Next-generation balanced throttle control valves for steam turbines

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Abstract

Control valves are immediate regulating devices in control systems of steam turbines. As of now, high aerodynamic drag and low dynamic reliability are the most notorious operational shortcomings of control valves. This paper surveys newly developed control valve designs ensuring high reliability and economy of steam turbine control components. Findings from simulation studies are presented. Our studies involved mathematical and physical simulation of new valve designs and were concerned with determining their flow-rate, force and vibration behavior. Several proposals for improving valve reliability and economy have been put forward. Among them, the greatest efficiency is promised by new layouts of steam-actuated valving and the use of finned wide-angle diffusers with cylindrical entry sections downstream of valve seats. Compared to known designs, next-generation balanced valves feature structural changes: shaped axially symmetrical confusor duct, enveloping surface of the spool and inlet section of the diffuser seat featuring perforation bands that open into a common dampening chamber offset toward the interior of the actuator (spool), an axial force balancing system that is disengaged automatically when the spool is lifted higher than 50% of its travel range, the use of longitudinally finned wide-angle valve diffusers, and a cylindrical section added downstream of the seat confusor section to improve flow uniformity upstream of the tapered diffuser. The designs make it possible to decrease almost 20% of hydraulic resistance inherent in control valves. The increase in efficiency of electric power production is followed by a decrease in specific fuel consumption in a power plant by about 0.2%. As a result, emission of carbon dioxide in the atmosphere decreases significantly.
Keywords: steam turbine, control valve, diffuser, pressure pulsations, vibration, throttling, steam-actuated valving.

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1 Introduction

Valves on an electric power plant are classified as auxiliary equipment yet their proper functioning is crucial for normal operation of all major process units. This is precisely why power circuit valves face numerous and quite rigorous requirements of which high availability and durability are paramount. Control valves are also required to minimize hydraulic resistance when the actuator is in the fully open position.

In many instances the compliance of valves with these requirements is directly determined by the behavior of process medium flowing through their pass-through sections [1]. It is unfortunate that the issue of valve optimization from a fluid dynamics standpoint was only receiving cursory attention until now. As a result, more than hundred-year old structural designs of angle valves, gates, safety and cut-off valves only experience minor evolutionary changes.

In its simplest version, the existing throttle control valve design comprises a body with inlet and outlet flanges having a diffuser seat inside that can be occluded by spherical, disc-like or shaped spool integral with valve stem (spindle) coupled in turn with a spool actuator system [2].

In valves of this type, the flow between the inlet and outlet/vent flanges makes a 90° turn inside the valve box. Each such turn induces a complex non-stationary velocity field featuring significant circular irregularity as it approaches the seat. If conditions inside the valve box prove favorable, intense circulation develops around the valve stem.

Consequently, the spool and the valve stem alike bear large dynamic loads, the diffuser effect in the diffuser seat diminishes, valve throughput degrades while hydraulic losses with a fully open spool rise.

In addition, if the spool is designed to have a large contact area with the seat and initial medium flow is intense enough, a spool lifted to a low height subjects the stem to extreme axial forces making it difficult to move.

The subsequent presentation will cover several innovative design improvements of pass-through sections in throttle control valves.

2 A survey of special design features in next-generation balanced control valves

Our innovative angular valve design was driven by the need to minimize the impact of the above-named factors on durability and functional reliability of control valves at hand. As a consequence, the structural appearance of these valves underwent dramatic changes.

A typical design of such valve with an axial force balancing system is shown in fig. 1 [2].

The valve spool is comprised of two parts. Its enveloping surface in contact with flowing medium features a shaped design with an internal cavity that opens toward the enveloping surface through three rows of perforation holes. This part of the spool is joined by buttress thread with a cylindrical hub with two wedge keys preventing the spool from rotating around its longitudinal axis when operating in a twisted process medium flow. In order to prevent the cylindrical spool section from directly contacting process medium, the spool moves inside a closed cup joined rigidly to a box. The shaped part of the spool features an orifice inside with a central discharge hole blocked by valve discharge integral with stem. Axial movement of the discharge valve is limited by a transverse spacer in the cylindrical spool section so that further axial movement of the stem makes the spool open.

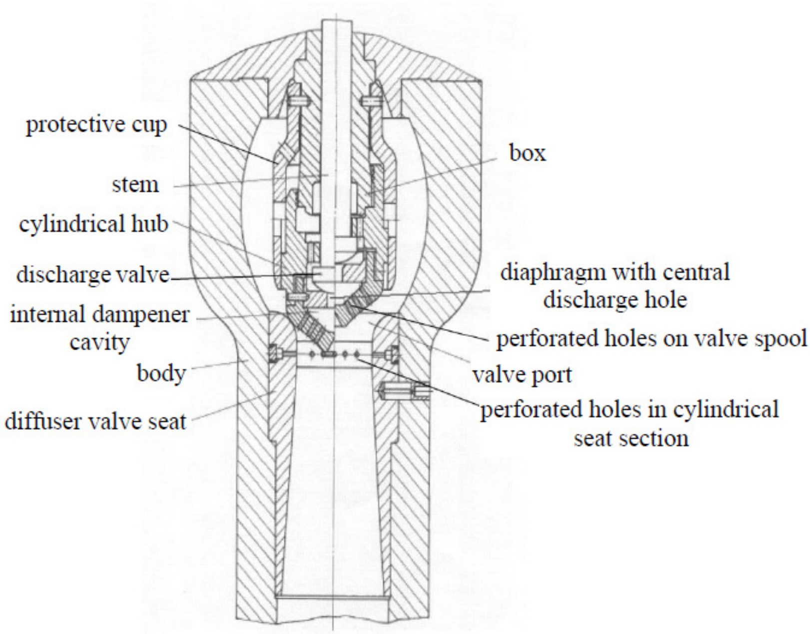


Figure 1: Balanced control valve with perforated enveloping surfaces of the spool and seat inlet section.

The valve seat comprises a streamlined confusor inlet part that forms a circular axially symmetrical duct together with the shaped surface. The confusor part of the seat touches its cylindrical part having a single row of perforated holes and followed by a tapered diffuser with pass-through section opening angle $\alpha = 7°$.

This valve incorporates an innovative system for balancing axial forces. A particular design feature is that the discharge valve and its seat are located inside the spool. This means that when the discharge valve is opened and the spool is forced to move, steam from the internal cavity instead of being driven toward the center of the diffuser seat passes into the dampening cavity whence it is discharged through perforated holes into the confusor valve port thus causing minimum disruption to the main flow of the process medium.

As the valve spool lifts, pressure downstream of the valve rises producing a sharp drop in the force that was pressing the spool to the head of the steam. This creates a situation where even modest fluctuations of pressure under the valve may cause the spool to lose its axial stability, bringing about a real risk of self-sustained oscillations within the free-travel range of the discharge valve. As a means of preventing oscillations of this kind, when the spool is lifted above 50–60% of its total travel range the bottom part of the box is introduced into a cylindrical borehole in the partition wall of cylindrical hub blocking steam flow to the discharge valve. This inhibits the balancing of axial forces on the valve and maintains its spool tightly pressed against the spool head across the entire spool travel range.

We complete our description of the valve by pointing out the role played by perforated holes that appear both on enveloping surfaces of the spool and the cylindrical seat section. This solution with perforated holes short-circuited into a common dampening chamber inhibits circumferential unevenness of flow and breaks the rigid link between pressure fluctuations in the flow and the magnitude of dynamic forces exerted by these fluctuations on the valve stem. Further, with a narrow spool opening and a high pressure drop on the valve stem, perforated surface will promote intense dissipation of the wave structure of supersonic flow.

Several modifications have been devised using this valve design as the base. As an example, fig. 2 shows a balanced angle valve with a differently shaped three-part spool.

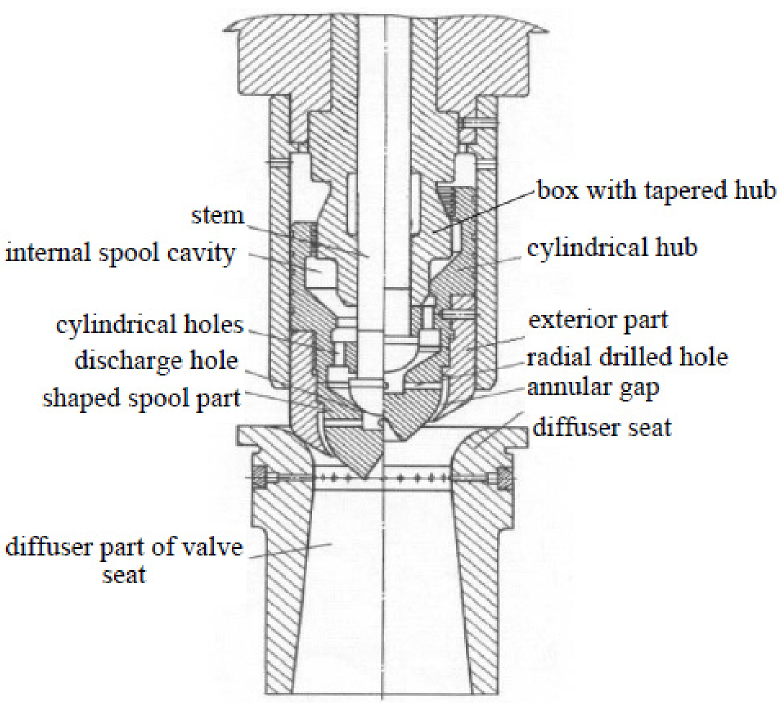


Figure 2: Balanced control valve with a tangential system for process medium evacuation from the internal cavity.

In this case, its enveloping section consists of an exterior part around a complex shaped spool section. Its inner part features a discharge hole blocked by the discharge valve. However, in this case the process medium is evacuated from the internal spool cavity through a row of radial holes and annular gap. Such a solution serves the primary purpose of enabling letting an extra flow in along walls of internal enveloped part of the spool described here. This decreases the likelihood that the flow will detach from the enveloping surface with low-lifted spool. Similar to the case shown in fig. 1, both halves of the compound spool mate with cylindrical hub.

Another angle valve design based on the initial version (fig. 1) is shown in fig. 3 [3]. Only the discharge valve opening system was changed in this case. While, for valves shown in fig. 1 and fig. 2, the discharge valve moves in axial direction within the free travel range and the spool hangs off its face freely, pressed into it by the medium, the valve shown in fig. 3 features cylindrical discharge valve with sectoral cuts at the bottom face. Similar to the cases described above, the cylindrical discharge valve is integral with stem but is installed between orifice and partition wall with a minimum gap only enough for the discharge valve to rotate. Such rotation will connect discharge holes of orifice to the internal cavity of the spool letting process medium into the pass-through section of the valve.

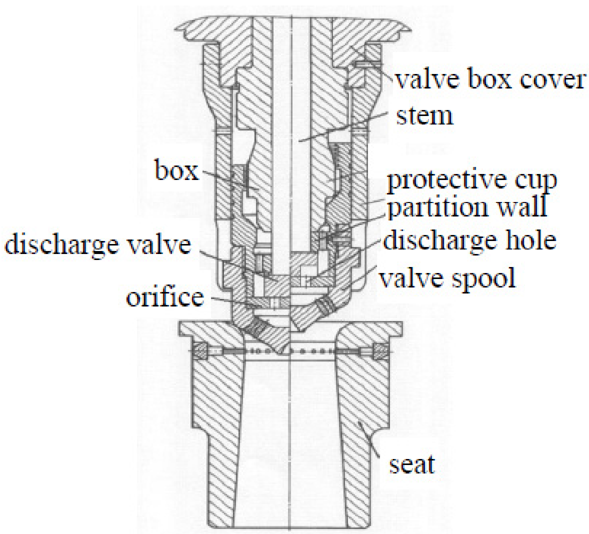


Figure 3: Balanced control valve with rotary discharge valve.

The absence of a gap between the discharge valve and the main spool virtually rules out self-sustained oscillations of the discharge valve.

The discharge valve is rotated using a special rotary coupling joining the valve stem with its actuator system.

All three valve options share identical flow-rate, force and vibration behavior.

3 Study methodology for balanced control valves: plotting flow-rate, force and vibration behavior

Fig. 4 shows a flow-rate curve characteristic of these valves where dimensionless medium flow rate q is expressed as a ratio of actual mass flow rate G to the theoretical maximum (critical) flow through a narrow seat cross-section G_*.

$$G_* = A \cdot \frac{P_0}{\sqrt{T_0}} \cdot F_1,$$
$$q = \frac{G}{G_*} = \frac{G \cdot \sqrt{T_0}}{A \cdot P_0 \cdot F_1} \tag{1}$$

where, P_0 – initial pressure of the process medium;
T_0 – initial temperature of the process medium;
F_1 – area of narrow seat section;
A – constant (for air: A = 0.0404, for superheated steam: A = 0.0311).

$$F_1 = \frac{\pi \cdot D_1}{4} \tag{2}$$

where, D_1 – diameter of narrow seat section.

The horizontal axis in fig. 4 shows dimensionless pressure expressed as a ratio between pressure P_2 downstream of diffuser seat to full braking pressure P_0 upstream of valve.

$$\varepsilon_2 = \frac{P_2}{P_0} \tag{3}$$

Curves in fig. 4 have been plotted for three spool positions determined by its dimensional lifting height $\bar{h}$ and two diffuser seats having identical expansion n = 2.2 but different opening angles of the pass-through section equal to α = 7° and α = 10°.

$$\bar{h} = \frac{h}{D_1} \tag{4}$$

Some advantages of the seat design with angle α = 7° become evident here if valve throughput with low-lifted spool ($\bar{h}$ = 0.154) is considered. If the spool is lifted further ($\bar{h}$ = 0.231 and $\bar{h}$ = 0.308) within the range of relative low velocities (ε_2 > 0.95) the two properties compared here become virtually identical. However our measurements of pressure fluctuations downstream of the seat together with measurements of dynamic forces acting on valve stem have shown that an increase in diffuser seat opening angle α from 7° to 10° is accompanied with intense increases both of pressure pulsations and magnitudes of dynamic forces acting on the stem of the valve at hand.

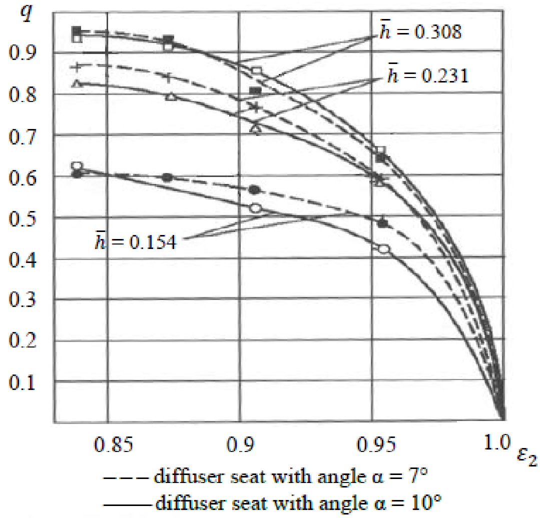


Figure 4: Flow-rate behavior of control valve series under study.

A typical force behavior curve for our series of innovative balanced angle valve is shown in fig. 5 where the dimensionless force $\overline{Q_0}$ acting on zero-diameter stem is plotted as a function of absolute spool lifting height h at various pressure drops ε_2 acting on the stem.

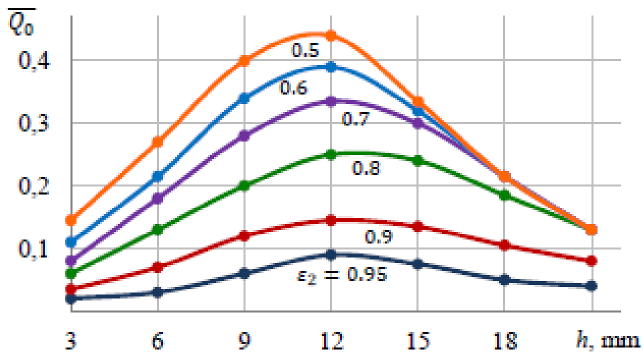


Figure 5: Typical force behavior of innovative control valves.

The dimensionless actual force $\overline{Q_0}$ comprises a ratio of the actual force ($Q_0 = Q - \frac{\pi}{4}\mathrm{d}_{st}^2 \cdot (P_0 - B)$) adjusted for zero stem diameter to the maximum force Q_{max} exerted by process medium pressure that presses the spool to the seat at zero pressure P_2 downstream of the valve.

$$Q_{max} = \frac{\pi}{4} \cdot D_b^2 \cdot P_0 \quad (5)$$

In this case $\overline{Q_0}$ behavior is considered for a valve with a diffuser seat having a narrow section diameter D_b = 80 mm.

$$\overline{Q_0} = f(\varepsilon_2, h) \quad (6)$$

As follows from relations shown above, once the spool becomes detached from the seat surface, the force required to shift the spool increases sharply, as flow accelerates rapidly at the inlet section of shaped spool surface, consequently reducing the pressure on the enveloping surface of the spool significantly below pressure P_2 downstream of diffuser seat. This increase in force is observed roughly until the spool travels halfway of its total range ($\bar{h} \cong 0.150$). Later, as the spool leaves the active flow zone, this force declines rather quickly.

Time charts of forces produced by simulation studies provide some understanding of dynamic forces applied to stems of innovative shaped discharge valves.

Time charts shown in fig. 6 have been obtained at various spool positions and various pressure drops (ε_2 = var) to show that dynamic forces on stems are negligible compared to all known angle valve designs. In essence these time charts testify to remarkable effectiveness of the fluctuation dampening system in the pass-through section of these valves.

4 Control valves with wide-angle diffuser seats

As already noted above, from vibration reliability standpoint it would be reasonable to opt for control valve diffuser seats having opening angle α smaller than 7°. However, size constraints in this case will result in low ratios of diffuser channel expansion $n = F_2/F_1$ (where F_2 is the outlet section area of the seat, F_1 is the inlet section area), rarely exceeding 1.8 ÷ 2.

These constraints inhibit pressure recovery in diffuser seats greatly, hampering complete utilization of the diffuser effect that would help to decrease the hydraulic resistance of angle control valves.

As special research [4, 5] indicates, this issue may be addressed by choosing diffuser seats with wedge-shaped finning of the enveloped surface.

Such a method of affecting flow behavior in flat diffusers was first given detailed treatment in [5] where it was shown that longitudinal finning of the enveloped surface of a flat diffuser with angle α = 15–20° based on 10 mm trapezoid fins caused its vibration condition to be identical to that of a flat diffuser with α = 7°.

However, [5] did not fully address the option of using longitudinal finning in axially symmetrical diffuser valve seats and circular diffusers of gas turbines where severe circumferential and radial flow irregularity may be compounded by intense twisting of the process medium flow.

Extended studies of wide-angle axially symmetrical diffusers with uneven inlet velocity fields at different flow vortexing conditions have highlighted the high efficiency of longitudinal finning in axially symmetric as well, regardless of the inlet velocity field profile.

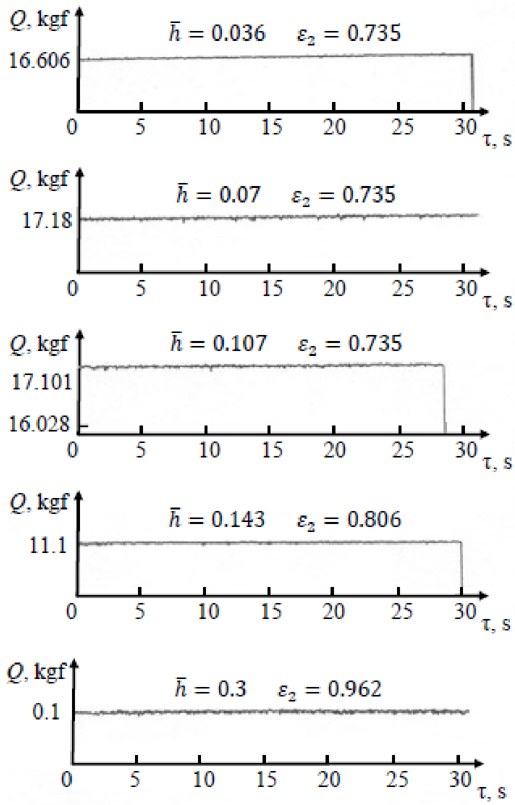


Figure 6: Simulated time charts of forces applied to stems of next-generation balanced valves.

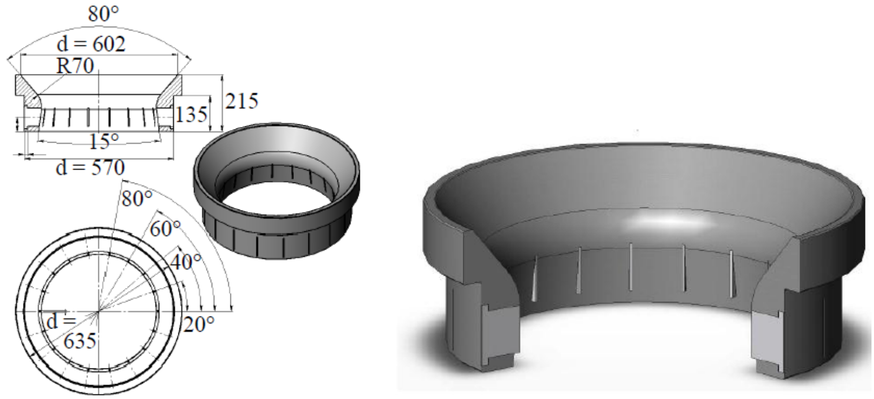
Of all fin shapes surveyed in [4] best results were obtained with the use of wedge-shaped fins with wedge top angle β computed as

$$\beta = arctg\frac{0.4 \div 0.45}{Re_L^{0,2}} \tag{7}$$

where, Re_L – Reynolds number.

$$Re_L = \frac{c_1 \cdot L}{\nu} \tag{8}$$

where, c_1 – average design medium velocity at diffuser inlet;
L – diffuser length;
ν – kinematic viscosity factor;
A – diffuser seat of a control valve with finning of this type is shown in fig. 7.



(a) (b)

Figure 7: Control valve seats for K-1000-60/1500 turbines shown with machined slots for insertion of wedge-shaped fins (a) and with fins pressed in (b) [5].

The efficiency of the proposed approach to dealing with increased vibration of wide-angle diffusers is illustrated by curves plotted in fig. 8.

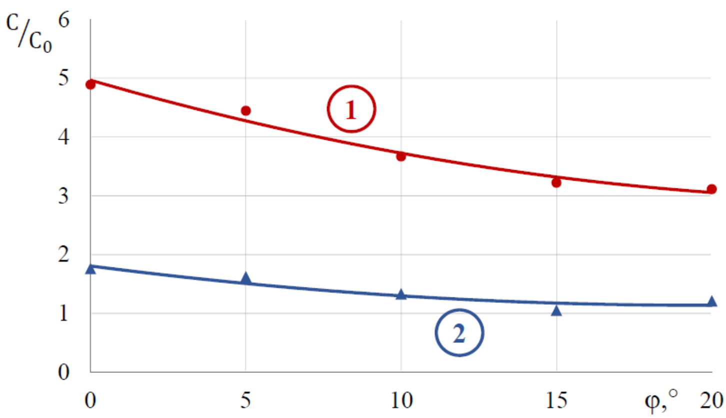


Figure 8: The effect of wedge-shaped finning on the vibration behavior of wide-angle diffuser seat in innovative valves.

This illustrates the change in relative vibration velocities measured at diffuser valves at their outlet when no fins are present (curve 1) and when the enveloped surface has wedge-shaped finning (curve 2) with flow twisting angles of 0° (axial inlet flow) to 20°. Diffusers chosen for comparison have identical geometry (generatrices opening at an angle $\alpha = 15°$, expansion factor $n = 4$).

To facilitate understanding, all measured vibration velocity readings c have been shown relative to their values in a finless diffuser with $\alpha = 7°$ and no twisting of flow in the inlet section c_0.

The plots show that vibration velocities declined by factors of 2.7 to 3 after wedge-shaped fins were installed.

It should be noted that wedge-shaped fins have failed to bring down pressure recovery ratio in the wide-angle diffuser under study, the ratio remaining at about 73–75% at any flow twisting angles. These findings warrant special attention from fluid dynamics standpoint. It is a consequence of the fact that wedge-shaped fins do not disrupt flow in diffuser inlet section. As a result, two flow patterns become established in its pass-through part.

Axial flow develops at walls where fins protrude to a height commensurate with the boundary layer thickness, while the main flow pattern (whether twisted or not) develops away from the fins. When fins are present, areas adjacent to walls are sheltered by the fins from the main flow, serving as a kind of “liquid screen” for the latter to prevent direct contact between twisted flow and diffuser walls.

As a result, total energy losses have been found to change almost identically depending on flow twisting angle both with and without the above-mentioned fins in pass-through sections of diffusers.

These findings are significant for control valves as they make it possible to expand the pass-through section area of existing seats almost twice without increasing their axial length, thereby doing away with almost 20% of hydraulic resistance inherent in such control valves.

Control valves of steam turbines are executive bodies of regulation system. They provide stability of operating modes of the turbine and its safety in emergencies depends on their action. Presented valves allow to exclude emergence on rods of valves of noticeable dynamic efforts, having provided thereby increase of vibration reliability of the considered type of the control valves. As a result, reliability of whole power plant. So relative vibration velocities for the developed valves at installation of wedge-shaped finning decreased by 2.7–3 times. Decrease in vibrations thus increased reliability of the turbine plant in general.

5 Conclusions

According to the above research we can draw the following conclusions:

1. Compared to known designs, next-generation balanced valves feature structural changes: shaped axially symmetrical confusor duct (when the spool is fully open), enveloping surface of the spool and inlet section of diffuser seat featuring perforation bands that open into a common damper chamber offset toward the interior of the actuator (spool), a system for relieving the valve of axial forces that is disengaged automatically when the spool is lifted higher than 50% of its travel range, the use of longitudinally finned wide-angle valve diffusers and a cylindrical section added downstream of seat confusor section to improve flow uniformity upstream of tapered diffuser.

2. The introduction of highly efficient aerodynamically shaped dampers in these valves has virtually eliminated the action of aerodynamic forces on valve stems, conferring increased vibration reliability to this kind of control valves.

3. The design featuring a rotary balanced valve is of special interest as a basis for further improvement of control valve reliability.

4. Usually losses in control valves of the existing turbines make about 3%. The application of the developed valves leads to decrease in losses of a vapor pressure to 2.5 %. Decrease in hydraulic resistance leads to increase in efficiency of power generation. The increase in efficiency of the electric power producing is followed by decrease in specific fuel consumption on power plant about 0.2% as a result emission of carbon dioxide in the atmosphere decreases significantly.

Acknowledgements

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