

P O L S K A A K A D E M I A N A U K
I N S T Y T U T M A S Z Y N P R Z E P Ł Y W O W Y C H

P R A C E
I N S T Y T U T U M A S Z Y N
P R Z E P Ł Y W O W Y C H

T R A N S A C T I O N S
O F T H E I N S T I T U T E O F F L U I D - F L O W M A C H I N E R Y

90-91

W A R S Z A W A — P O Z N A Ń 1989

P A Ń S T W O W E W Y D A W N I C T W O N A U K O W E

PRACE INSTYTUTU MASZYN PRZEPLYWOWYCH

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

RADA REDAKCYJNA — EDITORIAL BOARD

TADEUSZ GERLACH · HENRYK JARZYNA · JERZY KRZYŻANOWSKI
STEFAN PERYCZ · WŁODZIMIERZ PROSNAK
KAZIMIERZ STELLER · ROBERT SZEWAŁSKI (PRZEWODNICZĄCY · CHAIRMAN)
JÓZEF ŚMIGIELSKI

KOMITET REDAKCYJNY — EXECUTIVE EDITORS

KAZIMIERZ STELLER — REDAKTOR — EDITOR
WOJCIECH PIETRASZKIEWICZ · ZENON ZAKRZEWSKI
ANDRZEJ ŻABICKI

REDAKCJA — EDITORIAL OFFICE

Instytut Maszyn Przepływowych PAN
ul. Gen. Józefa Fiszer 14, 80-952 Gdańsk, skr. pocztowa 621, tel. 41-12-71

Copyright
by Państwowe Wydawnictwo Naukowe
Warszawa 1989

Printed in Poland

ISBN 83-01-09017-0

ISSN 0079-3205

PAŃSTWOWE WYDAWNICTWO NAUKOWE - ODDZIAŁ W POZNANIU

Nakład 350+90 egz.	Oddano do składania 27 X 1987 r.
Ark. wyd. 27. Ark. druk. 19,5	Podpisano do druku w sierpniu 1989 r.
Papier offset. IV kl. B-1	Druk ukończono w styczniu 1990 r.
Zam. nr 2/90	K-8/295

DRUK ZAKŁAD POLIGRAFII WSP W ZIELONEJ GÓRZE

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.

JANUSZ GINALSKI

Zakład Metaloznawstwa Instytutu Energetyki, Warszawa (Metallurgical Department of the Institute of Power Engineering, Warszawa)

Stresses in Penstock Concave Bucklings During Operation of Reversible Turbine Sets

The paper presents the mechanism of accumulation of stresses at the bottom of a concave buckling in a cylindrical penstock shell. Also the results of tensometric measurements of these stresses in different phases of turbine set operation are presented.

1. Introduction

The problem of determining the actual value of stress acting in the shell of a large diameter penstock is of importance for strength diagnostics and forecasts of operating life.

The total instantaneous working stress occurring in a given area of the shell during turbine set operation is a tensor sum of component stresses generated by mechanical and thermal loads. The most important of the mechanical loads are loads from internal hydrostatic and hydrodynamic pressure, and of the thermal loads — those resulting from solar radiation under variable weather conditions.

Some of these components can be relatively easily and accurately determined analytically, providing the shell is cylindrical and in case of bucklings — axially symmetrical.

Difficulties in formulating and solving corresponding equations may appear in case of shells without axial symmetry. Then it is easier to determine all the above mentioned component stresses by experimental methods, e.g. by means of resistance strain gauges.

An example of such a situation is a buckling formed during welding together of two sheets with edge areas excessively bent during rolling which has been carried out in order to give them a constant curvature radius equal half of the pipeline diameter.

Such bucklings, in the form of so-called "concave roofs" have been found in some areas of shells of large-diameter penstocks at the Żarnowiec Water Power Plant, especially in their upper sections of 7 100 mm diameter, made of 18G2A steel sheets of 17-19 mm thickness.

2. Stress Distribution in the Buckling Bottom Area

In the buckling area of a cylindrical shell under internal pressure p there occurs an accumulation of stress caused by bending moments.

Its mechanism is best explained by a simplified model in the form of a flat rectangular sheet. The main stresses σ_w operate on its edges parallel to the buckling axis, and the V type concave buckling is passing throughout the sheet width, as shown in Fig. 1. Such a model is quite close to the case of a light-wall large-diameter pipe of a small curvature $1/R$, the concave buckling running along the generatrix throughout the pipe length and the pipe being equipped with a telescopic compensator which eliminates longitudinal stress.

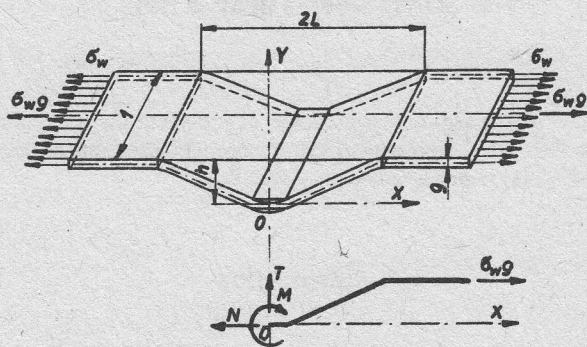


Fig. 1. Concave buckling in a rectangular sheet and diagram of buckling bottom loading

By leading an imaginary cross-section along the buckling axis and reducing the loads to the centre of this section, the equation of the sum of moments relative to point 0 is obtained for a sheet section of unit width

$$\sigma_w gh - M_0 = 0. \quad (1)$$

The sum of projections on the x and y axes yields

$$N = \sigma_w g; \quad T = 0. \quad (2)$$

The largest normal stresses due to the bending moment appear in the discussed section in the extreme fibres of the surface layers and will be defined by a relationship taking into account the sheet stiffness

$$\sigma_G = \pm \frac{M_0}{W_p} = \pm 6(1-\nu^2) \frac{h}{g} \sigma_w \quad (3)$$

where

$$M_0 = \sigma_w gh \quad \text{and} \quad W_p = \frac{g^2}{6(1-\nu^2)}.$$

The normal stress σ_N which is caused by force N and distributed uniformly on the whole cross-section, has the value of:

$$\sigma_N = \sigma_w. \quad (4)$$

The resultant stress σ_z in the surface fibres of the buckling concave side is an algebraic sum of component stresses σ_G and σ_N and is defined by the equation:

$$\sigma_z = \sigma_G + \sigma_N = \left[6(1 - \nu^2) \frac{h}{g} + 1 \right] \sigma_w. \quad (5)$$

Assuming roughly that the σ_w stress corresponds to the stresses appearing under loading of internal pressure p in a light-wall cylindrical shell of radius R and thickness g , that is

$$\sigma_w = \frac{pR}{g}, \quad (6)$$

we can determine the stress in the buckling bottom area of such a shell from the approximate relationship:

$$\sigma_z = \left(5.46 \frac{h}{g} + 1 \right) p \frac{R}{g} = \alpha \sigma_w \quad (7)$$

where Poisson ratio $\nu = 0.3$ has been assumed and α denotes the stress concentration ratio.

It results from equation (7) that the increase of stress in the buckling bottom area depends mainly on the ratio of the bend depth h to the shell thickness g .

Taking for instance the model shown in Fig. 2, the stress concentration ratio α defined by relationship (1):

$$\alpha = 6(1 - \nu^2) \left(\frac{h}{g} \cos \beta + \frac{L^2}{2gR} - \frac{h^2}{2gR} \right) + \left(\cos \beta - \frac{h}{R} \right) \quad (8)$$

can be calculated from the equation of the sum moments relative to point 0. For small β and R large relative to h and L

$$\cos \beta = 1; \quad \frac{L^2}{2gR} = 0; \quad \frac{h^2}{2gR} = 0; \quad \frac{h}{R} = 0$$

and equal values of α in relationships (7) and (8) may be assumed. Symbols used in these relationships are defined in Fig. 2.

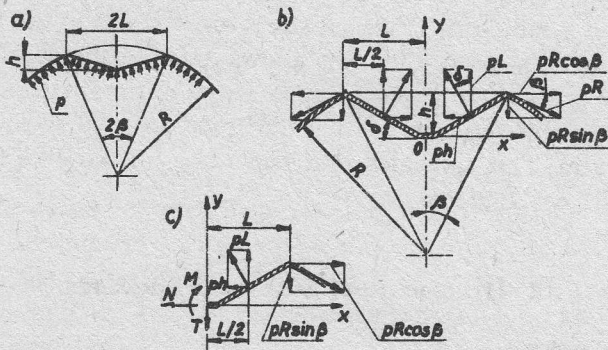


Fig. 2. Concave buckling in a cylindrical shell and diagram of buckling loading by a system of forces equivalent to pressure operation

In this model, the moment acting in the plane of the cross-section, due to the connection of the buckling with the cylindrical shell, is neglected and the connection is considered an articulated joint.

Incidentally, it should be mentioned that a severe plastic strain occurred at the tensometrically tested buckling during the first cycle of loading in the pressure test, in effect of which it changed its profile and decreased its depth. During the second cycle, when it was loaded to a maximum pressure that was lower than in the first cycle, the both tested bucklings deformed elastically and stresses in corresponding measurement points were of same value. An important for diagnostic tests conclusion results from this observation:

— in order to evaluate the state of strains in the buckling bottom area it is not necessary to select the deepest buckling, because every buckling at the bottom of which a permanent set appears is sufficient. After the first loading cycle (during a pressure test with maximum pressure higher than the work pressure), deep bucklings will get permanently deformed (rounded) proportionally to their depth, and in the next cycles stresses in these bucklings will have similar values.

In shallow bucklings not set permanently, the stresses will be equal to or lower than stresses in deformed bucklings.

3. Quasi-Static Load Stresses in Bucklings

The stress accumulation phenomenon appeared both for hydrostatic and thermal loading of the penstock. Circumferential stresses from hydrostatic loads (after strengthening the material in the first loading cycle) in the buckling bottom area were twice as large as the stresses in the neighbouring geometrically correct shell area, and were a linear function of pressure defined by the head measured between the upper reservoir surface and the measurement point.

Thermal load stresses caused by sun radiation on the penstock shell occurred in the buckling area only and reached $\pm 0.8 \text{ kN/cm}^2$ at average weather (passing clouds, temperature about 18°C , light wind). Under these conditions no thermal stresses in the geometrically correct shell area were observed.

The regular and symmetrical, thermal stress distribution in the buckling bottom area with regard to the welded joint axis allows drawing the conclusion that these were not apparent (caused by wrong compensation) but real stresses, and that they appeared due to the difference in heating (different angle of incidence of sun rays on the bend surface). This indicates that sun radiation stresses should be taken into account in the calculations.

4. Dynamic Load Stresses in Bucklings

The values of stresses in circumferential direction were determined by analyzing deformation spectra recorded at various phases of turbine set operation.

The largest stress of 8.0 kN/cm^2 appeared in the tested area for a very short time, with a leap, during an emergency putting the turbine set out of turbine operation.

This situation may be neglected since it occurs rarely. Slightly lower values were recorded during starting the pump operation after opening the butterfly valve and later after closing the guide ring during normal stopping. These stresses were of attenuated, cyclic character with a defined constant frequency. In both cases the first cycle amplitude is about $\pm 7.5 \text{ kN/cm}^2 = 75 \text{ MPa}$, the attenuation ratio is 0.83 and pulse frequency is 0.15 s^{-1} for valve opening and 0.18 s^{-1} for guide ring closing. No influence of head water level on the above values was observed. During starting the turbine set to pump operation mode, after opening the guide ring, the first pulse cycle amplitude did not exceed $\pm 6.0 \text{ kN/cm}^2$. In transient states the amplitudes reached $\pm 3.5 \text{ kN/cm}^2$ with a stochastic character of pulse spectrum. The smallest stress pulsations occurred during steady operation and reached up to $\pm 0.5 \text{ kN/cm}^2$.

Except the emergency putting out of operation, stresses in all tested turbine operation states were significantly smaller than during corresponding pump operation states.

5. Working Stresses in Bucklings

The main stresses resulting from mechanical loads i.e. hydrostatic and hydrodynamic pressure may be computed as an algebraic sum of caused by them components, since they are generated by the same source (static pressure or its pulsations) and therefore will operate in the same directions.

Thermal load main stresses may deflect from these directions, but because of the axial symmetry of the shell, the deflections will be small and may be disregarded.

In this way, the working stresses are an algebraic sum of components of main stresses generated by all the above mentioned loads. When the compensator operates correctly, the longitudinal (axial) stresses are small and practically have no influence on material effort, i.e. on the reduced (equivalent) stresses.

The total value of working stress for various water levels in the upper reservoir (for various hydrostatic pressures) can be easily determined from the curves in Fig. 3; it needs drawing a σ -axis parallel line through the point on the abscissa corresponding to the actual water level in the upper reservoir. Through the point of intersection of this line with the curve of stresses in function of hydrostatic pressure with thermal stresses added, a second line parallel to the abscissa should be drawn. This line is the stress pulsations spectrum which are generated by cyclic changes of hydrodynamic pressure after opening the butterfly valve during putting the turbine set into pump operation. The family of curves shown in the drawing represents the envelopes of these spectra at a given water level in the reservoir.

The point of intersection with the straight line defining the permissible stress

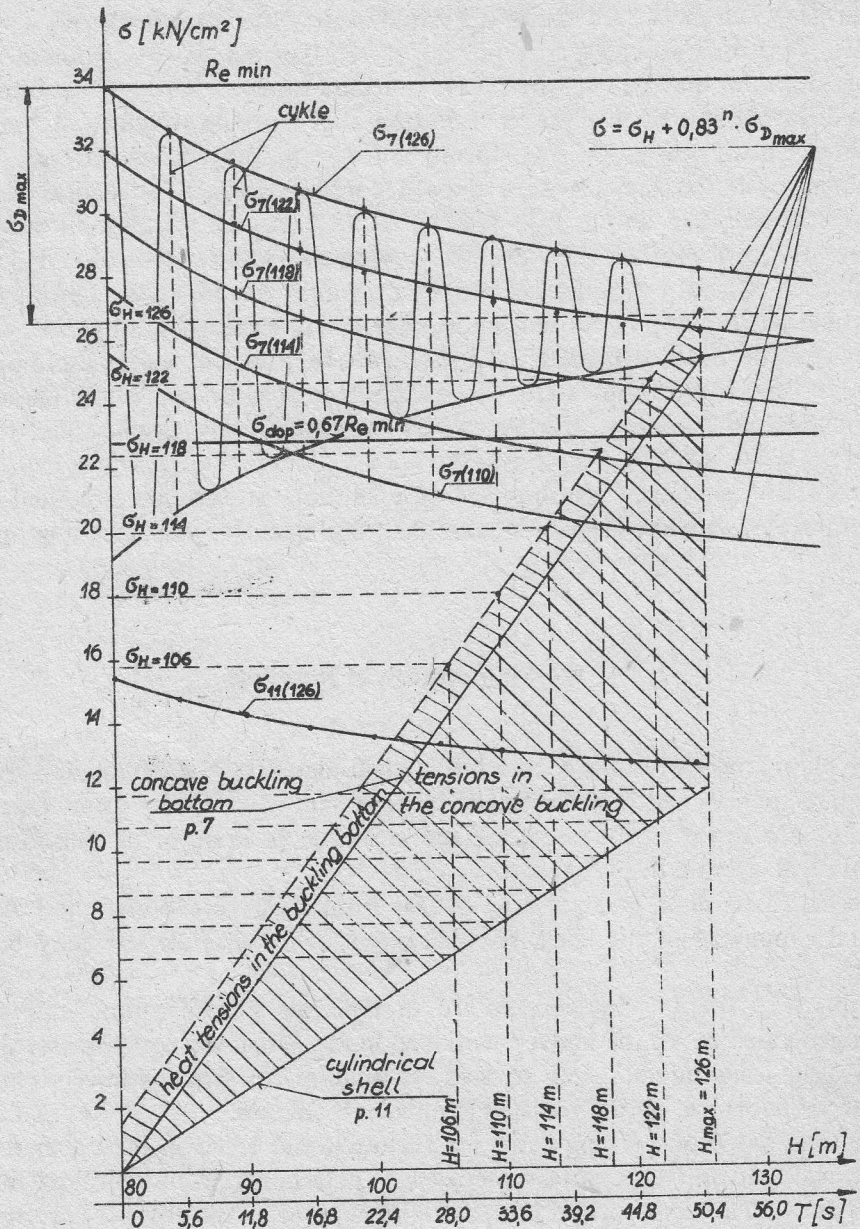


Fig. 3. Stresses at the bottom of a concave buckling in a cylindrical shell generated by hydrostatic, cyclic hydrodynamic and 24-hour thermal loads in the summer season

for a given buckling shell material enables determining the number of cycles which exceed this boundary. This gives a basis for calculating the temporal resistance of the buckling to fatigue, after taking into account the own technological stresses which should be determined by means of additional tests.

References

- [1] J. Ginalski, J. Muszyński, *Naprężenia w załomach wklęsłych powłok cylindrycznych* (Stresses in concave bucklings of cylindrical shells). Dozór Techniczny no 4, 1984.

Przebiegi naprężeń w załomach wklęsłych rurociągów derywacyjnych podczas pracy turbozespołów odwracalnych

Streszczenie

Omówiono mechanizm spiętrzenia naprężeń w obszarze osiowego załomu wklęsłego powłoki cylindrycznej na przykładach płyt: płaskiej (rys. 1) i zakrzywionej (rys. 2).

Z zależności wyprowadzonych dla przyjętych, uproszczonych modeli obciążenia wynika, że największy wpływ na wartość współczynnika spiętrzenia naprężeń wywiera stosunek głębokości załomu do grubości powłoki.

Badania tensometryczne, przeprowadzone na cienkościennym rurociągu wielkośrednicowym w obszarach dna dwóch załomów o różnych profilach i głębokościach wykazały, że jeżeli pod wpływem obciążenia rurociągu ciśnieniem wewnętrznym dna załomów odkształca się plastycznie, to pomimo początkowych różnic głębokości i kształtu profilu, przy następnym cyklu obciążenia maksymalne współczynniki spiętrzenia naprężeń w obydwóch załomach będą sobie równe.

W załomach, poza koncentracją naprężeń od ciśnienia hydrostatycznego i hydrodynamicznego, powstają również naprężenia wywołane zmiennym opromienianiem przez słońce. Tych naprężeń wywołanych obciążeniami termicznymi nie wykryto w obszarach o prawidłowej geometrii powłoki.

Przedstawione graficznie na rysunku 3 przebiegi maksymalnych naprężeń roboczych w obszarach dna załomu powstających podczas uruchomienia turbozespołu do pracy pompowej pozwalają określić liczbę cykli pulsacji naprężeń przekraczających wartości dopuszczalne dla zmęcowanych warunków pracy.

Ходы напряжений во вогнутых загибах деривационных трубопроводов во время работы реверсивных турбоагрегатов

Резюме

Обсуждается механизм концентрации напряжений в зоне осевого вогнутого загиба цилиндрической оболочки на примерах плит: плоской (рис. 1) и искривлённой (рис. 2).

Из зависимостей выведённых для принятых, упрощённых моделей нагрузки следует, что наибольшее влияние на значение коэффициента концентрации напряжений имеет отношение глубины загиба к толщине оболочки.

Тензометрические исследования, произведённые на тонкостенном трубопроводе большого диаметра в зонах dna двух загибов различных профилей и глубин доказали, что если под влиянием нагрузки трубопровода внутренним давлением dna загибов пластически деформируются, то мимо начальных разниц глубины и формы профиля, при очередном цикле нагрузки максимальные значения коэффициентов концентрации напряжений в обоих загибах будут себе равны.

В загибах, кроме концентрации напряжений от гидростатического и гидродинамического напряжений, возникают также напряжения вызываемые переменным солнечным облучением. Эти напряжения, вызванные термической нагрузкой, не обнаружены в зонах характеризующихся правильной геометрией оболочки.

Представленные графически на рис. 3 ходы максимальных рабочих напряжений в зонах dna загиба, возникающих во время пуска турбоагрегата в насосном режиме, позволяют определить количество циклов пульсаций напряжений превышающих значения допустимые для усталостных условий работы.