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POLISH ACADEMY OF SCIENCES

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Aims and Scope

Transactions of the Institute of Fluid-Flow Machinery have primarily been established to publish papers from four disciplines represented at the Institute of Fluid-Flow Machinery of Polish Academy of Sciences, such as:

- Liquid flows in hydraulic machinery including exploitation problems,
- Gas and liquid flows with heat transport, particularly two-phase flows,
- Various aspects of development of plasma and laser engineering,
- Solid mechanics, machine mechanics including exploitation problems.

The periodical, where originally were published papers describing the research conducted at the Institute, has now appeared to be the place for publication of works by authors both from Poland and abroad. A traditional scope of topics has been preserved.
Only original and written in English works are published, which represent both theoretical and applied sciences. All papers are reviewed by two independent referees.

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Editorial

These Special Issues of the *Transactions of Fluid-Flow Machinery*, Nos. 113 and 114 contain selected papers from the International Conference on *Turbines of Large Output* devoted to commemorate 100$^{\text{th}}$ Birthday Anniversary of Prof. Robert Szewalski. The conference was held in Gdańsk, Poland on September 22-24, 2003.

The Conference is a continuation of previous conferences held at the Institute of Fluid-Flow Machinery PAS in former years dedicated to technology of steam turbines. Series of conferences bearing the same name took place in the years 1962, 1965, 1968, 1993. In 1997 organised has been a conference on steam turbines and related topics, but with a slightly amended title – *Problems of Fluid-Flow Machinery.* At present at the Institute of Fluid Flow Machinery there are conduced research of fundamental character encompassing both the issues related to steam turbines and fundamentals of power engineering.

Organisers of the present conference have returned to the traditional name, i.e. Conference on *Turbines of Large Output* to mark the respect to the memory of Professor Robert Szewalski (1903-1993), the founder and a director of the Institute of Fluid-Flow Machinery for several years, and initiator of the conference series devoted to the problems of steam turbines.

The Editors are very grateful to the referees of the papers presented in this issue of the *Transactions of Fluid-Flow Machinery*: J. Badur, W. Batko, E. S. Burka, J.T. Cieśliński, P. Doerffer, A. Gardzilewicz, B. Grochal, J. Kiciński, G. Kosman, T. Król, J. Krzyżanowski, J. Mikielewicz, A. Neyman, W. Ostachowicz, R. Puzyrewski, R Rządkowski, J. Świrydczuk, M. Trela, T. Uhl and Z. Walczyk.

We appreciate cordially the manifested authors interest in our conference. Special thanks are conveyed to Ms 'Maya' Bagińska, for her fruitful editorial help related to directing the papers to reviewers and correspondence with the authors of contributions.

Prof. Jarosław Mikielewicz
Editor-in-Chief

Dr Edward Śliwicki
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TRANSACTIONS OF THE INSTITUTE OF FLUID-FLOW MACHINERY

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Control valves for turbines of large output

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Abstract

Intensive vibration of inlet piping system was discovered during test period of 1000 MW SKODA turbine operating with inlet-saturated steam. Unwanted oscillations of the piping system required new view on construction of control parts of the turbine. Intensive research of valve models took place, numerical simulation were done. The original alternative of the valve with shaped plug was compared with the valve characterized by flat-bottom-plug and a muffler.

Keywords: Control valves; Vibrations; Experiments; CFD methods

1 Introduction

The introductory test of the 1000 MW SKODA steam turbine in Temelin showed some insufficiencies of technical character, which made reliable operation of the turbine in lower output levels impossible. Intensive vibration of inlet piping system in wide spéctrum of frequencies was the most important among them [1]. The existence of supersonic flows in the valves and numerous shock state changes [2] were identified to be the main cause of this problem. Valves were made according to extrnal license. They had been used in operation and no serious problems were expected. The valves had been tested in wind tunnel too [3]. Improved flow and force characteristics for states of steam close to the nominal regime were reserved. Measurements with flow conditions modeling small turbine loads are technically difficult and in laboratory conditions not easy to realize. Only after the first test of whole energetic system in power plant dangerously high intensity of vibration in the inlet piping was discovered.

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The steam turbines of 1000 MW in the nuclear power plant Temelin are required to be suitable for longtime service at no load run and with an output of around 50 MW for home consumption. Further requirement is a supply of electric energy to a so called "island network" which, as a rule, needs lower than full turbine output. That means that the required operation for a longtime service is within a range of 0-1000 MW. To meet given requirement meant completely redesign the valve. Leading specialists in fluid dynamics and dynamics of solids material firmness cooperated on solution of the problem. Many measurements on the valves in the power plant and on models in laboratories took place. Numerical simulation of the flow was done to verify the structure of the valve inner flow. The result of the development is new shape of the plug and a muffler on the seat of diffuser. Subject of this study is comparison of both types of valves.

2 Valve with shaped plug

Typical shape of the original valve is shown in Fig. 1. If the opening of the control valves is small, a convergent-divergent channel is created between the plug and the seat. In this so-called Laval nozzle during supersonic flow system of shock waves is created.

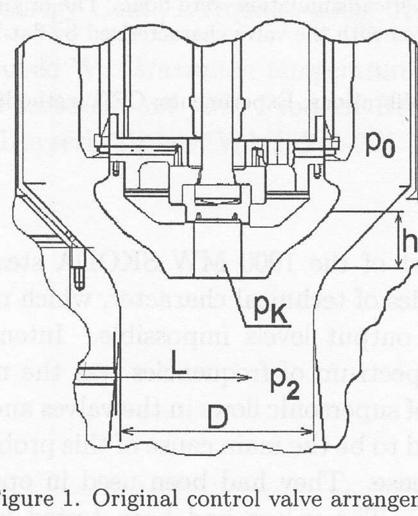


Figure 1. Original control valve arrangement.

Disturbances are spreaded in steam and in metal with the speed of sound. Shock waves are unstable and oscillate with high frequency. Also movement of shock waves in whole annular jet must affect the 3D flow. The vortices in wakes behind the plug will be unstable too. Sum of all disturbances causes vibration in wide spectrum of frequencies.

The frequency characteristic of pulsating pressure, which was measured in

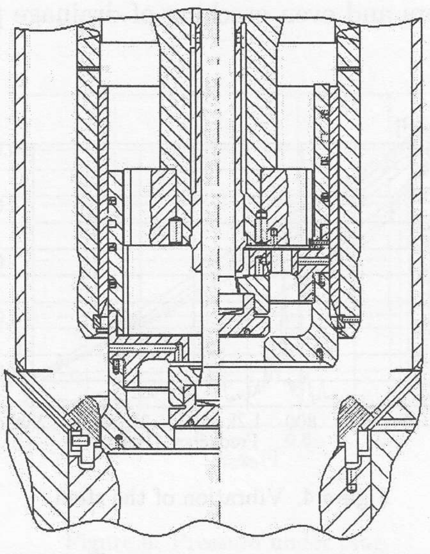


Figure 2. Redesigned control valve with muffler.

the inlet piping while the turbine was at idle run, is depicted in Fig. 3. The steam pressure in front of the control valve was 6.3 MPa and behind the valve was around 0.6 MPa. It seems that the values of the pulsating pressure within the range of lower frequencies (up to 350 Hz) reach maximum value of 270 kPa. Intervals with high values of the pulsating pressure reach a maximum at 340 kPa within a range of high frequencies above 500 Hz.

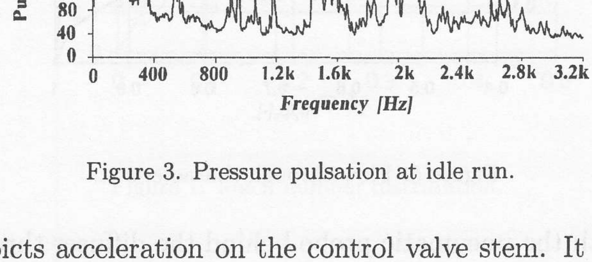


Figure 3. Pressure pulsation at idle run.

Figure 4 depicts acceleration on the control valve stem. It reaches up to 20 m/s^2 at frequency of 900 Hz. It is no exception for the vibration of the piping itself to reach values up to 180 m/s^2 when the turbine is at idle run. Rapid damage of accelerometers and even cracking of drainage piping occurred at this intensity of vibration.

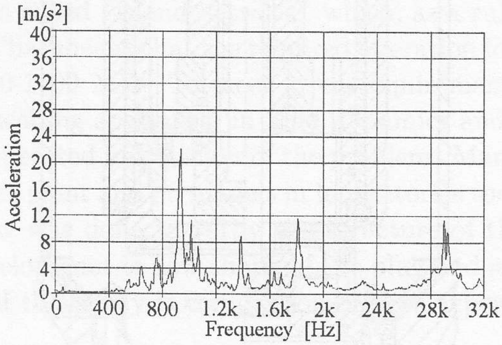


Figure 4. Vibration of the stem.

Figure 5 shows how turbine output and mass flow ratio changes with different valve pressure ratios. Only at output of around 650 MW, the conditions for under critical flow are created. Supersonic flow still occurs under the plug and in the diffuser throat. It is clear from Fig. 6, where the pressure under the plug p_k is shown. Local supersonic flow can occur on covergent – divergent nozzle even if the overall valve pressure ratio is already under critical. The control valve is very closed at low turbine outputs. The flow area in the valve is becoming larger. Also separation of flow from the wall connected with occurrence of recirculation flow appears besides shock waves.

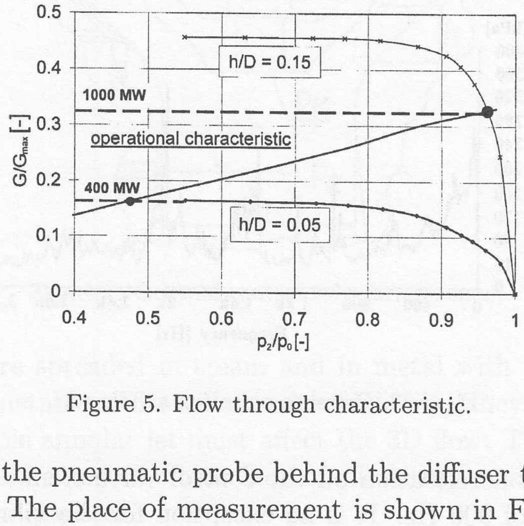


Figure 5. Flow through characteristic.

Traverses with the pneumatic probe behind the diffuser throat were done on a valve model [4]. The place of measurement is shown in Fig. 1. Velocity field depends on the pressure ratio and on the relative opening of the valve. Mach

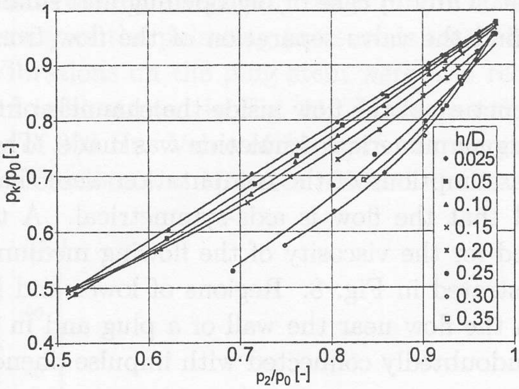


Figure 6. Pressure under plug.

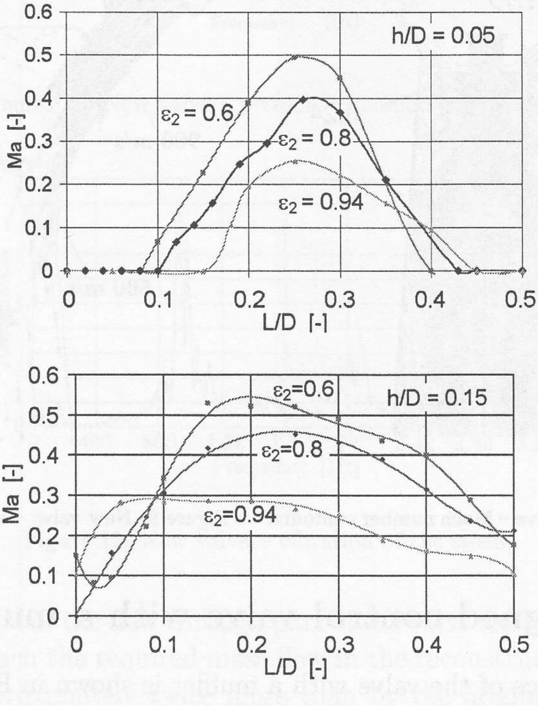


Figure 7. Mach number distribution.

number distribution for two relative lifts of plug and three pressure ratios is shown in Fig. 7. The flow has a typical structure of a core flow with a separation of flow from the seat. Only in case of big opening and small pressure difference in front of and behind the valve separation of the flow from the seat does not occur.

To verify the structure of the flow inside the channels of the control valve according to LMZ design a numerical simulation was made. The code FLUENT 5.3 was used [5]. The assumptions of the computation were that the flowing steam is an ideal gas and that the flow is axis-symmetrical. A turbulence model of RNG $k-\varepsilon$ was used for the viscosity of the flowing medium. The result of the computation is illustrated in Fig. 8. Regions of lower and higher transonic velocities alternate in the flow near the wall of a plug and in the core flow in the diffuser, which is undoubtedly connected with impulse phenomena.

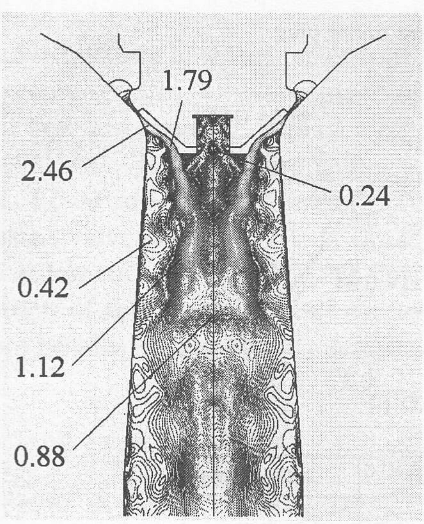


Figure 8. Original valve – Mach number contours.

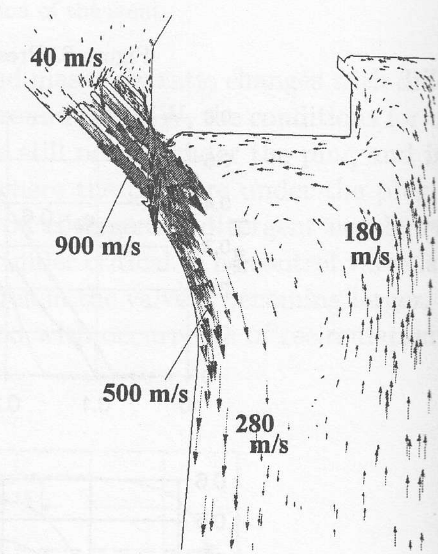


Figure 9. New valve – velocity vectors.

3 A redesigned control valve with a muffler

The appearance of the valve with a muffler is shown in Fig. 2. The plug has a flat bottom and the muffler is created by perforated ring. The muffler has a conical shape and covers only a part of the flow cross section in order to disturb the flow through the control valve as little as possible.

The changed design of the valve led to outstanding delay of piping vibration and pressure pulsation. The spectrum of the pulsating pressure of the idling turbine is show in Fig. 10. It reaches a maximum of pulsating pressure of 70 kPa inside the low-frequencies range (up to 350 Hz). The maximum of the pulsating pressure is 18 kPa inside the high-frequency range (above 500 Hz). A considerable lowering of the intensity of the pulsating pressure is reached inside the range of high frequencies. Vibrations on the plug stem were also reduced. Acceleration measured in idle run is illustrated in Fig. 11. It reaches the maximum of only 2 m/s^2 at frequency of 900 Hz. Value of 14 m/s^2 was recorded on the drainage piping. Intensity of vibrations was 10 times reduced.

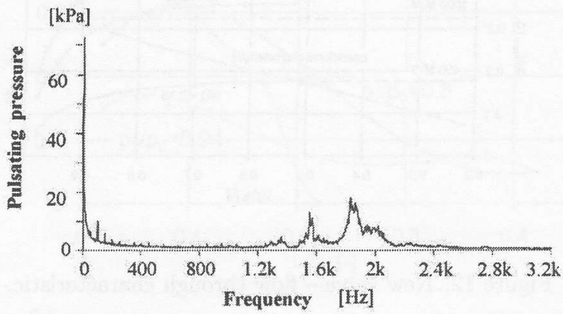


Figure 10. New valve – pressure pulsation of idle run.

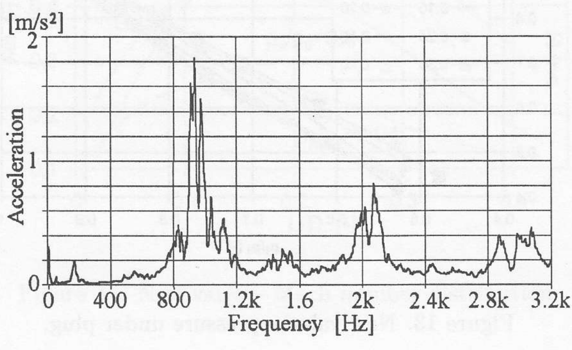


Figure 11. New valve – vibration of the steam.

The modification of the flow through part of the valve led to increased opening of the valve. To reach the required mass flow in the reconstructed valve, the plug must be lifted approximately twice more than by the original valve. Relations among the output of the turbine, valve pressure ratio and relative lifting of the plug are illustrated in Fig. 12. The plug is driven in the muffler up to the output of 700 MW. *The steam is forced to flow through the muffler in whole range of overcritical pressure drops. In this case, the annular flow of steam behind the muffler is slowed down and remains attached to the wall of the diffuser. In all* operational regimes *the pressure under the plug is kept lower than the pressure* behind the valve, as shown in Fig. 13. In the central part of the diffuser and under the plug occurs recirculation flow. The main stream of the steam remains attached to the wall of the diffuser. Annular vortex occurs under the plug even if the valve is opened to the maximum. The throut of the diffuser is almost unaffected by the annular vortex.

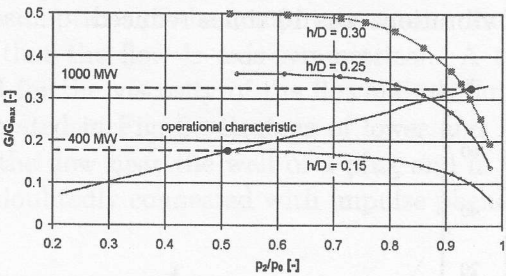


Figure 12. New valve – flow through characteristic.

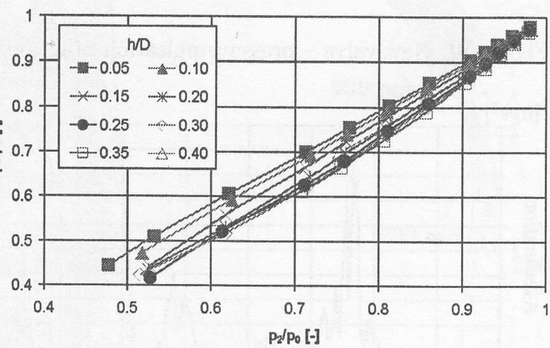
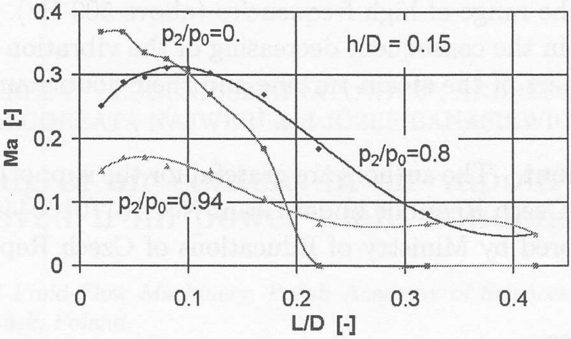


Figure 13. New valve – pressure under plug.

Measurement of velocity fields in the cross section close so the diffuser throat took part [6]. Distribution of Mach numbers for three pressure ratios and two plug openings is illustrated in Fig. 14. Relative opening $h/D = 0.15$ corresponds to opening $h/D = 0.05$ for the shaped plug without muffler. The changes in Mach number distribution are obvious. The flow remains attached to the diffuser wall for all pressure ratios in the valve. For opening $h/D = 0.3$ and pressure ratio $p_2/p_0 = 0.94$ is the velocity profile in flow through area very well balanced.

The flow field in the redesigned control valve was studied numerically with the help of the FLUENT 5.5 code as a 3D flow including the holes in the muffler. The wet steam was replaced again by an ideal gas and its viscosity was considered with the help of a RSM turbulence model. The Fig. 9 shows some of the results of the computation for the opening of a control valve corresponding to the turbine output of 300 MW. The muffler is circa 3/4 opened during this regime. The inlet pressure of the steam is 6.2 MPa and behind the valve the pressure is 2.7 MPa. The flow through the control valve is overall supersonic with a maximum local value of Mach number at 2.4 on the outside edge of the plug.



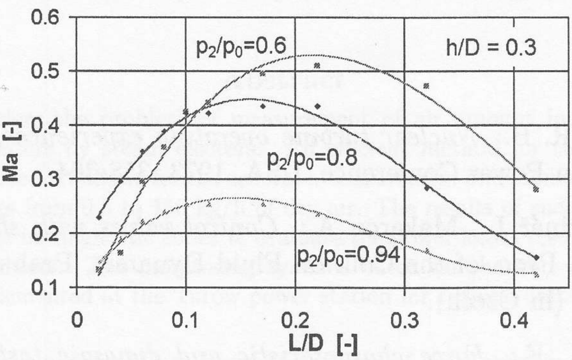


Figure 14. New valve – Mach number distribution.

4 Conclusions

The source of the high-frequency pressure pulsation in the steam flow in HP part at small outputs of a steam turbine 1000 MW SKODA at Temelin nuclear power plant was an unsuitable flow structure in the channels of the control valves. Complicated flow fields with non stationary interactions of shock waves and vortex area along the periphery of the diffuser were created at the core flow in the diffuser. The cause of the core flow was the shape of the plug of the original valve according to the license LMZ, Petersburg from 1983 resulting in a creation of a convergent-divergent channel between the plug and the seat when the control valve was only slightly opened. The pulsating pressure of the flowing steam resulted in the following: intensive vibrations of the inlet piping system and of the control valves and a high noise level.

The redesign of the control valve brought these main improvements of the original situation:

The maximum of the pulsating pressure of the steam in the inlet piping with the turbine idling decrease 3.9 times in the range of low frequencies (to 350 Hz), and 19 times in the range of high frequencies (above 500 Hz).

This resulted in the consequent decreasing of the vibration of the inlet piping system into HP part of the steam turbine and their double amplitude decreased 2.4 times.

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