Research Paper

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Vibration Control and Condition Monitoring of Turbine Alternator in Hydro Electric Power Plants

Engineering

KEYWORDS : Bulb turbine, Cavitations, Synchronous Generator, Vibration Analysis, Air Injection Setup

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ABSTRACT

Handling of large power generating units and keeping the size of the turbine generator sets are economically small that requires higher running speeds, which means greater pressure differences between the two sides of the vanes of a runner. These forces also cause vibrations of the stator which are transmitted through the frame to any adjoining structure, cavitations due to discharge causes the main vibration effects in most of the hydroelectric power plants. Vibration faults in the generator subject to stresses beyond its design limits and cause high temperature increase, amplify and distort air gap torques, and create unbalanced flux densities. Alternating magnetic forces in the air gap between the stator and rotor of the machine are essential for the working of the machine. An air injection setup is introduced to control cavitation bubbule and for smooth operation of the turbine Generator sets.

I.INTRODUCTION

The vibrations are detrimental to the desired running of the machine. Vibration occurs in every machinery as a consequence of forces which are generated in such machines. These could be due to a variety of causes, such as rotor imbalance, misalignment, faulty bearings or gears, eddy currents, electromagnetic forces etc. These forces act through the bearings to move the stationary parts of the machine. At certain frequencies machine vibrations also result in the transmission of unwanted tonal noise. In a commercial situation tonal noise is undesirable.

Performance of hydro generator depends on various characteristics of the machine and the efficiency can be written as

$$\eta\ HG = \frac{\text{Electric Power delivered}}{\text{Mechanical Input}}$$

$$\eta\ HG = \frac{\text{Electric Power output}}{\text{Electric Power output} + \Sigma\ \text{Losses}}$$

P_{stray} and P_{mech} subjected to stray load losses are orderly ferrite material ,mechanical ,Short circuit ,excitation, brush and ring stray powers.

Power potential is a function of head and discharge quantity of water at any point of time and is determined by using the following formula.

$$P = \rho\ g\ Q\ H\ \eta_0 / 1000 \qquad (1)$$

Where P -power output in kW, ρ -density of water kg/m^3, g-acceleration due to gravity m/s^2,Q-discharge in m^3 / s,H - net head in meters,η_0- overall unit efficiency in percentage

$$P_g = 9.81\ \eta_t\ \eta_g\ \gamma\ Q_t\ H_t\ (W). \qquad (2)$$

where P_g - generator output power,η_t - turbine efficiency,η_g - generator efficiency,Q_t -discharge through turbine, H_t - Head on the turbine,γ- specific weight of water

If the pressure at any point in the vane reaches the critical pressure for causing cavitation, the vane gets damaged. The average pressure in a hydraulic turbine runner falls from inlet to outlet. Vibrations are induced in the hydraulic turbines due to intense pressure fluctuations caused by the growth and collapse of bubbles during cavitation this may cause change of flux distribution in the srator fluxes

The stator of a radial flux external stator electrical machine is a stack of laminations with a ring shaped back-of-core and teeth that protrude radially inwards. Currently all stator windings are powered by poly-phase power supply which is most commonly three phase. Oscillating electromagnetic forces are generated in the air gap of the machine between the rotor and the stator and in the stator and rotor iron. The electromagnetic forces that act across the air gap on the stator teeth are Maxwell forces. The magnetostriction effect causes the electromagnetic forces that act within the stator iron. The oscillating Maxwell forces and the magnetostriction effect deform the stator and cause vibrations. Noise is predominantly caused by unwanted sound radiating through air from a vibrating surface. Noise can be transmitted as pitched or unpitched and the Maxwell equations are listed below.

$$\nabla \times \mathbf{h} = j + \frac{\partial d}{\partial t} \qquad \oint \mathbf{h}\, dl = \int \left(j + \frac{\partial d}{\partial t} \right) ds = I_{total} \qquad (3)$$

$$\nabla \times \mathbf{e} = -\frac{\partial b}{\partial t} \qquad \oint \mathbf{e}\, dl = -\int \frac{\partial b}{\partial t}\, ds = \theta \qquad (4)$$

A stator is a cyclically symmetric structure, and this property occurs because a stator can be divided into subsections, with every one of the subsections being identical to every other subsection. Incidentally, each subsection includes one tooth, a section of the windings and the corresponding segment of the back of core. Cyclically symmetric structures exhibit unique vibration characteristics and these characteristics are described briefly here. The mode shapes of acyclically symmetric structure can be described by a mode number **n,** the mode number describes the number of complete deformation waves around the circumference that the structure deforms into elliptical shape.

II.ELECTROMAGNETIC NOISE REDUCTION METHODS

Existing methods for reducing electromagnetic noise often focus on

- Controlling forcing frequencies so that they do not coincide with resonant frequencies.

- Reducing the magnetic loading on the machine.
- Reducing the susceptibility of the stator to forced vibrations by changing the geometry and properties of the stator or rotor.
- Utilizing the control available through power electronic converters.

Knowledge of the amplitudes and frequencies of the flux density harmonics in the machine and the forces associated with these are required before alterations to the harmonics can be carried out. Varying the supply current will alter the resultant forces in the machine and make it possible to avoid exciting some of the resonances in the machine

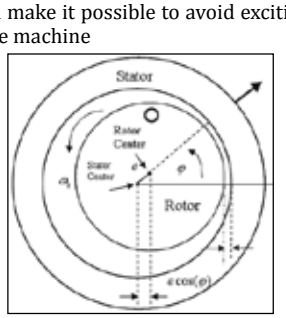

Fig 1. Eccentricity Circle of Stator and Rotor

III. DIRECT AND QUADRATURE AXIS FLUXES

The direct axis is defined as the direction along the rotor that the field winding current causes magnetic flux to flow. The quadrature axis is defined as the axis located π/2 electrical radians behind the direct axis of the rotor shown in Figure.1. The stator internal voltage is that portion of the terminal voltage due to the flux caused exclusively by the field current Xd and Xq will vary with respect to the Eccentricity of stator and rotor interactions as shown in Figure1.

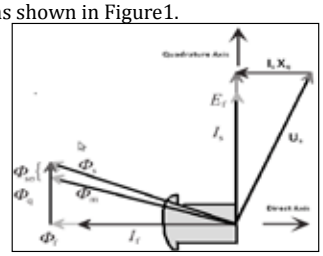

Fig.2.Direct and quadrature flux distribution

And the power delivered by the synchronous generator is given by

P=

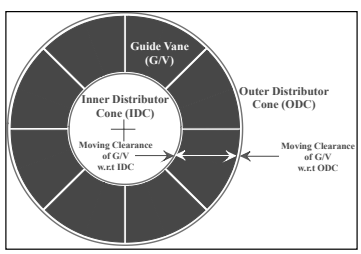

Fig 3.Turbine ODC and IDC

$$\left[\frac{EqVa}{Xd} \sin(\delta) + \frac{Vd}{2} \left(\frac{Xd-Xq}{XdXq}\right) \sin(2\delta) \right] \quad (3)$$

$$\left[\frac{EqVa}{Xd} \sin(\delta) + \frac{Vd}{2} \left(\frac{Xd-Xq}{XdXq}\right) \sin(2\delta) \right] \quad (5)$$

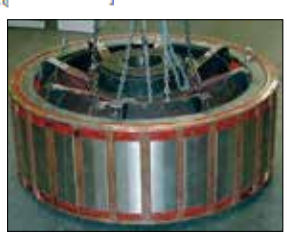

Fig.4. Permanent Magnet Rotor

IV . VIBRATION MEASUREMENT AND ANALYSIS

Vibration can be measured using displacement, velocity and acceleration transducers that convert machine motion to electrical signals. From these, the machine's signature can be generated and analysed Machines generate an array of vibration signals at different frequencies which can be seen and analysed in the frequency spectrum of the signals. These are basically plots of amplitudes versus the various frequency components and represent machine motion as a function of frequency. On the other hand, a time plot of the signals is used to determine the resultant vibrations magnitudes of the various frequencies in the frequency spectrum and represent machine motion as a function of time.

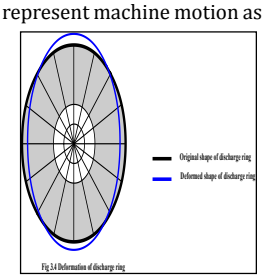

Fig.5. Deformation of discharge ring

$$P = 3\left[\frac{EqVa}{Xd} \sin(\delta) + \frac{Vd}{2} \left(\frac{Xd-Xq}{XdXq}\right) \sin(2\delta) \right] \quad (5)$$

The imbalance can be controlled by simple field-balancing techniques, there are some cases where balancing does not seem to work very effectively. Such cases often pose challenge to the engineers and might require the services of outside experts, and yet sometimes the solutions can be amazingly simple. In most cases, the technique of Frequency Analysis of Vibration (also known as Spectrum Analysis) is essential to get to the root cause of the vibration. This technique is based on the fundamental concept that if there is a vibration at a certain frequency, its cause must also be at the same frequency. Thus frequency information of the vibration is essential to fault-finding. A wealth of information exists on the internet and otherwise on interpreting the results of Frequency Analysis.

V.DYNAMIC BEHAVIOUR TESTS

- Measurement and spectrum analysis of bearing vibrations (Turbine/Generator).
- Measurement and spectrum analysis of shaft vibrations along with orbit analysis (Turbine/Generator).
- Vibration measurement in penstocks.
- Speed rise and pressure rise at load throw off.
- Looseness of wedges, spacers and bindings.

In addition to above diagnostic testing's like polarization index, Tan delta and tip up tests, Dielectric loss measurement, Partial discharge measurement and winding resistance measurement and AC/DC H.V. test can be carried out on the stator winding.

Stator Core

Stator core is inspected for major external changes such as:

- Looseness of core laminations.
- Mechanical damages.
- Locally overheated spots and core burning.
- Deposits in cooling/ventilating ducts. core flux test should be carried out to check the shorts in the core and its laminar resistance.

Rotor

The rotor is inspected thoroughly for:

- Signs of overheating.
- Cracks in shaft end or in body.
- Damages/cracks in coupling system.

frequency, which is the exact indication of the balance between supply and demand. The operator in the control room, on receipt of orders from the central load dispatch office, adjusts the speed level of the unit to assist the system to maintain frequency at 50 Hz. In the case of units fitted with automatic load-frequency control device, the speed level is adjusted by load frequency control equipment and even by the load dispatch office itself shown in figure.4.

The cavitation process is inherently an unsteady one and may involve large fluctuating forces. If one of the frequency components of these fluctuations matches a natural frequency of a portion of the equipment, vibration may result. Cavitation causes blade vibration in hydraulic turbine runners. The vibration is usually of fairly high frequency, ranging from several hundred to several thousand cycles per second. Another important effect is that cavitation can lead to system instability.

The parameter most widely used to denote the state of the cavitation is the Thoma cavitation factor

$\sigma = (h_a - h_{vp} - h_s)/H$ Where h_a is atmospheric pressure in meter of water column

h_{vp} is vapor pressure of the flowing water in meter of water column

H is head on the turbine in meter of water column

$$B = \sum_{i=1}^{12} 9.81 \times 24 \times 30 \eta Q_i h_i \mu \quad (6)$$

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Where η = overall efficiency,μ = energy tariff (Rs/kWhr),Q_i = the monthly discharge (m^3/s), h = head on the turbine (m)

A method is suggested for air injection in a bulb turbine to avoid cavitation, to check its performance and reduction in vibration and noise levels. Vibration analysis of machine stability and foundation are also made with an experimental setup with specifications as shown in Table 1.

TABLE 1. SPECIFICATIONS TURBINE AND GENERATOR

Sl. No.	Description of Parameters	Data
1.	Rated output	15600 kW
2.	Rated Discharge	271 m^3/s
3.	Rated Head	6.5 m
4.	Rated Speed	75 rpm
5.	Over Speed	104 rpm
6.	Runaway Speed	205 rpm
7.	Specific Speed	902
8.	No. of Guide vanes	16
9	No. of Runner vanes	4
10.	Diameter of runner	6250 mm

VI.EXPERIMENTS FOR CAVITATION PREDICTION

Experiments are carried out on the bulb turbine to predict the plant cavitation factor critical cavitation factor and cavitation factor at bubble formation. A correlation has been derived between the specific speed of the turbine and the above variables for the cavitation prediction and for the formation of cavitation bubbles. The experiment results are tabulated in Table .2.

VII.CAVITATION AT RUNNER VANE MOVING ZONE

The bulb turbine units are used in the run-of-river applications for low head ranges. The runner vanes and guide vanes are synchronized to produce the maximum turbine efficiency for a given head and flow conditions. The upstream river geometry significantly influences the flow pattern to the turbine intake. The improper orientation of the turbine with respect to the direction of flow in the river has resulted in flow separation. Water hammer forces are found to act on the guide vane assembly during sudden load rejection and adversely affects the turbine performance.

The runner vane assembly with runner boss and runner cone. The discharge ring connects the distributor cone assembly and draft tube. As the velocity of the water leaving the runner vanes is higher due to high specific speed, the water boils inside the discharge ring due to lower vapour pressure of water inside the discharge ring. As the growth of bubbles and their collapse take place at a high speed, the damage to the discharge ring is large and the damaged metal surface appeared with numerous holes of different diameters varying from The cavitation occurs at the discharge ring of the bulb turbines. The discharge ring assembly houses 3 - 6 mm with depth of penetration from 6 -10 mm.

TABLE 2. MODEL TEST RESULTS FOR CAVITATION FACTORS

Specific Speed (N_s)	Plant cavitation factor (σ_p)	Critical cavitation factor	Cavitation factor at bubble formation
		By model test report(σ_{ct})	By model test report (σ_{bt})
680	0.875	0.95	1.26
687	1.704	0.95	1.26
688	0.967	0.95	1.26
753	0.935	1.00	1.28
758	2.048	1.00	1.26
760	1.015	1.00	1.28
802	1.322	1.43	1.77
811	1.437	1.43	1.77
814	1.758	1.43	1.77
896	1.549	1.69	1.98
899	2.172	1.69	1.98
902	1.69	1.69	1.98

The discharge rings have deformed almost in all the eight machines with increase in moving clearances of runner vanes at the top and bottom and decrease in clearances at left and right sides. The circular shape of the discharge ring has been distorted, buckled and has become elliptical in shape as shown in Fig 5. The detail on the deformation of discharge ring is shown in Table 2.

X.CAVITATION AT GUIDE VANE MOVING ZONE

The bulb turbine is a horizontal shaft axial flow turbine. The machine delivers 16.5 MW of power under a gross head of 7m at a discharge of 270m^3/sec. To avoid the cavitation, the turbine runner centre is kept at a depth of 5.0 m below the tailrace water level. The diameter of the runner is 6.250 m, therefore the suction head 'h_s' on the machine is 1.875 m at the minimum tailrace water level and is 4.375 m at the maximum tailrace water level.

XI.DEFORMATION OF OUTER DISTRIBUTOR CONE

The guide vane clearances are measured in all the eight turbines. The clearances measured on the inner distributor indicate a maximum gape of 5 mm at guide vanes 1and 16 at the top and between guide vane 8 and 9 at the bottom. The left hand and right hand side guide vane clearances measured at guide vanes 4, 5 and 12, 13 respectively show almost zero mm. The contact clearance between guide vanes 3,4,5 and 11,12,13 has increased abnormally to an average of 10 mm from the designed value of zero mm. Pressure at the prevailing temperature, water boils and small bubbles or cavities form in large numbers. The appearance of cavities at a place depends on the difference between the atmospheric pressure and the vapour pressure.

AIR INJECTION IN BULB TURBINE

When the water flows into a region where the pressure is reduced below its vapour pressure. The cavitation occurs at the discharge ring of the bulb turbines in this case study. The most effects of Air injection study has been carried out in the bulb Admission of compressed air to the low-pressure regions cushioned the cavity collapses and reduced its destructive effects. The effect of introducing air at the cavitation regions was to break the vacuum there, thus raising the pressure and to eliminate the cavitation. The air introduced in this manner forms large bubbles, which are compressed downstream in the high-pressure region but are prevented from collapsing by the cushioning effect of the large quantity of air. The air injection in to the runner vane moving area of the bulb turbine as shown in Figs.6 reduces the vibration by 50% and the noise level has been reduced by 10 dB. The readings are shown in Tables 3 and 4.

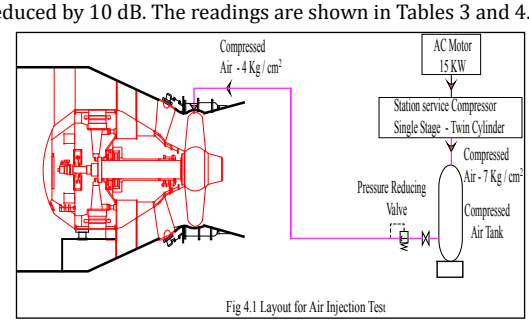

Fig.6.Air Injection Setup

XII.EXPERIMENTAL SETUP FOR AIR INJECTION TEST

A air injection study has been carried out in the bulb turbines to check their performance and to find out the reduction in vibration and noise levels. The four numbers of 20 mm diameter copper pipes meant for runner vane cooling water for condenser operation are used for air injection into the turbine. In order to check the air requirements inside the discharge ring during the operation, the dummy flange has opened from the head and it has been found that water flows freely at 30% guide vane opening. But as soon as the guide vane opening is increased from 30% to 34%, the free water flow stops and a metallic bell ringing noise with vibration are noticed. Therefore it is confirmed that a vacuum

is created inside the discharge ring at the time of cavitation, which has started with vibration and noise is shown in figure.6 .

Speed	Vibration measurements with existing stiffness	Vibration measurements with modified stiffness
	By actual measurements	By actual measurements
10	9.0	-
20	11.0	2.00
30	14.0	5.00
40	16.0	9.0
50	19.0	12.0
60	21.0	15.0
65	23.0	17.0
70	40.0	19.0
75	25.0	20.0
80	26.0	22.0
85	27.0	40.0
Operating condition	Dangerous	Safe

TABLE.3 VIBRATION AND NOISE LEVEL MEASUREMENTS TAKEN ON THE TOPE OF DISCHARGE RING IN VERTICAL DIRECTION

If the frequency of the dynamic forces and moments generated in a machine or structure is close to the natural frequency of the whole system, the phenomenon of resonance occurs which is experimental to the functioning of the machine or the assembly. Resonance induces excessive responses, leading to failures. Vibration control through proper design of machines and machine foundations becomes imperative for low maintenance costs, productivity, prolonged machine life, low noise and high comfort, low power consumption and above all good quality control listed in Table 3 and Table 4..

TABLE.4. TEST RESULTS

Description of parameters	Power House-1		Power House-2	
	U1	U2	U1	U2
Displacement peak-peak(D) (m)	220	210	215	225
Velocity peak-peak(V) (mm/sec)	13.00	12.50	12.60	13.20
f = 19098.55 * (V/D) (cpm)	112.8	113.6	111.85	111.97
G/V passing frequency	1120	1120	1120	1120

XII.CONCLUSION

The vibration induced during the mechanical spin of the turbine reveals that there is an unbalanced mass in the rotating system. Since the rotor is assembled at the power house site with the core plate stacking and pole assemblies there is an expected unbalanced mass in the generator portion. The unbalanced mass in the generator rotor causes an abnormal vibration when the turbine shaft attains it critical speed at 70 rpm. A resonance condition is created when the natural frequency of concrete structure coincides with the critical frequency of the rotating masses at the existing condition. Therefore a modification in the stiffness of the supporting structure for the bulb turbine is made to avoid the resonance condition in the operating range of the turbine. The analytical equation derived for the both the condition gives very close results compared with that obtained by actual measurements.

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