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Two-tier low pressure cylinders for condensing steam turbines

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Abstract

The paper deals with new construction of low pressure cylinder (LPC) for condensing steam turbines. The flow path of these cylinders is formed on the base of new two-tier stages. As opposed to well known Baumann's stages newer two-tier stages is a combination of two independent stages with own blades set. Such construction allows to decrease the number of LPC in the existing turbines or to ensure their operation with very high vacuum in the condenser or increases in 1.5 times power capacity of new turbines without increasing of the last stage blade length and LPC number.

Keywords: Two-tier stage; Two-tier low pressure cylinder; Fork-shaped blade

1 Introduction

In consideration of the steam turbine cycles efficiency increase problem the tendency is further initial steam parameters growth. This tendency leads to application of considerably complex technologies of high-temperature steam turbines production. Notable growth of Rankine cycle efficiency may be attained at the expense of vacuum increase which essentially determined by the cooling water temperature at the condenser inlet [5].

In case of turbine power capacity lower than 800 MW with last stage blade length l equal to 960 mm, designed vacuum, P_c, is 3.5 kPa. For the more powerful steam turbine ensuring of vacuum on the level mentioned above appears feasible in case of last stage blade length increase up to 1200–1315 mm only. In the final analysis this fact result in increasing pressure in the condenser of turbines

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K-1000-65 up to 6 kPa as pressure in the condenser decreasing leads to intensive growth of steam specific volume. For example specific volume of saturated steam at pressure in the condenser $P_c = 6$ kPa is 23.7 m^3/kg and if the pressure P_c equals to 3.5 kPa that the value of specific volume increases up to 35.5 m^3/kg. Thereby steam volume flow rate increase by 50% requires the last stage exit area extension by the same value.

So for the usage of high vacuum thermodynamic advantages it is necessary to fined methods of turbine low pressure cylinders capacity or increases the number of low pressure cylinder (LPC) or passes to two-tier steam turbines or uses high vacuum for the steam turbines with power capacity lower than 300 MW (in case of application of the last stage blade length without change). The passage to the two-tier low pressure cylinders allows to considerably decrease the complexity of incipient problems associated with high vacuum application in the turbine condensers. These cylinders ensure growth of the LPC capacity by 50% without last stage blade length increase. This fact makes possible the operation of powerful steam turbines with vacuum 3–3.5 kPa [1–3].

This paper deals with the possibilities of such two-tier LPC creation.

2 Factors which determines the ultimate steam flow trough the last stage of a single-flow steam turbine

In case of single-flow low pressure cylinder application the connection between steam flow to the condenser, G_c, and last stage exit area is defined by following expression:

$$G_c = \frac{C_{az} F_z}{v_z} = \frac{M_{az}^* a_z F_{z'}}{v_z}\,, \tag{1}$$

where: v_z – specific volume of steam after the last stage, a_z – velocity of sound in dry-saturated steam, M_{az}^* – Mach number, in which exhaust manifold resistance of steam turbine begins to increase sharply. For the most exhaust manifolds of these turbines the value of M_{az}^* equals to 0.65. For the aerodynamically perfect exhaust manifolds M_{az}^* equals to 0.75. In a rate instances the value of M_{az}^* equals to 0.85.

At fixed value of maximum permissible weight-average M_{az}^* complex $M_{az}^* \frac{a_z}{v_z} = \varphi(P_z)$ is one-valued function of pressure P_z after the condensing turbine last stage and as calculation shows this function is linear in pressure P_z:

$$\varphi(P_z) = a + bP_z\ ,$$

$$G_c = \varphi(P_z) F_z = (a + bP_z) F_z\,. \tag{2}$$

In case of $M_{az}^* = 0.65$

$$\varphi(P_z) = 0.14 + 192.7\, P_z\,, \tag{3}$$

and in case of $M^*_{az} = 0.75$

$$\varphi(P_z) = 0.16 + 222.37\, P_z\,. \tag{4}$$

In the formulas mentioned above pressure P_z should be substituted in kPa and area in square meters.

For $P_z = 3.5$ kPa and $M^*_{az} = 0.65$ mass flow to the condenser, G_c, equals to $6.88\, F_z$. In case of P_z increase up to 6 kPa G_c equals to $11.7\, F_z$. If aerodynamically perfect exhaust manifold is used ($M^*_{az} = 0.75$) that $G_c = 13.5\, F_z$ with $P_z = 6$ kPa.

By-turn the exhaust area of the last stage is defined by mean diameter D^m_z and blade length l_z of this stage ($F_z = \pi D^m_z l_z$). If last blade length l_z equals to 1200 mm and mean diameter equals to 3000 mm that exhaust area F_z is 11.3 m^2.

Maximum limit mass flow of steam through such stage may change from $G_c =$ 77.74 kg/s (at $P_c = 3.5$ kPa) to $G_c = 128.7$ kg/s (at $P_c = 6$ kPa) for standard exhaust manifolds($M^*_{az} = 65$ kPa). In case of last stage blades increase up to 1320 mm with $D^m_z = 3320$ mm and up to 1500 mm with $D^m_z = 3800$ mm the maximum limit mass flow may be increased up to $G_c = 94.7$–161 kg/s and $G_c =$ 123–209.5 kg/s respectively. However application of such last stage blades in high-speed turbines ($n = 3000$ rev/min) connects with problems of its' reliable operating ensuring.

Alternative solution of the problem under consideration is the passage to two-tier stages. The typical example of two tier stage is Baumann stage. In case of LMZ (Leningradsky Metallichesky Zavod) last stage with a blade length $l_z =$ 1200 mm usage as a base that exhaust area F_z may be increased from 11.3 m^2 up to 16.95 m^2 (with $P_z = 3.5$ kPa). This value is commensurable with exhaust area of stage with blade length $l_z = 1500$ mm and $D^m_z = 3800$ mm.

Disadvantages inherent to Baumann stage may be removed in case of passage to two-tier low pressure cylinders with two-tier stages where blades of top tier is not a prolongation of lower tier blades. It's talked about two independent stages which is situated one above the other.

3 Constructive and aerodynamically features of new two-tier stages

New two-tier stage construction of new two-tier stage low pressure cylinder with steam mass flow into condenser equal to 178 kg/s with $P_c = 4$ kPa is shown in Fig. 1.

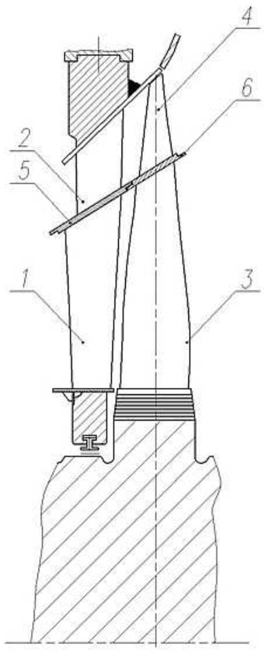

Figure 1. Two-tier stage.

The lower tier nozzle block 1 of this stage consists of 48 vanes with profile chord in vane root section equal to 220 mm. The top tier nozzle block 2 has 120 vanes with chord 120 mm. Rotor wheel of two-tier stage includes 36 twisted blades in lower tier 3 and 108 blades in a top tier 4. Top tier and lower tier is separated by partitions 5 and 6. For decrease of stem leakage from lower to top tier partition 5 of nozzle block overlaps partition 6 of a rotor wheel 6 with radial clearance 2 mm (Fig. 2). On the technological side lower tier blade is a carrier element for three blades of rotor wheel top tier. Isometric view of such 'fork-shaped' blade is presented in Fig. 3. The blade profile chord in a root section of rotor blade equals to 250 mm. Such chord ensure permissible stress in the most loaded regions of rotor blades. The total blade length in concerned stage is 1060 mm.

There are four two tier stages and one single tier stage (last stage), which is included in the lower tier flow path in the two-tier low pressure cylinder. New two-tier low pressure cylinder is shown in Fig. 4.

Thereby available enthalpy drop is utilized at optimal values of kinematic parameter $X_f = u_i/C_f$ (u_i – circumferential velocity, $C_f = \sqrt{(2\,\Delta h)}$ – theoretical velocity which corresponds to available enthalpy drop (Δh) for each stage) in four stages of top tier and five stages of lower tier.

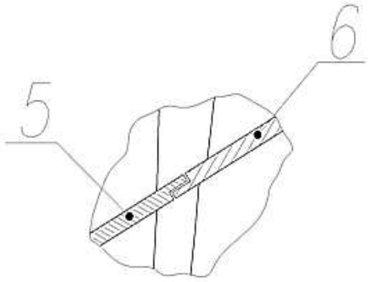

Figure 2. Partitions for steam leakage decrease in two-tier stage.

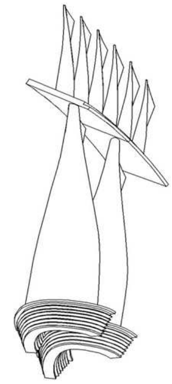

Figure 3. Fork-shaped blade of two-tier stage.

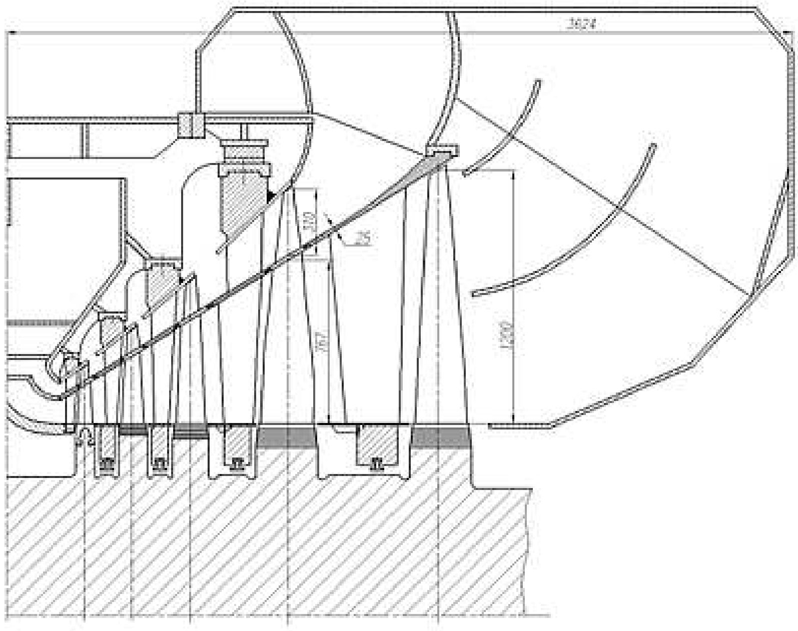

Figure 4. Two-tier low pressure cylinder.

4 Steam flow simulation results in two-tier stage of new two-tier low pressure cylinder

For the estimation of velocity field in newer two-tier stage LPC flow path of powerful condensing turbine and determination of its' integral characteristics mathematical modeling in LPC flow path was made. The greatest interest are simulation results of last two-tier stage which is presented in Fig. 1.

The simulation was realized in computational fluid dynamics (CFD) with k-$\in$ turbulence model application. Finite-element mesh for blade channel of two-tier stage consists of 8 million tetrahedral volumes.

Results of flow simulation in stage lower tier are presented in Fig. 5 where velocity fields for the root 5a, average 5b and peripheral 5c sections. On these pictures velocity fields in rotor wheel are presented in relative motion. Since concerned stage is constructed with high reaction degree which growths strongly in the direction of peripheral section that steam velocity w_2 increases actively in the mentioned direction. Thereafter exit velocities from nozzle block decreases in the peripheral direction. It should be noted sufficiently good coincidence of flow outlet angles from the nozzle block with inlet angles to rotor wheel β_1. As a result there are no flow separation areas in the blade channels. This fact predetermines rather high blade efficiency of concerned stage lower tier. Its' value accordingly to performed calculations is $\eta_{rb}^{lt} = 92.8\%$. For example, LPC last stage efficiency of turbine K-800-23.5/50 LMZ is 92.4% [4]. Flow pattern in the top tier flow

a)

b)

c)

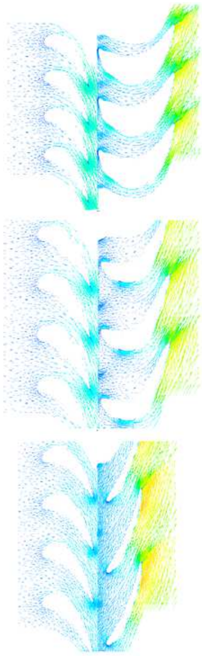

Figure 5. Velocity fields in lower tier of two-tier stage.

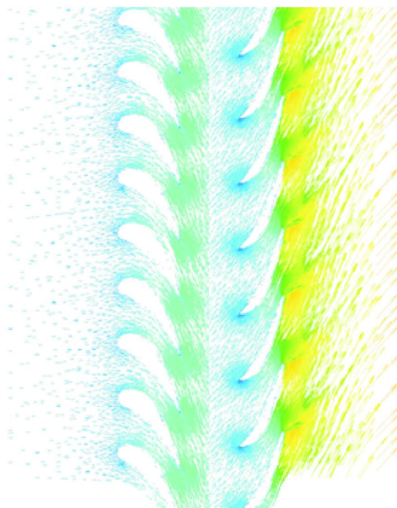

Figure 6. Velocity fields in lower tier of two-tier stage.

path of two tier stage is similar to flow pattern in the lower tier and illustrates by velocity field in the average section of these tier (Fig. 6). In this case very high circumferential velocity in the average section of top tier blade row results that inlet angle to the rotor wheel β_1 is 131° and the outlet angle from the rotor blades in relative motion β_2 equals to 33°. There is no flow separation in the top tier as well as in the lower tier of stage. However end losses turned out higher because of smaller absolute and relative blade length in the top tier. Besides, the energy of output flow in the top tier is lost fully. So blade efficiency of top tier goes down to 79.6%. As a result the average blade efficiency of the last two tier stage η_{rC}^{lt} is 86.8%. The total blade efficiency of two tier low pressure cylinder is 92.2%.

5 Conclusion

1. Overview of possible variants of steam flow increasing though the low pressure cylinder of condensing steam turbines shows that at present the passage to the two tier LPC constructed on the base of two tier stages is the best prospect solution of this problem.
2. Newer type of two-tier stages is offered. Two tier stage is a combination of two independent stages where lower tier shroud is a hub of top tier stage with independent set from lower tier blade.
3. For the two tier stage design the fork-shape two tier blade was constructed where on the separating partition of one blade of lower tier three blades of top tier are placed.
4. Simulation of steam flow in the constructed stage has allowed to get velocity and pressure fields and to estimate stage efficiency which appears by 0.2% lower than efficiency of standard one tier stage with blade length 1200 mm.

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