Contemporary Engineering Sciences, Vol. 7, 2014, no. 34, 1827 - 1837
HIKARI Ltd, www.m-hikari.com
http://dx.doi.org/10.12988/ces.2014.410192

Summary of Approaches for Improving Throughput of Low-pressure Cylinders in Steam Turbines Using Two-tier Stages

Arkady Zaryankin

National Research University "Moscow Power Engineering Institute"
Krasnokazarmennaya Street 14, Moscow, Russian Federation

Nikolay Rogalev

National Research University "Moscow Power Engineering Institute"
Krasnokazarmennaya Street 14, Moscow, Russian Federation

Andrey Rogalev

National Research University "Moscow Power Engineering Institute"
Krasnokazarmennaya Street 14, Moscow, Russian Federation

Ivan Garanin

National Research University "Moscow Power Engineering Institute"
Krasnokazarmennaya Street 14, Moscow, Russian Federation

Wadim Strielkowski

Charles University in Prague, Faculty of Social Sciences,
Smetanovo nábř. 6, 110 01, Praha 1, Prague, Czech Republic

Copyright © 2014 Arkady Zaryankin, Nikolay Rogalev, Andrey Rogalev, Ivan Garanin, and Wadim Strielkowski. This is an open access article distributed under the Creative Commons Attribution License, which permits unrestricted use, distribution, and reproduction in any medium, provided the original work is properly cited.

Abstract

The throughput of single-flow condensation steam turbines is determined by the throughput of their final stages, which in turn is mainly determined by the length of the blades at the final stage.
At the same time, blade dimensions have reached their limits by now, facing strength constraints.
This paper examines a totally new design of a two-tier stage which can be applied to improve throughput of low-pressure cylinders in heavy-duty steam turbines by transitioning to two-tier low-pressure cylinders based on such stages. A distinctive feature of two-tier cylinder is that a part of steam flowing through the upper tier bypasses the final stage and enters the condenser.
Two-tier low-pressure cylinders have the potential of significantly improving the power of condensation steam turbines.

Keywords: heavy-duty steam turbine, two-tier stage, two-tier low-pressure cylinder

1 Introduction

When the issue of improving economy of steam turbine cycles is considered, there is a clearly discernible trend toward increasing initial steam parameters even though that necessitates the use of more complex manufacturing technology for high-temperature turbines [2, 5, 13].

At the same time, the goal of increasing thermal efficiency of the Rankine cycle can be met just as well by achieving deeper vacuum in the condenser which is generally a matter of controlling cooling water temperature at condenser inlet.

At turbine power ratings below 800 MW with the final stage blade length l = 960 mm the vacuum is equal to P_K = 3,5 kPa. For more powerful turbines this degree of vacuum only becomes attainable with final-stage blade lengths increased to between 1200 and 1315 mm. These complications ultimately necessitate increasing vacuum in the K-1000-65 type turbine to 6 kPa, as lower condenser pressures result in a marked increase in specific steam flow volumes [3]. As pointed out by other researchers [1, 4, 7, 11], the problem of letting through a large volumetric flow is faced most seriously by designers of heavy-duty turbines for nuclear power plant. For example, at P_K = 6 kPa the specific steam volume is 23.7 m^3/kg, but if the pressure in condenser is brought down to 3.5 kPa, the specific volume of dry saturated steam would rise to 35.5 m^3/kg. As a result, a 50% increase in volumetric steam flow rate requires that the final stage end face area must be increased by almost the same magnitude.

These considerations ultimately lead to an obvious conclusion that the options for exploiting thermodynamic advantages of high vacuum are restricted to either finding an approach for increasing the throughput capacity of turbine LPCs; or adding more low-pressure cylinders; or transitioning to two-tier steam turbines; or (if the length of final-stage blades were to be kept constant) using high vacuum only for steam turbines rated below 300 MW. Possible options for improving throughput of low-pressure cylinders in condensation steam turbines are explored in [6, 8, 9, 10, 11, 12, 14, 15, 16].

The complexity of problems that arise with the use of high vacuum in steam turbine condensers can be mitigated significantly by opting for two-tier low-pressure cylinder designs that promise a 50% increase in LPC throughput with final-stage blades lengths unchanged, enabling heavy-duty steam turbine units to operate at a vacuum of 3 to 3.5 kPa.

The options for these two-tier LPC designs are explored in the present paper.

2 Factors determining the maximum steam flow rate through the final stage of a single-flow steam turbine

If a single-flow low-pressure cylinder is used, the relationship between the steam input to the condenser G_C and end face area of the final stage F_Z is determined as a flow ratio

$$G_C = \frac{C_{az} \cdot F_Z}{V_Z} = \frac{M^*_{az} \cdot a_Z \cdot F_Z}{V_Z} \tag{1}$$

In the equation (1) V_Z is the specific volume of steam downstream of final stage, a_Z is the speed of sound in saturated steam, M_{az}^* is the Mach number beyond which the resistance of exhaust manifold of a condensation steam turbine begins to rise sharply. Most exhaust manifolds of such turbines have $M_{az}^* = 0.65$. Aerodynamically perfect exhaust manifolds would have $M_{az}^* = 0.75$. $M_{az}^* = 0.82$ is feasible in rare cases.

Assuming a fixed maximum allowable flow-averaged number M_{az}^*, the complex $M^*_{az} \cdot \frac{a_Z}{V_Z} = \varphi(P_Z)$ will become a one-to-one function of pressure P_Z downstream of final stage of condensation turbine.

Further, as computations show, this function is linear relative to the pressure P_Z i.e.

$$\varphi(P_Z) = a + b \cdot P_Z \tag{2}$$

and $$G_C = \varphi(P_Z) \cdot F_Z = (a + b \cdot P_Z) \cdot F_Z \tag{3}$$

At $M_{az}^* = 0.65$

$$\varphi(P_Z) \cdot F_Z = 0{,}14 + 192{,}7 \cdot P_Z \tag{4}$$

At $M_{az}^* = 0.75$

$$\varphi(P_Z)\cdot F_Z = 0{,}16 + 222{,}37\cdot P_Z \tag{5}$$

In the formulas above, the pressure P_Z is dimensioned in bars and the area F_Z is dimensioned in square meters.

For P_Z = 0.035 bar and M_{az}^* = 0.65, $G_C = 6.88F_z$. If P_Z is increased to 0.06 bar, the result is $G_C = 11.7F_Z$.. For an aerodynamically perfect exhaust manifold (M_{az}^* = 0.75), $G_C = 13.5F_Z$ at P_Z = 0.06 bar.

In turn, the face area of final LPC stage is determined by the diameter D_Z and the length l_Z of blades at this stage ($F_Z = \pi\cdot D_Z\cdot l_Z$).

For a blade length l_Z = 1200 mm – as is the case with heavy-duty power turbines produced by LMZ (now OJSC Power Machines) in Russia – and an average diameter D_Z = 3000 mm, the area F_Z = 11.3 m^2.

Accordingly, the maximum permissible steam flow rate through such a stage will vary between G_C = 77.74 kg/s at P_Z = 0.035 bar and G_C = 128.7 kg/s at P_Z = 0.06 bar for standard-design exhaust manifolds (M_{az}^* = 0.65).

If the blade length is increased to 1320 mm at D_Z = 3320 mm and 1500 mm at D_Z = 3800 mm, the above-stated peak flow rate values may be increased to G_C = 94.7÷161 kg/s and G_C = 123÷209.5 kg/s, respectively. There is a drawback to this solution however, as high-speed turbines (n = 50 rpm) will face severe problems with ensuring reliable operation of stages that have blades designed this way.

The alternative solution to the problem at hand involves switching to two-tier stages of which Baumann's stages are representative. In this case, if an LMZ stage with blade length l_Z = 1200 mm is used as the base design, the area of LPC exhaust will increase from F_Z = 11.3 m^2 to F_Z = 16.95 m^2 (at P_Z = 0.035 bar) and will become comparable to a similar area of a stage having blade length l_Z = 1500 mm at D_Z = 3800 mm.

The deficiencies inherent in Baumann's stage design can be eliminated by switching to two-tier LPCs with new two-tier stages where upper-tier guide blade assemblies are not a continuation of the bottom-tier blades. Consequently, it is possible to speak of two separate stages stacked bottom-to-top.

3 Structural and aerodynamic features of new two-tier stages

Figure 1 shows the two-tier stage design serving as the basis for a new-generation two-tier LPC design with a condenser input capacity of 178 kg/s at a pressure P_C = 0.04 bar.

In this case the guide vane assembly *1* of this lower-tier stage comprises 48 vanes shaped by nozzle profiles with a chord dimension equal to 220 mm in root section.

The top-tier guide vane assembly *2* comprises 120 stages with a root chord dimension of 120 mm. Consequently, the bottom-tier impeller *3* houses 36 twisted blades whereas the top-tier impeller *4* has 108 blades in total. The two stages are separated by baffles *5* and *6*.

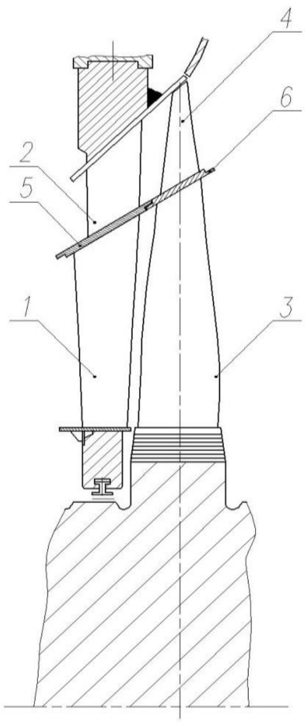

Fig. 1: A two-tier stage design

The baffle *5* minimizes cross-flow between the stages in question by closing above the impeller baffle *6* with a radial opening of 2 mm (Figure 2).

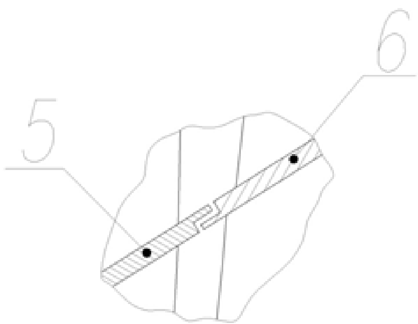

Fig. 2: Baffles reducing cross-flows in a two-tier stage

Process-wise the feather of a bottom-tier stage blade serves as a carrier for three top-tier impeller profiles. Figure 3 shows such a fork blade in axonometric prospective.

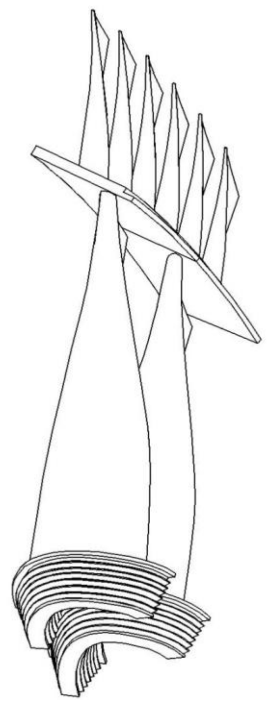

Fig. 3: Fork blade of a two-tier stage

A blade measures 250 mm along the chord of its root section. This dimension ensures that blade areas experiencing the highest loads are not overstressed beyond strength constraints. A blade at the stage in question measures 1060 mm in total.

Two-tier LPC designs shown in Figure 4 feature four two-tier stages and a single (final) single-tier stage entering into the flow channel of the bottom cylinder tier.

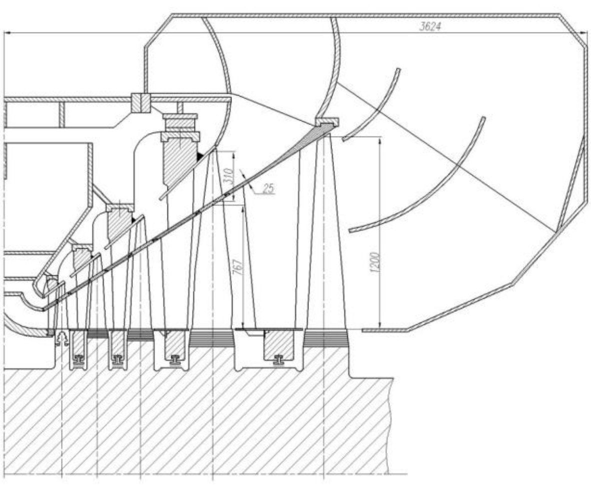

Fig. 4: Two-tier low-pressure cylinder

Thus, in this instance of low-pressure cylinder, the available enthalpy drop is exploited at optimum values of the kinematic parameter $X_a = \frac{u_i}{c_f^i}$ (*u* is the circumferential speed while $c_f^i = \sqrt{2 \cdot \Delta h_i}$ is the theoretical speed corresponding to the enthalpy drop at each stage) at the four top-tier stages and four bottom-tier stages of the cylinder.

4 Findings from mathematical simulation of steam flow in a two-tier stage of a two-tier LPC

Flows in the flow channel of the new LPC for heavy-duty condensation steam turbines have been simulated numerically with the goal of plotting velocity fields in the flow channel of said cylinder and identifying its integral properties. Owing to space constraints in this report, we will only indicate the numerical simulation data for the heaviest-loaded final two-tier stage depicted in Figure 1.

Simulation was performed with the Ansys CFX computational software suite using the k-e turbulence model. The computational grid for vane channels of a two-tier stage comprises 8 million tetrahedral elements.

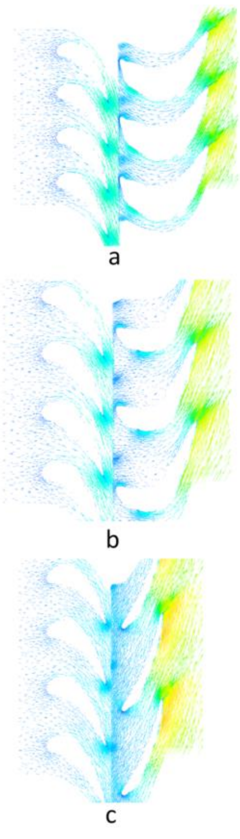

Fig. 5: Velocity fields for the bottom tier of a two-tier stage

Figure 5 illustrates findings from our simulation of the bottom tier of the stage at hand. Shown there are the resulting velocity fields for the root (5a), medium (5b) and peripheral (5c) sections. Velocity fields within the working grid in these figures are plotted as relative quantities. Considering that the stage at hand has been designed with a high reactivity that rises intensely toward peripheral sections, the steam velocity w_2 also increases outward. Consequently, exhaust velocities c_1 at the guide vane assembly decrease in the outward direction. It should be mentioned that the flow leaves the guide vane assembly at an angle rather perfectly correlated to the angle β_1 at the blade inlet. As a result, none of the steam flow patterns shown here have any areas of flow detaching from streamlined surfaces. High efficiency of the bottom-tier vanes is thereby ensured for the stage in question. Our computations show that this efficiency is η_av = 92.8%

In comparison, the efficiency of the penultimate LPC stage in a K-800-23.5/50 turbine is 92.4% [5].

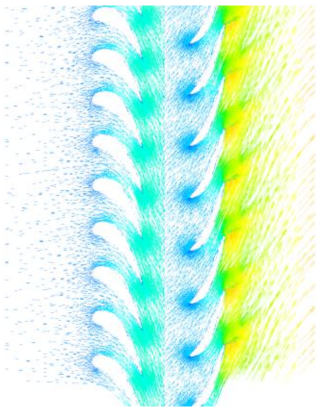

Fig. 6: Velocity fields for the top tier of a two-tier stage

Steam flow pattern in the flow channel of the upper tier of a two-tier stage is similar to that of the lower tier and can be illustrated as a velocity field in the -section of this tier (Figure 6). In this case, extremely high circumferential steam velocities in the mid-section of the top-tier guide vane assembly result in flow entering into blades at an angle β_1 of 131° while departing from the relatively moving impeller at an angle β_2 = 33°. Similar to the bottom tier of the stage, the flow remains attached. However, final losses have increased due to top-tier + s being shorter in absolute and relative terms. In addition, the flow departing the top tier of the final stage completely loses its energy. This has brought about a drop in vane efficiency of the upper tier to 79.6%. As a result, averaged in mass terms, the vane efficiency of the final two-tier stage is η_av = 86.8%.

The general vane efficiency of a two-tier LPC is 92.2%.

5 Conclusions

1. Our survey of the options for increasing steam throughput through low-pressure

cylinders of condensation steam turbines points to the transition toward double-stage LPC design based on two-tier stages as the most promising approach to this problem at present.

2. A new type of the two-tier stage is proposed comprising a combination of two separate stages where the exterior contour of the bottom-tier stage comprises the root contour of the top-tier stage with its guide vane independent from the bottom tier.

3. A two-tier fork-type blade has been designed for shaping two-tier stages. In this design, the separating baffle of a single bottom-tier blade supports two to three top-tier blades.

4. Our numerical simulation of steam flow in our two-tier stage design has enabled us to arrive at velocity and pressure fields and to determine the efficiency of this stage which has turned out to be 0.8% lower than that of a standard one-tier stage with 1200 mm long rotor blades.

Acknowledgements: This study conducted by National Research University “Moscow Power Engineering Institute” has been sponsored financially by the Russian Federation through the Ministry of Education and Science of the Russian Federation under Subsidy Agreement # 14.577.21.0072 of June 5th 2014 as a part of the Federal Targeted Programme for R & D in Priority Fields for the Development of Russia’s S & T Complex for 2014-2020. Applied Scientific Research Unique Identifier: RFMEFI57714X0072

References

[1] V. F. Abramovskii, Yu. F. Kosyak. Experience in developing steam turbines for atomic power stations at the turboatom productive combine and prospects for future improvements. *Soviet Energy Technology*, 1 (1986), 26-31.

[2] G. S. Booras, N. Holt, R. Viswanathan. Economic analysis of new coal-fired generation options. *Proceedings of the 4th International Conference on Advances in Materials Technology for Fossil Power Plants*. South Carolina, USA, 2004.

[3] V. A. Buzlukov, M. G. Teplitskii, A. A. Maksimenko, T. V. Poruchinskii. Full-scale testing of the KhTZ k-1000-60/1500-2 turbine plant at Zaporozh's nuclear power station. Thermal Engineering, 36 (1989), 69 - 75.

[4] M. Haraguchi, T. Nakamura, H. Yoda, H., T. Kudo, S. Senoo. Nuclear steam turbine with 60 INCH last stage blade. *Proceedings of the 21st International Conference on Nuclear Engineering.* Chengdu, China, 2013.
http://dx.doi.org/10.1115/ICONE21-16600

[5] A. Kostyuk, V. Frolov. *Steam and gas turbines*. Mir Publishers, Moscow, 1988, p. 342.

[6] L. Li, J.-D. Yang, W. You, X.-M. Wu, Z.-P. Feng. Investigation of the vapour-liquid two-phase flow in the low-pressure cylinder of a 1000 MW nuclear power steam turbine. Journal of Power and Energy, 228 (2014), 178 - 185. http://dx.doi.org/10.1177/0957650913511800

[7] T. Nakamura, K. Segawa, N. Isobe, T. Saito. Highly-reliable design technology of steam turbine for nuclear power plant. *Hitachi Review*, 58 (2009), 102 - 106.

[8] V. I. Nishnevich, M. B. Banuradze, Yu. S. Sachkov, P. V. Khrabrov, Yu. A. Porfiriev, V. M. Zhits, A. D. Kantemir, Yu. P. Tomkov, Yu. F. Kosyak, B. A. Arkadiev, Yu.E. Yushkevich, Ya.L. Iglinskiy. Patent of USSR No 819364, (1981).

[9] L. P. Safonov, V. K. Ryzhkov, Yu. A. Marchenko, E. V. Simdyanov. Investigating the low-pressure cylinders of LMZ high-capacity high-speed steam turbines with blades of the last stages 1000 and 1200 mm long. *Thermal Engineering*, 33 (1986), 661 - 664.

[10] Yu. E. Semenov, N. A. Korableva, G. E. Cherkashina. Determination of the Economic Effectiveness of Modernization of the Low Press Cylinder of the K-200-130 Turbine Made by Leningrad Metal Plant. *Teploenergetika*, 8 (1975), 13 - 15.

[11] E. V. Simdyanov, N. M. Zhuravlev, B. A. Golovkin. Studies of model low-pressure cylinders of low-speed and high-speed 1000-mw turbine sets. *Soviet Energy Technology*, 3 (1985), 8 - 13.

[12] B. M. Troyanovskij. New design of the low-pressure cylinders of Mitsubishi steam turbines. *Thermal Engineering*, 50 (2003), 170 - 172.

[13] P. S. Weitzel. A steam generator for 700C to 760C advanced ultra-supercritical design and plant arrangement: What stays the same and what needs to change. *Proceedings from the 7th International Conference on Advances in Materials Technology for Fossil Power Plants.* Waikoloa, Hawaii, USA, 2013.

[14] A. E. Zaryankin S. V. Arianov, V. A. Zaryankin, A. V. Pavlov. Prospects of using two-tier low-pressure cylinders in steam-turbine power units. *Thermal Engineering*, 56 (2009), 50 - 56. http://dx.doi.org/10.1134/S0040601509010091

[15] A. Zaryankin, N. Rogalev, G. Kurdiukova, A. Rogalev, E. Lisin, W. Strielkowski. Nuclear power plants with super-powerful high-temperature steam

turbine. *Contemporary Engineering Sciences*, 7 (2014), 457 - 468.
http://dx.doi.org/10.12988/ces.2014.4330

[16] A. E. Zaryankin, V. A. Zaryankin, S. V. Arianov. *A flow path for a steam turbine*. Patent of RF No 2296224, (2007).

Received: October 17, 2014; Published: December 3, 2014