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Fluid-Structure-Interaction (FSI) Analysis of Francis Turbine for High Head Operations

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Abstract— Pakistan's major electric production is from hydro-turbines in which Francis Hydro-turbines are situated at Warsak, Ghazi Brotha, Terbela and Mangla. In the presented work Mangla power plant's Francis Turbine is analyzed for high head operations as head variations are usual throught the year in this reservoir. In floody conditions turbines have to operate at overload conditions sometimes non-designed conditions. These conditions are the causes of different dangerous effects effecting performance. In presented article safe mass flow rate zones are found for the maximum Head Water Level (HWL) which increased after wall raising project of Mangla reservoir from 1202 to 1240ft. Inlet pressure at Francis Turbine blade increased dut the increase in HWL the reason why Mangla power plant faces some cases of blade damage in Floody conditions after wall raising project. Analytical found flow rates for the increased head are also checked for Flow Analysis where the pressure distributions were in the normal range in comparision with the recent studies on Francis Turbine. ANSYS CFX was used for Flow Analysis with K-Ɛ turbulence model.

Index Terms— Francis Turbine, Off-Design Operation, BEP, HWL, K-Ɛ Turbulence model, Flow Analysis, FSI Aanalysis

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1 Introduction

WATER has a lot of potential to produce electricity with much constant voltage value. Hydro-turbines are usually used to operate at veriable load due to different climate nature over the whole annum. Pressure fluctuations usually occor at startup or partial load conditions and this disturb the efficiency of turbine due to the production of vortex rope. These oscillations are called Rheingans oscillation [1] and these are due to high frequency loadings [2]. Loadings are the reasons of vibrations which result in fatigue failures [3].

In Francis Turbines steady and unsteady loading are there. Steady loading are due to the fluid pressure and centrifugal force of runner and unsteady loadings are due to pressure fluctuations and vortex rope phenomenon in draft tube. Fatigue failure occurs in different stages like: (1) changes in microstructure; (2) microscopic cracks formation; (3) microscopic flaws growth (dominant cracks); (4) dominant macro-crack propagate stably; (5) instability of structure/complete fracture. Nucleation phenomena depend upon the microstructure, environmental and mechanical factors [4, 5].

Speed more than a hundred rpm costs some millions a day. Once crack formed in high cycle loadings it can cause catastrophic failures before the designed life of the turbine. [7]

The wall raising project of Mangla reservoir increased almost 15% of its HWL which is much high from the designed turbine head. Operation of turbine at this high head level is risky for blade. Safe range flow rates for the extreme HWL must be fount out in order to keep turbine safe. Recent studies of turbo machinery are used to find flow rates. Moreover validation of extracted flow rates is carried out by Flow Analysis in ANSYS CFX.

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2. Methodology of Research:

2.1 Francis turbine runner

The experimental setup consists of medium speed Francis Turbine with specific speed ν = 0.41. 13 blades runner with the radius $R2e$ = 1.1455m. The runner, a welded constructed and blaed welded to the casted ring and crown.

Sub-models of whole rummer are created in 3D and all sub-models (Crown, Band and blade) are assembled together to get overall runner assembly. All the dimensions of runner sub-models are taken from manul drawings available from mangla power station. Pro-engineering 4.0 is used for the modeling purpose. Meshing is created in ANSYS TurboGrid 14.0 tool for flow analysis CFX 14.0 is used. K-epsilon turbulence model is used to overcome the turbulent stresses. Geometric models of band, blades and crown are shown in the assembly in Figure 1.

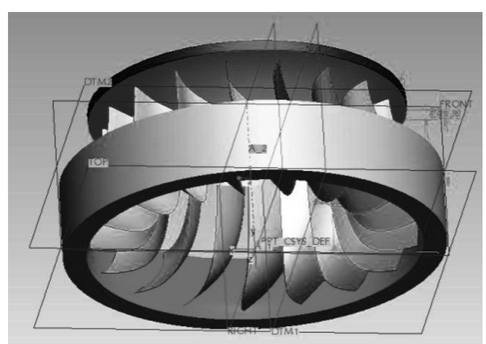

Figure 1: Francis Turrbine runner model

2.2 Analytical Modeling

Considering total raised inlet pressure at blades due to in-

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creased HWL at reservoir, safe flow rates should be found out in order to utilize turbine for long life. For achieving this goal recent studies available from literature of turbomachinary [3], [6], [20] and [21] are implemented. Extracted mathematical and experimental flow rates with efficiencies are displayed in Fig.6 and Fig.7 respectively.

2.3 Flow Simulation

Runner model was imported in TurboGrid CFX tool with 162300 tetrahedral elements and 125456 nodes. Meshing involves these steps and shown in Fig.2-5

(i) Blade Topology
(ii) Shroud Topology
(iii) Hub Topology
(iv) Final Topology and
(v) Final Topology

Using determined flow rates as input and using boundary conditions, the pressure distributions in flow analysis in CFX tool is shown in Figure 5. The maximum pressure is at the top of trailing edge of pressure side resembling the past studies as well.

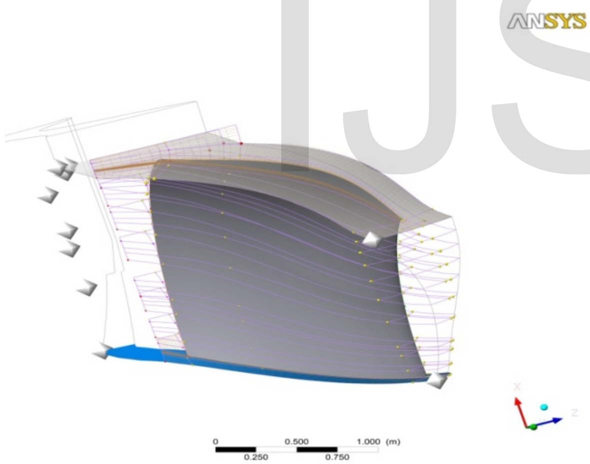

Figure 2: Blade Topology

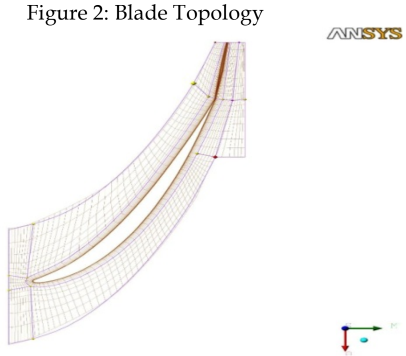

Figure 3: Shroud Topology

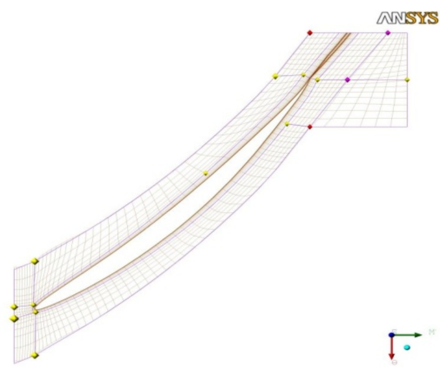

Figure 4: Hub Topology

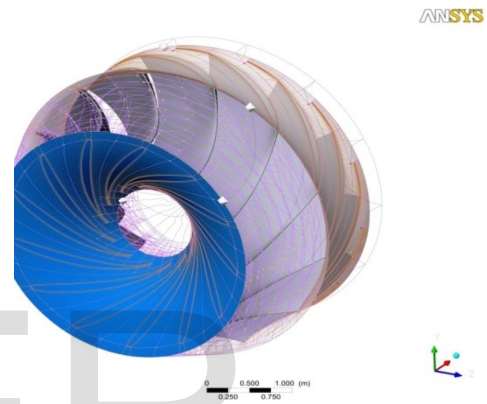

Figure 5: Final Topology

3. Results and Discussions

3.1 Analytical Results

Taking recent studies available for the turbomachinery the found-out mathematical results, Discharge Q, and efficiency η are checked with experimental results in Fig.2 and Fig.3. In order to get 115MW the required discharge will be lower for high HWL. This is the reason discharge graph goes down as HWL increased. The frictions between running parts of runner define the gap between two graphs. Due to resistances we have to putt some extra input hydraulic power to produce 115MW as compared to mathematical results. This is the main reason there is a off set between mathematical and experimental results graph.

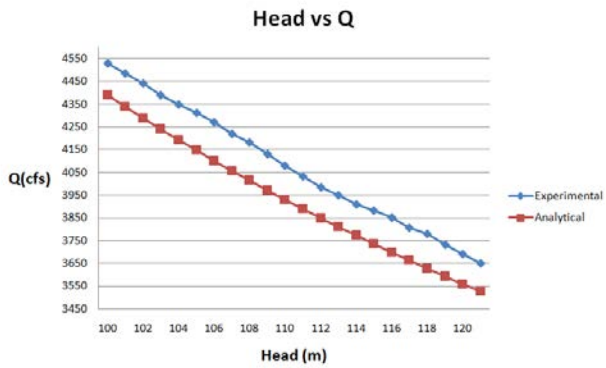

Figure 6: Head vs. Discharge

Efficiency graph also defines a gap between mathematical and experimental graphs. There is also the same reason as was in the Head vs Discharge graph that due to the overcoming of frictions between different parts of runner some large input hydraulic effort is requird to overcome it, hence the experimental efficiency is at a lower level off set of the mathematical graph. Some randome operational mistakes of operator are also there for not opening the required wicket gate angle and hence dropping the experimental efficiency at some instances.

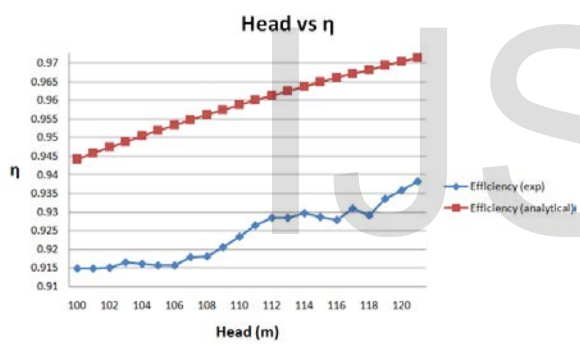

Figure 7: Net Head VS Efficiency

3.2 Flow Simulation Results

Using literature available from turbomachinary field the requird flow rates are carried out and using these mathematical flow rates as input with boundary conditions in CFX ANSYS tool pressure distributions are extracted throught the span of whole blade. Distributed pressure is shown in the graph Figure 5. As relating to recent studies maximum pressures are being observed at the top trailing edge of pressure side which also validates the available studies available for Francis Turbine Turbomachinary. Some of the studies i.e; Zoran Crija at al.2008 showed a study with maximum pressure of 1.2MPa that has very close conformity with the presented study that has 0.81MPa [24]. There is another study of Carija at al in which they showed the maximum pressure of 0.85Mpa that is in very close confirmation with presented one.

D Frunzăverde1, 2010, extracted maximum pressure value of 0.746MPa, a little below than presented work, is also a good comparison plateform for presented study. Presented study and recent studies are caompared in Table 1.

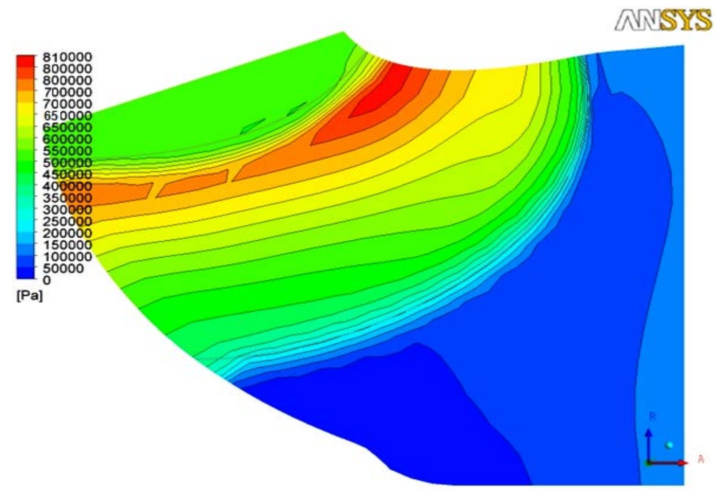

Figure 8: Pressure Distribution (Meridional View of Blade)

Contour of pressure at 20% span and velocity vector contour is also shown in Figure 9 and Figure 10.

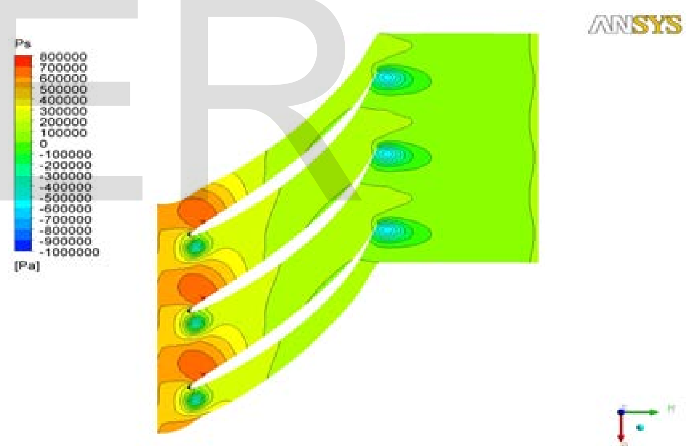

Figure 9: Contour of Pressure at 20% Span

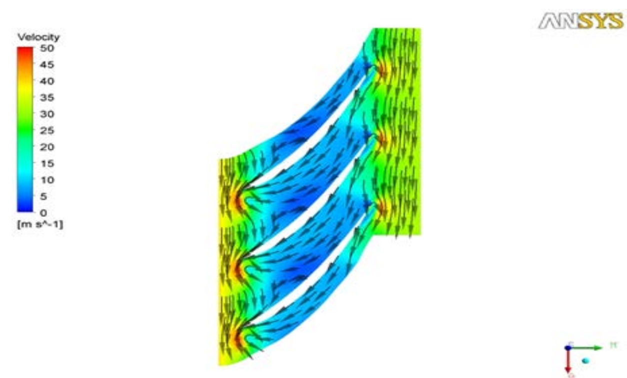

Figure 10: Velocity vector

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4. Conclusions

In this presented study only for the static loadings the analysis was carried out. The results are in very close conformity with the recent studies available for turbomachinary, Francis Turbine. Friction at different parts of the turbine crown, band, gates, penstock, tail-stock etc are not considered here and for good results these should be taken into account before recommending results to the Mangla Power Plant department. Also a complete structural and fatigue analysis is also required for the detailed study and to find the effects close to real time in dynamic loading conditions.

Table I: Comparison of present and previous results

	Presented Study	**José. [28]**	**R. Negru et al. 2012 [29]**	**Zoran Crija at al.2008 [24]**	**XIAO Ruofu et al. 2008. [30]**	**Z. Čarija et al. 2003. [31]**	**D Frunzăverde1, 2010 [32]**
CFD Max Pressure Result (MPa)	0.95	0.42	0.65	1.2	N/A	0.85	0.746
Max Pressure Location on Blade	Upper Trailing Edge (UTE)	UTE	UTE	N/A	UTE	N/A	UTE

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