6.1 Introduction

In a liquid, if the pressure at any point reduces to its vapour pressure, the liquid boils and vaporization starts. If the liquid contains dissolved gases, they are liberated at such low pressures. Large number of small size bubbles of liquid vapour and liberated gas are thus formed. These are then carried by the flow. These bubbles upon reaching a zone of higher pressure condense and collapse abruptly. The surrounding liquid rushes to fill the space or cavity created by the bubbles collapse. The liquid rushing from all the directions collides at the centre of cavity generating a pressure wave which travels almost at the velocity of sound thus giving rise to very high local pressures. If the bubbles collapse takes place at a solid boundary, it is subjected to these very high pressures. The boundary material may thus be stressed locally beyond its elastic limit, resulting ultimately in fatigue failure. Even if the bubble collapse and the consequent formation of cavities do not actually occur at the solid boundary, the intense pressures are propagated to the boundary through the pressure wave which originates from the cavity centre.

This sequence of formation of vapour bubbles, their transport to zones of high pressure and subsequent collapse repeat itself many times in a second, frequency depending directly on the velocity of the flow. These very high pressures, acting repeatedly over the solid boundary as repeated hammer-blows at a high frequency, cause severe damage to the boundary surface. The surface becomes badly dented, and at some places its material appears to have been eroded by the liquid flow. This phenomenon is known as pitting, and the surface is said to have been pitted by the action of the flowing liquid. The material of the boundary surface ultimately fails by fatigue. The entire phenomenon characterized by the bubble formation and the subsequent collapse is known as the cavitation. There may be considerable noise and vibrations associated with the cavitating flow. The efficiency of the system also
decreases on account of reduced flow-area due to the space occupied by the
bubbles as shown in Fig. 6.1 when cavitation occurs in a turbine or pump, the
noise may sound as if a small piece of stone and gravel are passing through it.

![Fig. 6.1 Variation of plant cavitation factor ($\sigma_p$) with efficiency ($\eta$)](image)

The cavitation may occur in high speed hydraulic machines such as
turbines, pumps, marine propellers and in hydraulic structures such as spillways
and openings of high dams. It may also occur in certain portions of the pipeline
and some devices used for flow measurement.

It may be noted that the damage to the solid boundary occurs at a point
of bubble collapse and not at the region of minimum pressure. And since the
collapse takes place in the region of high pressure zone, the damage, therefore,
lies downstream of the point of minimum pressure. It may, therefore, be seen
that the formation of vapour bubbles, is not damaging in itself, but represent a
source associated with severe damage of boundary with considerable vibrations
and noise.

Bernoulli’s equation expressed as energy per unit volume is given by

\[(\frac{\eta^* V_f^2}{2}) + p + \Delta Z = \text{a constant (say } Ev)\]  \hspace{1cm} (6.1)

The pressure at a point is

\[p = Ev - (\frac{\eta^* V_f^2}{2}) - \Delta Z\]  \hspace{1cm} (6.2)

The total energy per unit volume $E_v$ is a constant along a streamline in a
rotational flow and at all points in an irrotational flow. The pressure $p$ at a point,
then, depends upon the velocity and datum head. The pressure may get
reduced either by increased velocity or on account of higher elevation head or
due combination of both. At certain regions in the flow, these variables may be
so combined as to reduce the pressure to its vapour pressure value. Once this
pressure is obtained, the cavitation is imminent. Engineers are therefore
required to design and operate the hydraulic machines and structures in such a way that the possibility of occurrence of cavitation is ruled out. While designing, we must ensure that nowhere in the flow, pressure falls to such a low value as the vapour pressure of the liquid at the prevailing temperature. In case of water the absolute pressure head should not be allowed to fall below 2.5m in order to prevent cavitation.

6.2 Cavitation at Runner Vane Moving Zone

Bulb turbine units are used in the run-of-river hydroelectric power plants for low head ranges. The runner vanes operations and guide vanes operations are synchronized to produce the maximum turbine efficiency for a given head and discharge. The higher specific speeds in the bulb turbines make the full advantage of the concept of total energy utilization to result the maximum efficiency. The aim is to produce higher efficiencies over a lower net head range. This is especially critical in run-of-the river plants where dry seasons contribute too much of the annual operating periods. The upstream river geometry significantly influences the flow pattern into turbine. Poor orientation of the turbine inlet to the direction of flow in the river results in flow separation at the intake. Eddies and vortices form during draw at of water which adversely affect the turbine performances. A straight conical diffuser draft tube with a transition section from circular to rectangular cross section is utilized in these turbines.

Fig. 6.2 Runnervane assembly inside the discharge ring
The cavitation pitting are seen in the flow area of are at the discharge ring. The discharge ring assembly houses the runner vane assembly shown in Fig. 6.2. The discharge ring connects the distributor cone assembly and draft tube. The discharge ring assembly has four numbers of 20 mm copper pipes at the top half for the water spraying between the rotating runner vane and the discharge ring during condenser operations. As the velocity of the water at the exit of the runner vanes is higher as 9.37 m/s, the water boils inside the discharge ring and cavitation inception starts as local static pressure falls below vapour pressure. The vapour pockets thus formed at the exit of runner vane moves to a region of higher pressure inside the discharge ring where it collapses suddenly and causes noise, vibration and pitting of the discharge ring. As the growth of bubbles and their collapse take place at a high speed, the damage to the discharge ring is largely due to fatigue and the damaged metal surface appears with numerous drilled holes of different diameters varying from 3 mm to 6 mm with different depth of penetration from 6 mm to 10 mm.

6.3 Cavitation Pitting and its Effect on Discharge Ring

The discharge ring is subjected to a higher-pressure force due to cavitation which acts as repeated hammer blows. Whenever these forces are less than the elastic limit of the material of the discharge ring, there is no deformation. But when the quantum of cavitation force acting on the discharge ring is higher than the elastic limit of the material, then the deformation of discharge ring is permanent.

It has been observed, during the annual maintenance that the discharge ring which is deformed and gets back to the original shape when same is dismantled and reassembled. But the cavitation forces acted year after year have stressed the discharge ring beyond its elastic limit. Hence the discharge ring has to be replaced with a new at with a higher cost with a huge loss of revenue to the power utility services.
6.4 Runner Vane Clearance

The cavitation and water hammer forces are acting inside the discharge ring. The cavitation forces are emerging out due to low pressure region created inside the discharge ring at small guide vane openings. The water hammer forces are acting on the discharge ring at the time of shutdown of the machine due to deep setting of the turbine runner. Whenever the total forces are within the elastic limits of the discharge ring, the deformation is elastic. But when the forces exceed the elastic limit of the discharge ring the deformation at the discharge ring is in plastic region and the deformation is permanent. This was confirmed during the repairing and rectification work carried out on UNIT-I & II machines of the PH-III.

Equation for Condition 1: $a = 0.0010b^2 - 0.0762b + 3.2733$
Equation for Condition 2: $i = -0.0027j^2 - 0.1444j + 3.5957$

Fig. 6.3 Runner vane clearance at top

The equation for the runner vane Top clearance for condition-1 has been derived using Mat lab software as $a = 0.0001b^2 + 0.0762b + 3.2733$, where $a =$ Clearance in mm in the Top direction, $b =$ No. of hours worked in 1000. The curve fitting equation for the Top clearance for condition-2 has been derived as $i = -0.0027j^2 - 0.1444j + 3.5957$, where $i =$ Clearance in mm in the Top direction
and \( j \) = No. of hours worked in 1000. The clearances obtained by solving these curve fitting equations are compared with the measured clearances and are shown in Fig. 6.3.

\[
\text{Equation for Condition 1: } c = 0.0010d^2 + 0.0762d + 3.2733 \\
\text{Equation for Condition 2: } k = -0.0018l^2 + 0.1128l + 3.7397
\]

Fig. 6.4 Runner vane clearance at bottom

The equation for the runner vane Bottom clearance for condition-1 has been derived using Mat lab software as
\[
c = 0.0010d^2 + 0.0762d + 3.2733,
\]
where \( c \) = Clearance in mm in the Bottom direction. \( d \) = No. of hours worked in 1000. The curve fitting equation for the Bottom clearance for condition-2 has been derived as
\[
k = -0.0018l^2 + 0.1128l + 3.7397,
\]
where \( k \) = Clearance in mm in the Bottom direction and \( l \) = No. of hours worked in 1000. The clearances obtained by solving these curve fitting equations are compared with the measured clearances and are shown in Fig. 6.4.
Equation for Condition 1: \( e = -0.0025f^2 - 0.0290f + 2.8006 \)
Equation for Condition 2: \( m = 0.0018n^2 - 0.1258n + 3.0740 \)

Fig. 6.5 Runner vane clearance at left hand side

The equation for the runner vane LHS clearance for condition-1 has been derived using Mat lab software as \( e = -0.0025f^2 - 0.0290f + 2.8006 \), where \( e = \) Clearance in mm in the LHS direction, \( f = \) No. of hours worked in 1000. The curve fitting equation for the LHS clearance for condition-2 has been derived as \( m = 0.0018n^2 - 0.1258n + 3.0740 \), where \( m = \) Clearance in mm in the LHS direction and \( n = \) No. of hours worked in 1000. The clearances obtained by solving these curve fitting equations are compared with the measured clearances and are shown in Fig. 6.5.
The equation for the runner vane RHS clearance for condition-1 has been derived using Mat lab software as \( g = -0.0025h^2 - 0.0290h + 2.8006 \), where \( g \) = Clearance in mm in the RHS direction, \( h \) = No. of hours worked in 1000. The curve fitting equation for the RHS clearance for condition-2 has been derived as \( O = 0.0012p^2 - 0.1091p + 2.8805 \), where \( O \) = Clearance in mm in the RHS direction and \( p \) = No. of hours worked in 1000. The clearances obtained by solving these curve fitting equations are compared with the measured clearances and are shown in Fig. 6.6.

**6.5 Pressure Variation Inside the Discharge Ring**

A flow for which the streamlines are concentric circles is called a vortex. Fluid dynamics have identified two types of vortices that can be easily described mathematically. One is called the forced vortex in which the velocity increases linearly from the centre of rotation. The other type of vortex is the free, or potential, vortex. In the free vortex, the product of the velocity at a point and the radial distance from the vortex, centre to that point is a constant \( (Vr = C) \).

Flow patterns that are developed by – flow inside the discharge ring is an irrotational flow and this flow is closely approximated. The flow of water in this...
region where the streamlines are converging usually approximates irrotational flow quite closely. However, in regions where boundaries turn away from the flow so as to cause the streamlines to diverge, the flow usually separates, from the discharge ring and a recirculation pattern is generated in this region. This phenomenon is called separation. In the region between the high-velocity flow outside the zone of separation and the low velocity zone inside it, vortices are formed. These vortices are often called eddies. These vortices or eddies lead to the phenomenon called turbulence.

As the flow passages in the bulb turbines are converging, the flow is irrotational. Hence Bernoulli’s equation is used to obtain the pressure variation between points in the flow field, including points adjacent to the boundaries. Bernoulli’s equation between the reference point and any other point is

\[
\frac{p}{\gamma} + \frac{V^2}{2g} + z = \frac{p_0}{\gamma} + \frac{V_0^2}{2g} + z_0
\]  

(6.3)

Where \( P_0, V_0, \) and \( Z_0 \) are pressure, velocity, and elevation, respectively, at the reference point; and \( p, V, \) and \( z \) are pressure, velocity and elevation at any other given point.

By simple rearrangement, Eq.6.3 is written as

\[
p - p_0 = \rho(z_0 - z) + \frac{\rho}{2} (V_0^2 - V^2)
\]  

(6.4)

Equation expresses the pressure change in terms of the change in hydrostatic pressure (the first term on the right) and the change in kinetic pressure (the second term on the right). Thus the dynamic effect is given by

\[
\left( \frac{P}{\gamma} + z \right) - \left( \frac{P_0}{\gamma} + z_0 \right) = \frac{\rho}{2\gamma} (V_0^2 - V^2)
\]  

(6.5)

Equation (6.4) expresses the change in piezometric head as a function of the difference in the velocities squared, and it reduced to

\[
h - h_0 = \frac{V_0^2 - V^2}{2g}
\]  

(6.6)

where \( h \) is the piezometric head at a given point, and \( h_0 \) is the piezometric head at the reference point.

Hence,

\[
\frac{h - h_0}{V_0^2/2g} = 1 - \left( \frac{V}{V_0} \right)^2
\]  

(6.7)
Because the velocity is inversely proportional to the cross-sectional area through which flow occurs in a flow passage (that is, \( V/V_0 = A/A_0 \)), we can express the relative pressure distribution or piezometric – head distribution in terms of the dimensions of the flow passage. For two-dimensional flow, the streamline spacing is directly proportional to the flow area. Thus we have \( V/V_0 = n_0/n \) for the relationship between the relative-velocity distribution and the relative-stream-line spacing. Here \( n \) is the distance between two adjacent streamlines measured along the line (possibly curved) perpendicular to both streamlines.

As the flow pattern in a bulb turbine is symmetrical with either the vertical or the horizontal axis through the center of runner boss of bulb turbine, the pressure distribution on the surface of the runner boss, obtained by application of Bernoulli's equation, is also symmetrical. The relative pressure \( C_p \) is plotted outward (negative) or inward (positive) from the surface of the runner boss, depending on the sign of the relative pressure and on a line normal to the surface of the cylinder. It should also be noted that \( p_0 \) and \( V_0 \) are the pressure and velocity of the free stream far upstream or downstream of the body.

Thus it is seen that the points at the front and rear of the runner boss are points of stagnation (\( C_p = +1.0 \)) and that the minimum pressure (\( C_p = -3.0 \)) occurs at the midsection where the velocity is highest. When a fluid particle travels around the runner boss of bulb turbine, the fluid particle first decelerates, which is consistent with the increase in pressure, then it is accelerated to its highest speed by the action of the pressure gradient; that is, the pressure decreases over the entire path. Next, as the particle travels further against the adverse pressure gradient (pressure increases in the direction of flow here), the particle accelerates to the free stream velocity in its passage.

Cavitation occurs when the pressure at any point in the bulb turbine is reduced to the vapor pressure of the liquid. Under such conditions, vapor bubbles formed and then collapse (condense), producing dynamic effects that lead to decreased efficiency and deformation of discharge ring.
Therefore, bubbles collapse close to physical boundary of discharge ring in bulb turbine causes damage and deformation and this damage occurs in the form of a fatigue failure brought about by the action of millions of bubbles impacting (in effect, imploding) against the surface material over a long period of time, thus producing pitting of the material in the vicinity of the zone of cavitation.

The turbines at a particular power house (PH III) are subjected to the maximum internal water pressure and water hammer forces. These forces are compared to other units in other power houses. The discharge rings of these units are deformed. The discharge rings of the other units are also in strained. The internal pressure developed during the water flow and the water hammer pressure during shutdown of the machine are added to the cavitation pressures forces inside the discharge ring of these bulb turbines. Whenever, all the three forces acting inside the discharge ring exceeds the yield stress of the material, the discharge ring is likely to deform. The different forces acting the discharge are shown in Figs. 6.7 - 6.9.

Equation: \[ q = -0.0003r^2 + 0.1592r + 0.0399 \]

Fig.6.7 Hydraulic thrust on runner vanes

The hydraulic thrust acting on the runner vane is shown in Fig. 6.7. equation for the hydraulic thrust is derived by using Matlab software as \( q = -0.0003r^2 + 0.1592r + 0.0399 \), where \( q \) = Hydraulic thrust and \( r \) = quantity of water flow.
The Lift force acting on the runner vane is shown in Fig. 6.8 and equation for the Lift force is derived by using Matlab software as $s = -0.0056t^2 + 2.7078t + 0.3035$, where $s =$ lift force and $t =$ quantity of water flow.

Equation: $u = 0.0002v^2 + 0.018v - 0.0477$

Fig.6.9 Drag force on runner vanes
The Drag force acting on the runner vane is shown in Fig. 6.9 and for the Drag force is derived by using Matlab software as 
\[ u = 0.0002v^2 + 0.018v - 0.0477, \]
where \( u \) = Drag force and \( v \) = Quantity of water flow.

The lift force acts in the vertical direction of the discharge ring whereas the drag force acts horizontally. The lift force, drag force and hydraulic force are maximum for the maximum discharge of water flowing inside the discharge ring and cause the deformation of the discharge ring. The cavitation forces act in the discharge ring during low discharge of water inside the discharge ring. The combinations of these forces are responsible of the deformation of the discharge ring.

6.6 Conclusions

The turbine manufacturer recommended that the unit should not be run below 50% wicket gate (guide vane) due to excessive cavitation levels. This warning has been confirmed by measurements taken on the bulb turbines by this experiments also. It has been also found that the turbine experienced higher cavitation acoustic emission levels when the guide vane opening is less than 50%.

It has been found that energy modulation levels at the blade passing frequency for the hydro turbine indicate the reasons for severe erosive cavitation occurring when the hydro units are operated at guide vane openings below 50 percent and above 90 percent. Cavitation damage is minimal at around 60 percent guide vane opening. This is confirmed in this study by the vibration and noise measurements taken on the discharge ring for different guide vane openings from 30 to 90%. In this study it has been found that the tail race water level has influenced the cavitation occurrence in the bulb turbines which are located at different places with different suction head \( (H_s) \).

The deformation of discharge ring is calculated and correlated with the hours of operation. This is a very useful method to predict the life of discharge ring and for scheduling the repairs and replacement at appropriate time on bulb turbines. These findings also help the designer to take care of cavitation phenomenon and deformation of discharge ring by using these equations for bulb turbines.