Analysis of Fatigue Failure of Francis Turbine Runner at Derbendikhan Hydropower Station

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Abstract

Fatigue failures in Francis turbine runner are frequently occurred in hydropower plants, causing unexpected plant downtime and considerable financial loss. Decades of operational experience have shown that turbine runners develop fatigue cracks in areas, where stress concentrations and material defects coincide. In Francis turbine runners, cracks tend to propagate from the transition of the welded T-joint between the blade and the band or crown. This type of turbine runner, which operate under a wide range of heads and outputs, are subjected to considerable dynamic forces which can lead to fatigue cracking. The magnitude of these forces is a function of the hydraulic pressure, the water velocity and the geometry of the stationary parts guiding the water into the runner.

This paper presents the water pressure and bending stresses for different operational conditions in a Francis runner of a Derbendikhan hydropower station. At the first step, the dynamic fluid calculation is used to determine the fluctuating water pressure on the blade of the runner. At the second step, for a period of operational nominal bending stress due to the fluctuating water pressure are determined. The result indicates that the hydropower plant was operated with the high fluctuating water pressure which is responsible for inducing bending stress response. The high bending stress on the blade is a probable reason for fatigue failure. Furthermore, the paper discusses the fatigue analysis of the runner. Typical results are presented and discussed.

Keywords: Fatigue Failures, Francis Turbine Runner, Dynamic Fluid Calculation, Nominal Bending Stress.

1.1. Introduction

Development of fatigue cracks today is the major concern with regards to the structural integrity of Francis and reversible pump turbine runners. Fatigue crack usually initiates in a region of high stress at some metallurgical or structural discontinuity and if critical conditions of operation sustain, the crack may grow to failure. The failure mechanism is considered to be a combination of low-cycle (start-stop cycles) and high-cycle (hydraulic load fluctuations) fatigue. Fatigue cracks in the blade of the runner tend to occur either very early in life or after 10-20 years of operation. Blade life is mainly influenced by the static and dynamic stress fields on the blade, fatigue properties of the blade material, loading history and the environment of operation [1, 2].

A Francis turbine runner experiences steady and unsteady service loads. During the complete start-stop cycle, the steady loading goes from zero to a maximum under service condition and back to zero. Whereas hydropower station for pure-base load duty may only experience one start stop cycle per year, other station might experience up to ten start-stop cycle per day or more than 50 000 cycle in 20 years of service. Provided that steady loads are high enough, these cycle may initiate and propagate fatigue cracks from manufacturing defects. The size of the crack becomes critical, when the stress intensity range associated with unsteady flow causing vibration stresses exceeds the threshold value. The high cyclic fatigue (HCF) amounts to several millions per day. The crack my therefore grow to become unstable in a very short time compared with design life of the turbine runner. Each load case induces a nominal bending load to the runner blades. More reliable values for the static and the dynamic stresses under different operating conditions, will improve the basis for mechanical design of Francis and reversible pump turbine runners [1].

In general, when materials subjected to high stress levels beyond the yield point, show plastic behaviour, i.e. when all the forces acting on the body are removed, the body does not return to its original shape, a permanent deformation or a permanent strain is introduced in the material. For instance, if a specimen loads beyond the yield stress in uniaxial tension, then unloads and reloads it, the new yield stress point in compression is going to be smaller in magnitude than the original one, this is known as Bauschinger effect, which is believed to be due to residual stress in the microstructure [9].

Fatigue failures are influenced by many factors. These factors include stress concentrations, environment, loading, and residual stresses. While all four factors listed are worthy of researchers' attention, only the effect of residual stresses on the fatigue life is explored here. Residual stresses arise in runner blade as an unavoidable. It is commonly known that tensile residual stresses reduce fatigue life by encouraging early crack initiation. Strain ratcheting is defined as the phenomenon by which maximum strain progressively increases with fatigue cycles. This accumulation of strain leads to fatigue crack initiation, which in turn eventually leads to failure. Figure 1 present data for steel subjected to reversed cyclic loading. The strain histories used in these material tests, are representative of the strain history observed in actual structures [8].

Carpinteri et al. (2004) analyzed the fatigue behaviour of a T-joint blade in a hydraulic turbine runner both experimentally and numerically. They compared the numerical results deduced with those obtained from experimental tests and the agreement between the different results was quite satisfactory. Härkegard et al. (2000) suggested a design procedure for turbine runners, which consider the growth of a crack under start-stop cycles, until the crack becomes large enough to grow under vibration loading. In particular, they have analysed the fatigue crack growth in the transition between the blade and band/crown of a Francis turbine runner, where they have idealised the transition by a simple T-joint.

Besides the static pressure, the fluctuating part of the pressure signals at different locations of the turbine blades may be obtained and processed in order to predict the dynamic load in the turbine blades. Farhat et al. (2002) presented that the fluctuating pressure field appears as a result of a complex interaction of the flow through the rope and the wicket gates passing. Also, they provided that the knowledge of pressure fluctuation in different location of runner blades allows the estimation of the stress fluctuation to predict the turbine life cycle before fatigue cracking. Avellan et al. (2000) described a technique using miniature piezo-resistive pressure transducers embedded in the model runner blade, which has been successfully used to measure the mean and fluctuating pressures on the blade of the Francis model runners, and they transpose it to prototype conditions with application in the dynamic stress and fatigue analysis of the runner. Farhat et al. (2002) developed a specific procedure which allows fitting miniature pressure transducers in the runner blades without geometrical alteration. This paper present the study of the effect of improper operational conditions on the fatigue cracks in the runner of the Francis turbines. The equation developed by Huth, H. J. (2005) is used to fatigue analysis of the turbine runner blade. The Derbendikhan hydropower has been selected for this study.

1.2. Derbendikhan hydropower Station

Derbendikhan hydropower is operating under a constant speed and variable load as a power generation system. In this case, the hydropower is supposed to operate at constant speed, in reality they operate over a small range of speed due to variable grid frequency conditions. Since this hydropower station is subjected to wide range of pressure and frequently starting and shutting downs, they are subjected to variable mean and alternating stresses. The magnetite of the alternating stress depends on the load at given time.

During an inspection of unit 2 in Derbendikhan power station , in February 2002, it was discovered that some of the blades of the turbine had developed cracks. The runner of this turbine unit was subjected to six fatigue cracks, as shown in Figure 2.

Derbendikhan Hydropower Station is constructed in Iraq about 230 km northeast at Baghdad. The main data for the Derbendikhan Hydropower Station is shown in Table 1.

1.3.2. Data Analysis

The operational data of Derbendikhan Hydropower Station from May 1997 to February 2002 has been collected from the hydropower station which includes power output and net head. In general, the operating data for the hydropower station were registered for each hour.

The total numbers of operation hours for these seven years are more than 15000 hours. These data has been used to determine the fluctuation pressure on the turbine blade by using dynamic fluid calculation.
The turbines in Derbendikhan power station are the Francis turbine type in which the water inters the turbine through an arrangement of moveable wicket gates that control the water supply. From the main pipe which supplies water to the turbine, the water flows into spiral casing surrounding the turbine. From the spiral casing the water flows through the wicket gates and enter the runner.
The following equation is used to calculate velocity of flow (V_f) [7]:

$$Q = 0.95 * \pi * D * b * V_f \qquad (1)$$

Where Q [m^3/s] is the discharge and the following equation is used to determine the value of the peripheral velocity of the runner (v) [7].

$$v = \frac{\pi * D * N}{60} \qquad (2)$$

Velocity of whirl has been calculated by using the equation below [7]:

$$WHP = \frac{1000 * Q}{75}\left(\frac{V_w * v}{g}\right) \qquad (3)$$

And the value of the absolute velocity of water leaving the guides has been calculated by [7]:

$$V = \sqrt{V_w^2 + V_f^2} \qquad (4)$$

And the water pressure on the blade can be calculated by [7]:

$$\frac{P}{w} = H - \frac{V^2}{2 * g} \qquad (5)$$

Figure 3 show the load due to the water pressure on the turbine blade of the Derbendikhan hydropower station for the period of operation (1997-2002).

The steady fluid pressure on the runner depends on the water head and on the actual position of the guide vanes. At full load the largest nominal bending stress σ_b, on the blades may be roughly estimated by the equation [1]:

$$\sigma_b = 2 * b^2 * \Delta p * t^2 \qquad (6)$$

Where b is the blade height, i. e. the distance between ring and crown at the trailing edge, t is the blade thickness and ΔP the load range, i. e. the pressure on the blades due to the water head. Equation 6 is used to determine the nominal bending stress on the blade of the runner and the result has been shown Figure 4.

1.3. Result and discussion

Figure 4 shows that Derbendikhan hydropower station is operated for more than 15000 hours, and this hydropower station operated 77 hour at high stress levels greater than the yield stress. When the load on the blade is removed or decreased a permanent deformation or a permanent strain has been introduced in the material of the runner blades. The repetitions of the cyclic loads cause the accumulation of strain. This leads to fatigue crack initiation, which in turn leads to failure eventually. Figure 5 show the number of hours of operation for different bending stress intervals.
The above figure shows that the hydraulic turbine runner in Derbendikhan hydropower station is subjected to variable amplitude loading which can be defined by complex loading histories of varying cyclic stress amplitudes. It is believed that the main reason of failure is due to the 77 hours of operation at a high bending stress (more than the yield stress of 490 MPa).

1.4. Conclusion

This study has been aimed to investigate the improper operational conditions on the fatigue cracks in the runners of Francis turbines. The study is based on the observations of cracks in the hydraulic turbine runner of Unit 2 in Derbendikhan power station which are situated at the transition between the blades and crown/band. It has been shown that the failure occurred in some blades of the runner due to operation at high bending stress conditions for a relatively long period of time.

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تحليل الفشل بسبب الكلل في مروحة التوربينات من نوع فرانسيس في محطة دربندخان الكهرومائية

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المستخلص :
يحدث الفشل بسبب الكلل في مروحة التوربينات من نوع فرانسيس كثيرا في محطات توليد الطاقة الكهرومائية ، مما يتسبب في عطل غير متوقع للمحطات وخسارة مالية كبيرة . وقد أظهرت الخبرة التشغيلية الطويلة بأن مروحة التوربينات تحدث فيها شقوق الكلل في المناطق التي يتركز فيها الجهد والعيوب المادية المتزامنة . تتكون الشقوق في مروحة التوربينات فرانسيس من مناطق ملحومة بين ريش التوربين والحزام أو التاج . يعمل هذا النوع من المراوح تحت مجالات واسعة من ارتقاع الماء وانتاج الطاقة ولذلك يتم تعرضها لقوى ديناميكية كبيرة التي يمكن أن تؤدي إلى كسر المروحة . تتغير قيمة هذه القوه الديناميكية مع تغير قيمة الضغط الهيدروليكي وسرعة المياه والشكل الهندسي للأجزاء الثابتة التي توجه المياه إلى المروحة.

يعرض هذا البحث ضغوط المياه وجهود الانحناء لظروف تشغيل مختلفة لمروحة فرانسيس في محطة دربندخان الكهرومائية . في المرحلة الأولى من هذا البحث تم استخدام حسابات ديناميكية السوائل لتحديد ضغط المياه المتذبذب على ريش المروحة . اما في المرحلة الثانية تم تحديد جهد الانحناء بسبب ضغط المياه المتذبذب في فترة التشغيل . والنتائج تشير إلى أنه تم تشغيل المحطة الكهرومائية مع ضغط المياه المتذبذب بصورة عالية وهو المسؤول عن خلق جهد الانحناء العالي . جهد الانحناء العالي على ريش مروحة التوربين هو السبب المحتمل للفشل بسبب الكلل . واضافة الى ذلك يناقش البحث تحليل الكلل في مروحة التوربين . كذلك تم عرض النتائج ومن ثم مناقشتها .

الكلمات المفتاحية : توربين فرانسيس ، فشل الكلال ، ظروف التشغيل ، ضغط المياه ، اجهاد الإنحناء .

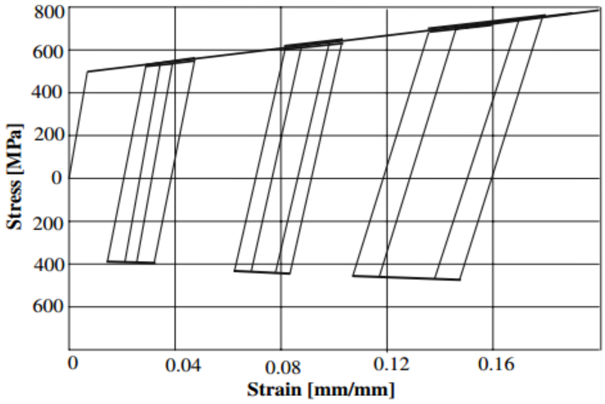

Figure (1) : Steel stress-strain history as predicted on the basis of plasticity theory [10].

In this section, the static and dynamic stress analyses in Francis turbine runner are reviewed.

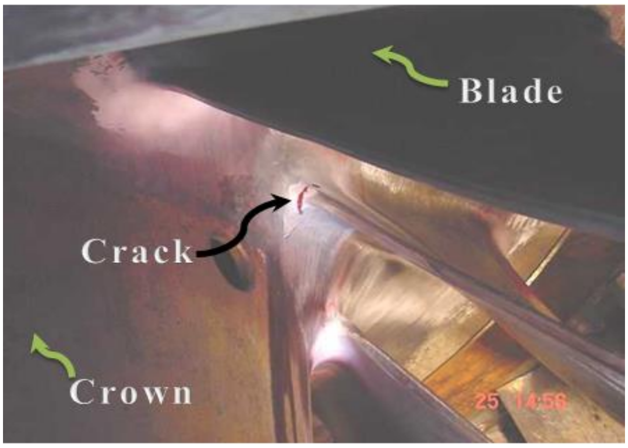

Figure (2) : Crack in the runner blade.

Table (1) : Derbendikhan Hydropower Station data [6]

Type of Dam	Earth Dam
Type of turbine	Francis turbine
Number of units	3
Rated head	80 m
Turbine output at rated head	83000 kW
Turbine discharge at rated head	113 m^3/s
Rated speed	187.5 rpm
Number of runner blades	13
Runner inlet diameter	3 m
Runner yield point	490 MPa
Runner tensile strength	637 MPa
Type of material	Stainless Steels - 410 AISI

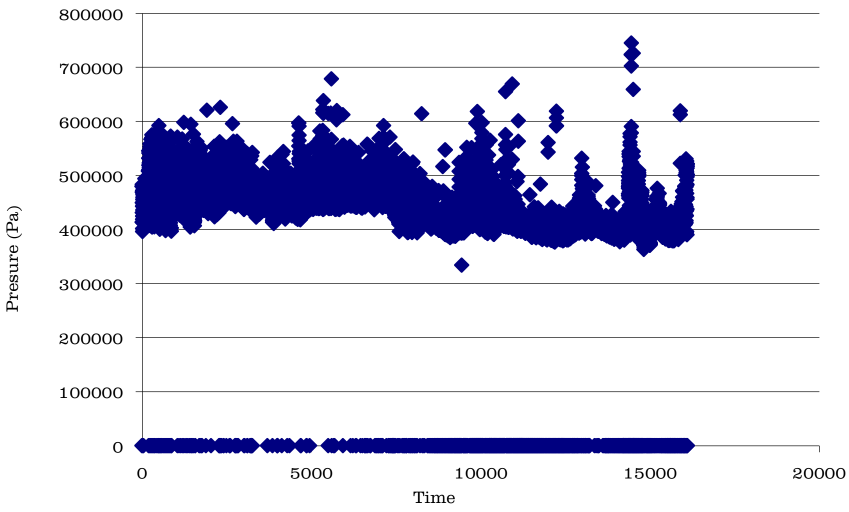

Figure (3) : Fluctuated Fluid Pressure on the Blade.

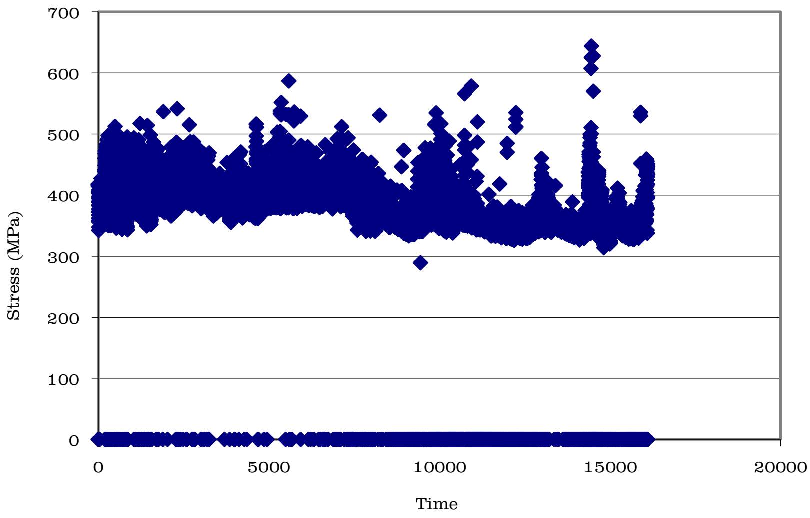

Figure (4) : Bending Stress on the Blade.

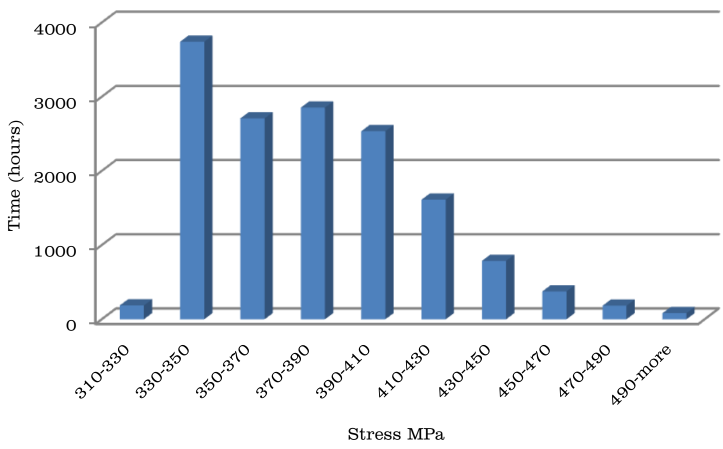

Figure (5) : Number of Hours with the Interval of Bending Stress.