*a*) Reynold number ☐ (*b*) Froude number ☐

(*c*) Mach number ☐ (*d*) Euler number. ☐

197. The square root of the ratio of inertia force to gravity force is called

(*a*) Reynold number ☐ (*b*) Froude number ☐

(*c*) Mach number ☐ (*d*) Euler number. ☐

198. The square root of the ratio of inertia force to force due to compressibility is known as

(*a*) Reynold number ☐ (*b*) Froude number ☐

(*c*) Mach number ☐ (*d*) Euler number. ☐

199. The square root of the ratio of inertia force to pressure force is known as

(*a*) Reynold number ☐ (*b*) Froude number ☐

(*c*) Mach number ☐ (*d*) Euler number. ☐

200. Model analysis of pipes flow are based on

(*a*) Reynold number ☐ (*b*) Froude number ☐

(*c*) Mach number ☐ (*d*) Euler number. ☐

201. Model analysis of free surface flows are based on

(*a*) Reynolds number ☐ (*b*) Froude number ☐

(*c*) Mach number ☐ (*d*) Euler number. ☐

202. Model analysis of aeroplanes and projectile moving at super-sonic speed are based on

(*a*) Reynold number ☐ (*b*) Froude number ☐

(*c*) Mach number ☐ (*d*) Euler number. ☐

Boundary Layer Flow

203. Boundary layer on a flat plate is called laminar boundary layer if

(*a*) Reynold number is less than 2000 ☐ (*b*) Reynold number is less than 4000 ☐

(*c*) Reynold number is less than 5×10^5 ☐ (*d*) None of the above. ☐

204. Boundary layer thickness (δ) is the distance from the surface of the solid body in the direction perpendicular to flow, where the velocity of fluid is equal to

(*a*) Free stream velocity ☐ (*b*) 0.9 times the free stream velocity ☐

(*c*) 0.99 times the free stream velocity ☐ (*d*) None of the above. ☐