PRACE
INSTYTUTU MASZYN
PRZEPŁYWOWYCH

TRANSACTIONS

OF THE INSTITUTE OF FLUID-FLOW MACHINERY

90-91

PRACE INSTYTUTU MASZYN PRZEPŁYWOWYCH

poświęcone są publikacjom naukowym z zakresu teorii i badań doświadczalnych w dziedzinie mechaniki i termodynamiki przepływów, ze szczególnym uwzględnieniem problematyki maszyn przepływowych

*

THE TRANSACTIONS OF THE INSTITUTE OF FLUID-FLOW MACHINERY

exist for the publication of theoretical and experimental investigations of all aspects of the mechanics and thermodynamics of fluid-flow with special reference to fluid-flow machinery

HYDROFORUM

KONFERENCJA NAUKOWO-TECHNICZNA

na temat

ZAGADNIENIA HYDRAULICZNYCH MASZYN WIROWYCH

Gdańsk-Władysławowo, 24-27 września 1985 r.

*

HYDROFORUM

SCIENTIFIC-TECHNICAL CONFERENCE

on

PROBLEMS OF HYDRAULIC TURBOMACHINES

Gdańsk-Władysławowo, September 24-27, 1985

*

ГИДРОФОРУМ

НАУЧНО-ТЕХНИЧЕСКАЯ КОНФЕРЕНЦИЯ

на тему

ПРОБЛЕМЫ ГИДРАВЛИЧЕСКИХ ТУРБОМАШИН

Гданьск-Владыславово, 24-27 сентября 1985 г.

VICTOR KANICKÝ

ČKD Blansko, Czechoslovakia

Some Results of Shear Pin Stress Measurements on the Żarnowiec Pump-Turbine

Field test measurements aimed at determining actual operating loads of shear pins of the Żarnowiec pump-turbine guide vane linkage are described. The procedure of dimensioning the shear pin with respect to fatigue using thus obtained experimental data is outlined.

1. Introduction

The guide vane linkages of pump-turbines are frequently protected against overloads caused by closure on debris by some properly designed rupture elements, such as pull rods or shear pins included in guide vane operating levers. However, it is well known that in some pump-turbines extensive problems occurred due to failure of these elements during steady-state operation, see e.g. [1] ÷ [5]. This could be referred to the fact that in dimensioning these elements a design procedure customary for classical turbines has been applied. In pump-turbines, the dynamic loads of guide vanes during transient operations may be many times higher than the steady loads obtained from hydraulically scaled model tests [1]. Thus, the quasistatic dimensioning procedure does not correspond to the actual conditions [7]. Since the maximum dynamic loads of guide vane linkage appear at some particular transients even the results obtained from tests with special dynamically oriented models (rigorous hydroelastically scaled models cannot be realized) should be converted to the real machines with extreme caution. Some important effects, either general (e.g. hydraulic system resonances [1]) or particular (e.g. actual clearances [6] or misalignments of distributor elements [8]) cannot be realistically estimated through model tests at all.

The manufacturer of Francis-type pump-turbines for Żarnowiec Power Plant was fully aware of the above mentioned facts and therefore the problem of dimensioning the shear pins was solved not only by computations.

2. Preliminary Design of the Shear Pin

In designing a shear pin for the guide vane linkage of the 180 MW pump-turbine usual strength computations were performed. The data defining the design loading were derived from the tests on the corresponding model turbine and from the results
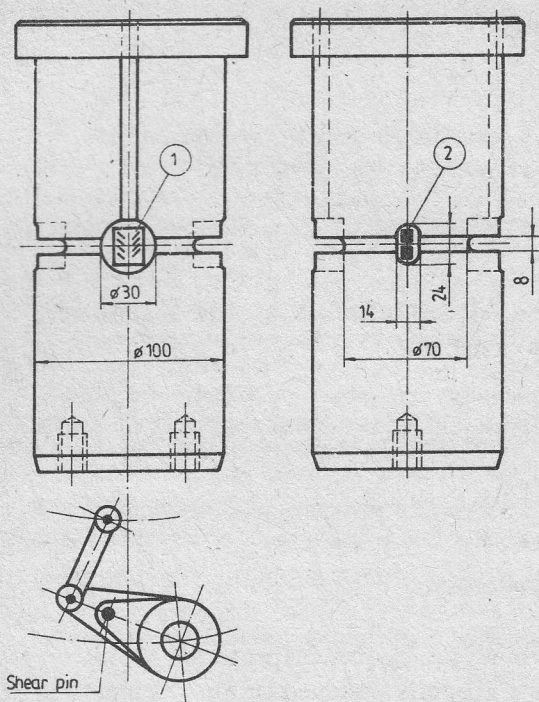
Fig. 1. Shear pin modified for measuring purposes
1 — strain gauges indicating the shear loading (full-bridge connection used), *2* — strain gauges indicating bending (two half-bridge connections used)

of field measurements on similar, already operating machines. The design strength was derived from standard shear strength validated for the applied shear pin material (a cast iron type). The preliminary design resulted in a 100 mm diameter pin with a 15 mm notch depth (see the basic outline in Fig. 1). Thereafter, the accepted basic shear resistance was verified experimentally. Two sets of produced shear pins (each set comprising seven pins selected at random) were tested using a special jig adopted for a conventional testing machine. The mean value of the total shear resistance force (900 kN) obtained in such a way corresponds well to that used in the preliminary design. With respect to uncertainties in theoretically predicting the actual guide vane linkage dynamic loads, refining fatigue strength analysis required for final dimensioning of the shear pin notch was postponed until some reliable data from field test measurements on the machines were available.

3. Field Test Measurements

The operating loads of shear pins were investigated within the scope of measurements carried out during the comissioning tests. Four specially adapted shear pins were used at guide vanes equally circumterentially spaced in order to get a reliable estimation of the shear pin design loads. These shear pins (Fig. 1) were provided with cylindrically shaped recesses for the installation of three strain gauge systems, namely one for sensing the shear deformations and two for checking the level

of bending. For signal conditioning 5 kHz carrier frequency measuring amplifiers with low-pass filters were used. Four of 18 channels of the UV-oscillograph engaged in commissioning test measurements were reserved to record the signals.

The shear pin loads were investigated in a broad range of operating conditions, even outside the normal range of steady state operation. Measurements were carried out at partial loads in turbining for 11 values of distributor openings (from 40 to 100 percent) and in pumping for 9 openings (from 40 to 90 percent). With few exceptions, the whole set of possible mode changes was investigated. All tests were run at the maximum, mean and minimum heads.

4. Analysis of the Results of Measurements

The records of strain gauge output signals were divided in an adequate number of sections, with the aim to make it possible to analyze signals in each section as quasi-stationary processes with eliminated trends in mean values. In each section, covering a certain, well defined phase of unit operation, the average mean values and predominant fluctuating components of signals were evaluated. Thus, for each phase of the unit operation, steady mean stresses in shear pins and the numbers of predominant load cycles with the respective peak-to-peak stresses were determined. For each phase, the set of obtained on the instrumented shear pin with the overall highest load intensity was selected as representative for the final estimation of the design loads.

A selection of representative shear pin stresses, which may be considered as being extreme for some typical operating conditions of the pump-turbine at the maximum, mean and minimum heads is presented in Table 1.

5. Final Dimensioning of the Shear Pin Notch

The final dimensioning of the shear pin notch was based on a simplified fatigue strength analysis. For each investigated phase of pump-turbine operation, from the three representative stress data sets obtained at characteristic heads, that one was selected as the respective design stress data set, which was estimated as the most significant in the fatigue life prediction (e.g. see Tab. 1). This approach was used because no reliable prediction of the time extent of operation at various heads could be made.

For the theoretical shear in fatigue life prediction, the real loading spectrum has to be appropriately described. In the problem under consideration a block spectrum loading representation was used with individual load blocks deduced from the design stress data sets (with certain simplifications). Based on the experience, a compound loading block spectrum (M) with the following structure was used to simulate the

Table 1

Some Results of Measurements

Characteristic operating conditions T — turbining P — pumping	Measured stresses (mean σ_m, peak-to-peak $2\sigma_a$)						Assumed design values			Block type
	σ_m [MPa]	$2\sigma_a$ [MPa]	σ_m [MPa]	$2\sigma_a$ [MPa]	σ_m [MPa]	$2\sigma_a$ [MPa]	σ_m [MPa]	σ_a [MPa]	number of cycles	
T — start-up	48	34	28	14	42	26	48	17	120	OT
T — stationary	44	3	42	0	42	2	44	1	cont.	T
T — shut-down	39	6	43	4	34	18	34	9	520	T01
T — shut-down	18	12	23	10	16	12	18	6	810	T02
T — load rejection	38	50	20	62	26	70	26	35	14	TR
T — emergency shut-down	30	70	15	78	24	86	24	43	9	TE
P — start-up	25	22	28	34	37	44	37	22	41	OP
P — stationary	22	2	27	2	31	2	31	1	cont.	P
P — shut-down	48	79	55	70	50	56	48	39	5	PO
P — emergency	66	86	50	84	45	72	66	43	8	PE1
shut-down	42	40	40	28	38	26	42	20	21	PE2
P— T direct	58	88	57	83	57	74	58	44	3	PT1
change-over	44	35	52	37	49	30	52	18	68	PT2

unit monthly operation:

$$M = 128\,A + 112\,B + 16\,C + 2\,D + 2\,E + 2\,F.$$

Here

$$A = \sum (OT)_k + \sum (T)_k + \sum (TO)_k$$
$$B = \sum (OP)_k + \sum (P)_k + \sum (PO)_k$$
$$C = \sum (OP)_k + \sum (P)_k + \sum (PT)_k + \sum (T)_k + \sum (TO)_k$$
$$D = \sum (OT)_k + \sum (T)_k + \sum (TR)_k$$
$$E = \sum (OT)_k + \sum (T)_k + \sum (TE)_k$$
$$F = \sum (OP)_k + \sum (P)_k + \sum (PE)_k$$

denote the loading block spectra corresponding to normal turbining cycle (A), normal pumping cycle (B), operation cycle involving direct pumping-turbining mode change (C), turbining cycles involving full load rejection (D) or emergency shut-down (E) and pumping cycle with emergency shut-down (F). The notation of individual load steps, e.g. $(OT)_k$, or individual load blocks, e.g. $\sum (OT)_k$, is obvious from Table 1.

The fatigue limits for the shear pin material were estimated to be at least $\sigma_c = \pm 80$ MPa (symmetrical cycle) and $\sigma_c = 52 \pm 52$ MPa (pulsating cycle). The fatigue limits for the shear pin, taking into account the size and surface quality effects and stress concentration were determined to be at least $\sigma_c^* = \pm 44$ MPa — for a symmetrical loading cycle, $\sigma_c^* = \pm 29$ MPa — for a pulsating loading cycle. For estimation of the influence of the load-cycle asymmetry the Haigh's diagram was constructed (Fig. 2). Here the points representing the load steps specified in Table 1 are indicated by dots.

In predicting the fatigue life, the Miner's rule of cumulative damage was applied. However, it is assumed [9] that only those loading cycles contribute to fatigue
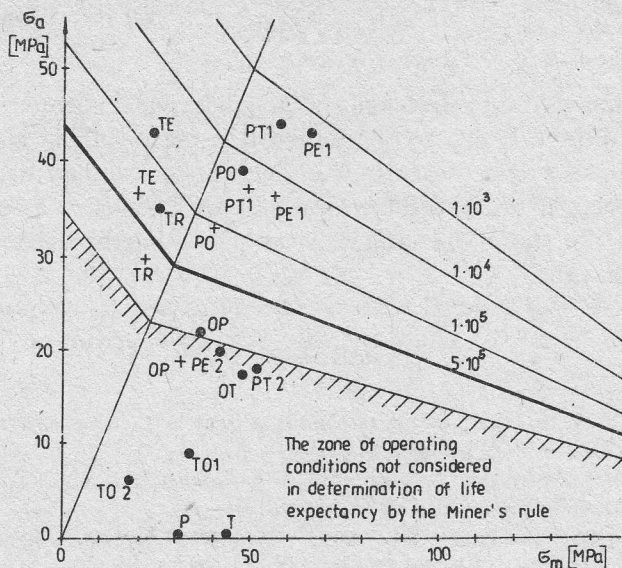
Fig. 2. Haigh's diagram

damage whose dynamic stress component amplitude exceeds 80 percent of the fatigue limit at the respective load cycle asymmetry (see Fig. 2). The damage D_i for n_i cycles at the i-th stress amplitude level is given by

$$D_i = \frac{n_i}{N_i}$$

where N_i is the number of loading cycles to shear pin fracture at this level. The predicted fatigue life expressed through the number of compound loading blocks Λ is given by

$$\Lambda_M = \left[\sum_i D_i\right]^{-1}.$$

In the case under consideration the computations resulted in

$$\Lambda_M = 15.8.$$

This corresponded to a shear pin fatigue life prediction of about 16 months of the pump-turbine continuous operation.

Therefore the depth of the notch was decreased in order to decrease the stress levels and calculations were repeated. For a notch depth of 12 mm, a reasonable fatigue life

$$\Lambda_M = 133$$

i.e. about 11 years, was obtained. In Haigh's diagram (Fig. 2), the points, representing the new load steps are indicated by crosses.

Consequently, it was recommended to manufacture the shear pins with this notch depth and to ensure regular replacements of shear pins in all units in ten-year periods.

6. Conclusion

Proper notch dimensions of shear pins of the guidevane linkage for pump-turbines of the Żarnowiec Power Plant were determined using a fatigue strength analysis, based on operating load spectra obtained in field test measurements. It can be concluded that only such a procedure (preferably refined) ensures a reasonable compromised between the static and fatigue shear pin strengths needed for safe protection of the guide vane.

References

[1] Swift W. L., Whippen W. G., *A Study of the Effects of Dynamic Loads in Reversible Pump/Turbine.* Proc. of the Eight Symp. IAHR, Leningrad 1976.

[2] Fujisaki M., *Large Pump-Turbines in Japan Achieved by Toshiba.* In: Toshiba Pump Storage Power Plants — Spec. Ed. for Symposium at Leipzig Messe, 1976.

[3] Beducci G. at al, *Commissioning and Efficiency Measurement in Modern Hydro-Electric Generating and Pumping Units: Comparison with Laboratory Measurements.* In: Proc. Symposium IAHR — Operating Problems of Pump Stations and Power Plants, Amsterdam 1982.

[4] Schmidt J., Sondermann K., *Erste Erfahrungen im Pumpenspeicherwerk Rönkhausen.* Elektrizitätswirtschaft, Jg. 69 (1970). Heft 10.

[5] O'Brien J. T., Swift W. L., *Field Investigation and Reduction of Wicket Gate Vibration in a Francis--Type Reversible Pump-Turbine.* Journal of Fluids Engineering, Trans. ASME, March 1974, pp. 16—20.

[6] Fischer G., Raithel F., *Paper on vibration phenomena during change-over from one mode of operation of pump turbines to another.* In: Joint Symposium on Design and Operation of Fluid Machinery. Proc. Vol. II, Colorado State Univ. Fort Collins, 1978.

[7] Grein H., Baumann K. M., *Commissioning problems of a large pump-turbine.* In: Water Power and Dam Construction, December 1975.

[8] Ginalski J., Roliński Z., *Influence of guide-vane setting on the dynamic behaviour of pump-Turbines* (in Polish). In: Conference Proc. — Hydroforum'80, Part I, Porąbka-Kozubnik 1980.

[9] Bílý M. et al., *Strength of Structural Components and Materials Dynamically Loaded* (in Czech.). VEDA, Bratislava 1976.

Niektóre wyniki pomiarów naprężeń w kołkach ścinanych pompoturbiny dla elektrowni w Żarnowcu

Streszczenie

Jest rzeczą znaną, że niektóre pompoturbiny uległy poważnym uszkodzeniom na skutek złamania jednego po drugim, kilku kołków bezpiecznikowych ścinanych. Kołki ścinane mają za zadanie chronić rozdzielacz przed przeciążeniem, spowodowanym dostaniem się obcych ciał pomiędzy łopatki kierownicy. Jednakże w pompoturbinach, gdzie występują duże obciążenia dynamiczne, kołki te mogą być przyczyną poważnych awarii. Powodem jest to, że przy ich wymiarowaniu stosuje się metody projektowania takie jak dla klasycznych turbin. W pompoturbinach dynamiczne składowe obciążenia mogą być w stanach nieustalonych nawet kilkakrotnie większe niż ustalone obciążenia otrzymane z prób modelowych przeprowadzonych w odpowiedniej skali hydraulicznej. Tak więc procedura quasi-statycznego wymiarowania nie odpowiada warunkom rzeczywistym. Ponieważ maksymalne obciążenia dynamiczne pojawiają się w niektórych stanach przejściowych nie da się ich wyznaczyć z możliwą do przyjęcia dokładnością za pomocą hydraulicznego modelowania w skali.

Producent pompoturbiny Francisa dla elektrowni w Żarnowcu był w pełni świadom tych faktów. Z tego względu problem wymiarowania kołków ścinanych rozwiązano nie tylko przez obliczenia, ale również za pomocą pomiarów w warunkach normalnego użytkowania. Pomiary naprężeń w kołkach bezpiecznikowych ścinanych przeprowadzono w całym zakresie warunków pracy włącznie ze stanami przejściowymi. Ostatecznego ich wymiarowania dokonano opierając się na zmierzonych wartościach naprężeń i własnościach zmęczeniowych zastosowanego materiału.

W pracy opisano przeprowadzone pomiary, przedstawiono niektóre z otrzymanych wyników, procedurę wymiarowania kołków ścinanych oraz wnioski co do ich konstrukcji.

Некоторые результаты измерений напряжений в срезываемых штифтах насосотурбины для электростанции в Жарновце

Резюме

Является известным, что некоторые насосотурбины подвергались значительным повреждениям ввиду сломления, один за другом, нескольких срезываемых предохранительных штифтов. Задачей срезываемых штифтов является предохранение распределителя против перегрузке, вызванной попаданием чужих тел между лопатки направляющего аппарата. Однако в насосотурбинах, где выступают большие динамические нагрузки, эти штифты могут быть причиной серьёзных аварий. Поводом является то, что определяя их размеры применяются проектные методы такие как для классических турбин. В насосотурбинах динамические составляющие нагрузки могут оказаться в неустановившихся состояниях даже несколько раз большими чем установившаяся нагрузка, определённая модельными испытаниями, проведёнными в соответствующем гидравлическом масштабе. Таким образом процедура квазистатического определения размеров не отвечает действительным условиям. Так как максимальная динамическая нагрузка появляется в некоторых переходных состояниях, нет возможности их определения с возможной для приёма точностью при помощи гидравлического моделирования в масштабе.

Производитель насосотурбины Френсиса для электростанции в Жарновце был вполне сознателен этих фактов. По этому поводу проблема определения размеров срезываемых штифтов решалась не только путём расчётов, но также с помощью измерений в условиях нормального употребления. Измерения напряжений в предохранительных срезываемых штифтах производились в целом диапазоне условий работы, включая переходные состояния. Окончательное определение их размеров произошло на основе измеренных значений напряжений и усталостных свойств применяемого материала.

В работе описаны произведённые измерения, представлены некоторые из полученных результатов, процедура определения размеров срезываемых штифтов, а также выводы относительно их конструкции.