SHEAR PIN ANALYSIS AND DESIGN REVISITED; CASE STUDY OF MINI-HYDRO TURBINE GUIDE VANE

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Abstract

Shear Pin, as it was known, is used for fail safe mechanism of various mechanical systems such as aircraft engine mounting, valve, coupling, gear train, flocculator, guide vane etc. The shear pin should withstand the static as well as dynamic or fatigue loads but it should fail under a certain overload. Depending on the loading speed, it may fail under quasi-static, intermediate strain rate or low velocity/bar impact load. Thus an accurate load determination is very important.The choice of shear pin material and its dimension is also critical to assure that it works properly. In this paper, a case study of analysis and design procedure of a shear pin for guide vane of a mini-hydro water turbine is presented. For load calculation, FLUENT- CFD (Computational Fluid Dynamics) and ANSYS- Finite Element softwares are employed to calculate the load for various positions of the guide vane. The load calculation result is used as data base for static and dynamic loading of the shear pin as well as the guide vane. Furthermore, simulation for overload condition and water hammer phenomenon are also conducted. By using the softwares the accuracy of the loading is assured, thus it is enhanced the design and analysis procedure. Shear test using quasi-static is performed to verify the procedure.

Keywords: Shear Pin, Analysis and Design, Quasi-static and Low Velocity Impact Loading, Guide Vane, Mini-Hydro Power.

1. INTRODUCTION

Shear pin has been widely used for fail safe mechanism. The design and analysis is simple, however an accurate determination of the static and dynamic loading as well as careful choice of material are needed. Design and operational error sometimes occurs. Azevedo et.al. (2009) conducted an experimental failure analysis of hydro electric plant shear pin due to bending fatigue. Abhay et.al. (2008) studied the microstructure of failed shear pin exposed to fatigue and impact loading.

The load acted on shear pin depends on where the shear pin is used. Shear pin design for flocculator [Smith et.al., 2007] and hydraulic safety valve for petroleum [Ibragimov and Gazarov, 1975] were conducted to assure that the shear pin is not failed in operation. Laksana (2009) reexamined the shear pin design for flocculator to make it more reliable.

In this paper, a case study of analysis and design procedure of a shear pin for guide vane of a mini-hydro water turbine is presented. Guide vane has to be reliable because of its vital functions, its price is relatively expensive (made from martensitic stainless steel), and its installation is relatively complicated. The mini-hydro plant, PLT Sawidago, has been designed and installed by PT.GREAT (Ganesha Reverse Engineering and Tools Making) an incubator company of Institut Teknologi Bandung in

Poso, Central Sulawesi. The aim of the mini-hydro project is to develop an appropriate technology project for remote area as social entrepreuner activity to enhance the welfare of the people.

The focus of the presentation of this paper is on the load determination for the guide vane and shear pin which can be calculated accurately due to the availability of sophisticated engineering softwares. For load calculation, FLUENT-CFD (Computational Fluid Dynamics) and ANSYS-Finite Element softwares are employed to calculate the load for various positions of the guide vane. Dynamic force analysis received by the shear pins due to water hammer phenomenon was also conducted based on ASME standards [ASME, 1933]. For experimental validation, shear stress test using single notched shear pin is conducted. Later, the most suitable material of the shear pin will be recommended for use.

2. LOAD ANALYSIS OF GUIDE VANE

Types of load affecting the guide vane and shear pin are normal operating load, overload, and dynamic load. Normal operating load occurs when the turbine is operated in its specified condition. Overload happens when a foreign object inhibits the guide vane movement during opening or closing. Dynamic load arises due to water hammer phenomenon or in load rejection case. The load scenario and requirements for guide vane as well as shear pin are shown in Table 1.

2.1 Normal Operation Load Case of Guide Vane

Guide vane and shear pin have to endure their opening position when the turbine is operating in its specified condition. Normal operation load case is modeled using Fluent 6.2 software. The modeling aimed to get the static moments and reaction forces acted on the guide vanes at various opening positions. Guide vanes are analyzed at opening positions of 75^0, 72.5^0, 70^0, 60^0, 45^0, 30^0, and 15^0 to the horizontal line (x-axis), respectively.

K-epsilon standard flow model of Fluent 6.2 software is chosen based on its advantages. The model described the full flow of fluid, but relatively fast in computation time. In addition, the k-epsilon model is also chosen because it is suitable for field conditions of water flow [Tuakia, 2008]. In this case, 2D model is created for steady state condition. Moreover, it is using selected segregated implicit solver. Segregated implicit solver calculates the Reynolds Average Navier-Stokes (RANS) equations in stages and solved the equations separately [Blazek, 2001].

Figure 1. shows the typical result of FLUENT6.2 modeling. The biggest moment because of pressure and friction of water is received by the lower guide vane of the turbine at opening position of 30^0. The maximum moment of 649.9 Nm clockwise should be endure by guide vane as well as shear pin as it was stated in Table 1.

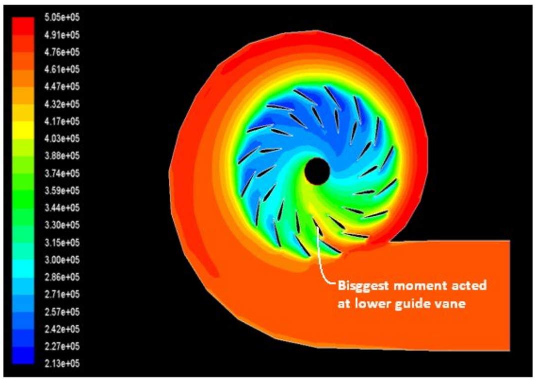

Figure 1. Pressure contour acted on guide vane at 30^0 opening position

2.2 Overload Case of Guide Vane

For overload analysis, a foreign object such as a stone is assumed flowing into the water and inhibits the movement of the guide vane. The presence of the stone could not make guide vane completely closed. On the other hand, the guide vane is forced continuesly to close. The overload

ISBN : 978-602-97742-0-7

may cause the failure of the guide vane and probably deformed permanently. In order to release the load, shear pin must fail first before the failure of the guide vane occurs.

The overload model is conducted using ANSYS software (incorporated in Autodesk Inventor 2010). Von Misses failure or distortion energy (DE) criteria is employed in the analysis. Safety factor of modeling result is shown in Figure 2. Table 2 shows the maximum moments and reaction forces on the guide vane which should be avoided to prevent failure. The maximum moment received by the guide vane is 1135.37 Nm clockwise.

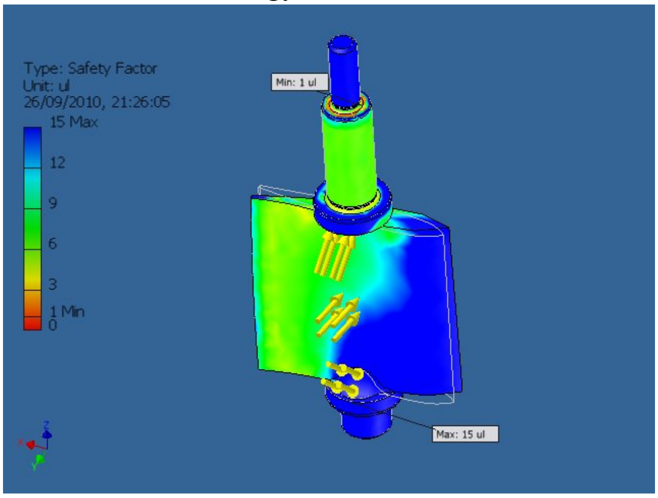

Figure 2. Guide vane modeling for overload case

Table 2. Reaction force and moment of guide vane in overload case

Constraint name	Reaction force		Reaction moment	
	Max Value	**(X,Y,Z)**	**Max Value**	**(X,Y,Z)**
Fixed Constraint	82,14 N	-7,016 N	**1135,37 N m**	4,95 N m
		72,3 N		0,21 N m
		-38,36 N		-1135,35 N m
Pin Constraint	10349,8 N	-393,89 N	587,67 N m	586,32 N m
		-10302,2 N		-39,68 N m
		910,19 N		-0,076 N m
Frictionless Constraint	2141,49 N	138,31 N	32,23 N m	-24,85 N m
		-1942,29 N		20,22 N m
		-891,28 N		-3,61 N m

2.3 Water Hammer Phenomenon Case

Water hammer phenomenon causes the water flow pressure change dynamically. The high pressure can initiate the failure of both the guide vane and the penstock. In the load rejection case, the effect of water hammer can be reduced by longer closing time of the guide vane. In the contrary, the longer closing time can cause the uncontrol increase of the speed and unbalance of the turbine rotor. Due to these constraints, the closing time should be chosen properly.

Water hammer analysis is done using the well-known Joukowsky equation [Permakian, 1955]. The differential equation links the change in pressure to change in fluid velocity on the pipe as follows,

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$$\frac{\delta p}{\delta t} = \rho a \frac{\delta C}{\delta t} \qquad (1)$$

where p is the water pressure, C is the water velocity during the guide vane opening or closing and a is the speed of pressure waves that move along the pipeline respectively. The value of the speed of pressure waves is obtained using Equation 2,

$$a = \frac{1}{\sqrt{\rho\left(\frac{1}{k} + \frac{d}{Ee}\right)}} \qquad (2)$$

where :

a = Pressure waves speed
k = Bulk modulus of water
E = modulus elasticity of the pipe
d = pipe inner diameter
e = pipe thickness
ρ = specific mass of water

Allievi graphs is used to obtain the maximum head of the turbine [Permakian, 1955]. Changes in the head as function of the guide vane closing time are calculated using numerical methods. The calculation of the normalized head rise is shown in Figure 3 and Table 3 for closing time of 2, 4, 7, and 10 second respectively.

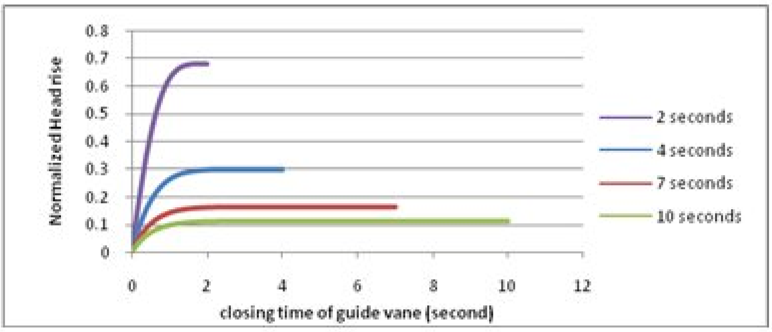

Figure 3. Head alteration versus closing time of the guide vane

The guide vane closure case is more critical and must be analyzed further than the opening case since it increases the total head of the turbine. It can be observed also that the head alterations is smaller when longer time is chosen for opening or closing the guide vane.

3. FAIL SAFE MECHANISM USING SHEAR PIN

As mentioned earlier, the concept of fail safe mechanism is to sacrifice the shear pin to save the guide vane in case of unwanted conditions. The static load scenario for shear pin failure is described in Table 1. To satisfy the requirements, the shear pin static working area should be defined. Static working area is the area bounded by the static lower limit and upper limit of the Ultimate Shear Strength (USS) of the shear pin. Lower limit of the USS shear pin is obtained from calculations of the normal operating load case. Shear pin must withstand the shear stress due to the normal operating load. On the contrary, shear pin must fail when the shear stress exceeding the upper limit of USS shear pin. The upper limit of the USS shear pin is obtained from calculations of the overload case when the guide vane starts to fail, or plastically deformed. In this case, shear pin must fail first before the guide vane fails.

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Table 3. Water hammer head calculation using Allievi graph

No	Case	Parameter	Value (m)	Normalized Head rise
1	Opening or closing time = 2 seconds	**H_{max} opening**	**-29.93**	**-0.63**
		H_{max} closing	**32.3**	**0.68**
2	Opening or closing time = 4 seconds	**H_{max} opening**	**-19**	**-0.4**
		H_{max} closing	**13.3**	**0.28**
3	Opening or closing time = 7 seconds	**H_{max} opening**	**-11.88**	**-0.25**
		H_{max} closing	**7.6**	**0.16**
4	Opening or closing time = 10 seconds	**H_{max} opening**	**-8.55**	**-0.18**
		H_{max} closing	**5.23**	**0.11**

3.1 Calculation of Lower and Upper Limit USS Shear Pin for Static Load

Free body diagram of shear pin and guide vane is shown in Figure 4. Moment arm is assumed constant at 90 mm.

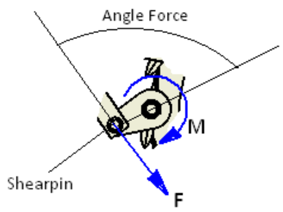

Figure 4. Free body diagram of guide vane and shear pin

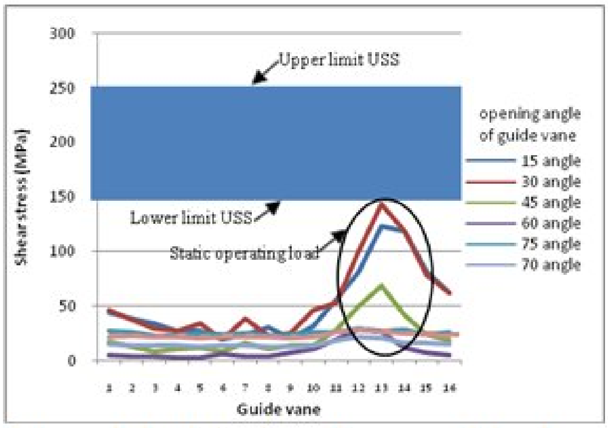

Figure 5. Static load working area of the shear pin

By using the maximum moment of 649.9 Nm received by the lower guide vane at the normal load operating case, the value of shear stress occurs in shear pin is equal to 143.22 MPa. This shear stress value becomes the lower static limit of the USS shear pin. Upper limit of USS shear pin can be determined in the same manner as the lower limit of the USS shear pin, by using maximum moment from the overload model of 1135.37 Nm. Hence the upper limit USS shear pin is 250.97 Mpa. Figure 5 depicts the working areas for the static load of the shear pin (light blue area). The static loads of guide vanes in various positions are drawn in the lower part of Figure 5 for additional illustration.

3.2 Dynamic Load due to Closure Time of Guide Vane

Closure time of the guide vane as described earlier is something that must be determined carefully. The selection of closure time is critical in order to avoid water hammer phenomenon. Water hammer is very dangerous for both penstock and guide vane.

Penstock pressure is increased during the closing process of the guide vane. Rapid changes in pressure and high maximum pressure can cause penstock and guide vane failure. Rapid changes in pressure create dynamic loading. The type of

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the load can be determined by analyzing its strain rate [Panov, 2006].

Strain rate can be estimated using Equation 3, where dH/dt is the rate of change of total head obtained in the previous analysis of water hammer,

$$\frac{d\varepsilon}{dt} = \frac{\rho \; x \; g \; x \; \frac{dH}{dt} \; x \; r}{E \; x \; t} \qquad (6)$$

Results from the calculation of water hammer effect on the penstock are shown in Table 4. Equation 7 is used to determine the safety factor based on the theory of distortion energy (DE) where σ_t, σ_a, and σ_r is the principle stress for the pipe.

$$SF = \frac{\sigma_{yield}}{\left[\frac{(\sigma_t - \sigma_a)^2 + (\sigma_a - \sigma_r)^2 + (\sigma_t - \sigma_r)^2}{2}\right]^{1/2}} \qquad (7)$$

Table 4. The result of calculation of strain rate and safety factor based on distortion energy (DE) criteria

Parameter	Case 1	Case 2	Case 3	Case 4
Closure time (t)	2 second	4 second	7 second	10 second
Maximum water pressure ($P_{i\ max}$)	0.78	0.59 MPa	0,53 MPa	0,51 MPa
Maximum penstock stress (1^{st} *principle stress*) (σ_{max})	31.2	23.6 MPa	21.2 MPa	20.4 MPa
Head rate (dH/dt)	$dH/dt = 15.63t^2 - 58.54t + 53.95$	$dH/dt = 2.673t^2 - 14.052t + 17.32$	$dH/dt = 0.312t^2 - 2.672t + 5.19$	$dH/dt = 0.066t^2 - 0.814t + 2.184$
Strain rate ($d\mathcal{E}/dt$)	$d\mathcal{E}/dt = 1{,}9 \times 10^{-6} \times (15.63t^2 - 58.54t + 53.95)$	$d\mathcal{E}/dt = 1{,}9 \times 10^{-6} \times (2.673t^2 + 14.052t + 17.32)$	$d\mathcal{E}/dt = 1{,}9 \times 10^{-6} \times (0.312t^2 - 2.672t + 5.19)$	$d\mathcal{E}/dt = 1{,}9 \times 10^{-6} \times (0.066t^2 - 0.814t + 2.184)$
Maximum strain rate	8.63×10^{-5}	3.29×10^{-5}	9.86×10^{-6}	4.15×10^{-6}
Load type	Quasi static	Quasi static	Quasi static	Quasi static
Safety factor	7.6	8.2	9.1	9.5

Fluent 6.2 software is employed once again to calculate the moment of guide vane during its closure. Head alterations that previously analyzed is used for input data to determine the moment of the guide vane in five opening positions, from 75^0 to 15^0. The closure process of the guide vane is considered a constant closing speed. Result of the modeling is shown in Table 5.

Tabel 5. Guide vane load and shear pin stress during water hammer case

Closure percentage	**0%**	**25%**	**50%**	**75%**	**100%**
Position	**75^0**	**60^0**	**45^0**	**30^0**	**15^0**
Case 1 (2 second)	0	0.5	1	1.5	2
Head (m)	47.24	67.55	77.131	79.89	79.75
Pressure (Mpa)	0.46	0.66	0.75	0.78	0.78
Moment z-axis (N.m)	-122.8	-160.7	-485.4	-936	-997.5
Shear stress on shear pin (Mpa)	27.14	35.53	107.31	206.93	220.52
Case 1 (4 second)	0	1	2	3	4
Head (m)	48.52	59.705	62.184	61.303	62.408
Pressure (Mpa)	0.46	0.58	0.6	0.6	0.61
Moment z-axis (N.m)	-122.8	-141.2	-396.5	-683.67	-746.7
Shear stress on shear pin (Mpa)	27.14	31.212	87.66	151.14	165.07
Case 2 (7 second)	0	1.75	3.5	5.25	7
Head (m)	49.34	54.89	55.598	54.81	55.878
Pressure (Mpa)	0.46	0.54	0.54	0.54	0.54
Moment z-axis (N.m)	-122.8	-130.7	-332.33	-629.2	-673.2
Shear stress on shear pin (Mpa)	27.14	28.89	73.47	139.1	148.82
Case 3 (10 second)	0	2.5	5	7.5	10
Head (m)	49.45	52.71	52.945	52.2175	52.59
Pressure (Mpa)	0.46	0.52	0.52	0.52	0.52
Moment z-axis (N.m)	-122.8	-123.4	-322.75	-576.74	-636.5
Shear stress on shear pin (Mpa)	27.14	27.28	71.3	127.5	140.7

By combining the results of analysis of static load and dynamic loads due to water hammer, the new working area of shear pin can be determined. Figure 6 shows changes in shear stress on the shear pin versus time due to water hammer phenomenon.

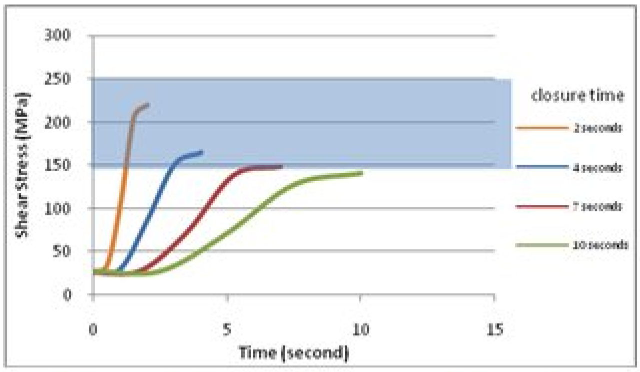

Figure 6. The working area of shear pin

As mentioned above the speed rise of turbine rotor and the pressure rise in penstock are two constraints which should be considered to determine the closing time of guide vane in the load rejection case. To avoid speed rise, closing time of guide vane has to be fast. In this case, 2 seconds of closing time maybe a good choice. On the contrary slow closing time is needed to prevent pressure rise and 10 seconds of closing time seem to be more suitable.

Examination of Figure 6 indicates that the shear stress increase significantly due to the closing time difference especially for 2 seconds case. After careful consideration of the aspects for speed and pressure rise, 4 seconds of closing time is finally chosen. Lower limit of the USS shear pin is changed to 165.07 MPa. If the closing time is less than 4 seconds, the shear pin should fail to save the guide vane as described previously in Table 1.

4. EXPERIMENTAL VALIDATION

The test aims to determine the suitable shear pin material to be used as sacrificial material. Shear pin should fail in the working area previously determined. A simple direct shear test is designed to conduct the validation using a Tension Testing Machine with a special shear test fixture.

For experimental validation, the shear pins are chosen from two different types of materials, Al2024 with artificial aging and Al2024 with natural aging treatment. Three samples are tested to observe the statistical variation and consistency of the test procedure. Test results of shear stress are shown in Figures 7. From the experimental validation, material which is suitable to be used as shear pin is Al2024 with natural aging treatment since it has an average shear strength of 186 MPa which is a little higher than the lower USS of 165 Mpa.

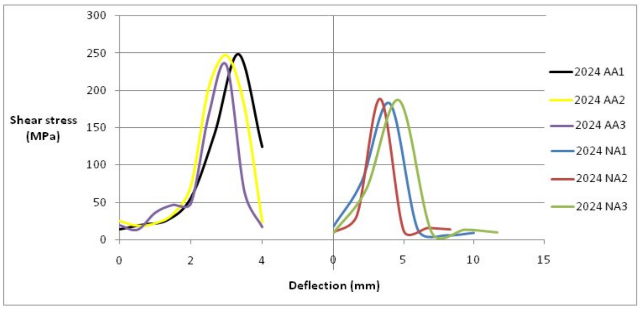

Figure 7. Shear stress of Al2024 shear pin with artificial aging (left) and natural aging treatment (right)

5. CONCLUSION

This paper shows the calculation procedure for static, static overload and dynamic loads acted on the guide vane and shear pin of the mini-hydro turbine. The working area of the shear pin is created by the above mentioned loads and used as important data base for choosing the dimension and the material of the shear pin. From experimental validation, Al2024 with natural aging treatment is suitable for shear pin material.

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