TURBINES

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Contents

Summary

The history of the development of hydraulic, steam and gas turbines and turbine units is

briefly stated. From general common positions the principles of work of turbines of various types are considered, their classification is given. Basic thermodynamic and kinematic parameters of the one-stage turbine are shown. The change of parameters in a turbine stage is shown. The equations for definition of parameters of a turbine stage are given; the losses influencing the specific work and efficiency of a stage are specified. The necessity of application of multistage turbines for the purpose of increasing their efficiency is shown. The necessity of change of parameters with height of a flowing part is shown and the laws developed for such change are given. Some characteristics of turbines giving representation about change of their parameters at change of operating modes are shown. The typical parameters of steam turbines are given; the characteristics of condensing and other turbines are described. Some thermodynamic features of modern steam turbine cycles are shown. Ways of increasing the efficiency of steam turbine units are considered: increase of the initial parameters of steam, application of steam reheat, reduction of steam pressure in the condenser etc. The features of gas turbines and of gas turbine units are shown. The thermodynamic bases of design of gas turbine units are stated. The comparison of gas turbine units working on the simple thermodynamic cycle, on a cycle with regeneration of heat and multi-modular units is given. It is shown that the combined units with steam turbine and gas turbines are characterized by the highest parameters. The common feature of all modern gas turbines – the necessity of cooling of the most heated up and most loaded details of a flowing part – is marked.

1. Introduction

1.1. Hydraulic and Steam Turbines

Water wheels and windmills, which found use in the deep antiquity, are considered as the historical predecessors of modern turbines. In essence, the ancient windmill is the predecessor of modern wind turbines. In ancient times, the idea of using ‘wind force’ of steam for creation of rotary movement was born also. This idea was embodied in the numerous projects of steam and gas turbines during many centuries, but nobody managed to create a practical engine. Only in 1884 C.A. Parsons and in 1890 G.P. De Lavals constructed the first industrial samples of steam turbines.

Hydraulic turbines by then have reached significant perfection and their work was thoroughly investigated.

Further hydraulic and steam turbines were constantly improved due to the work of scientists, designers, metallurgists, and technologists; in particular, their power and their efficiency were raised significantly. Power of hydraulic turbines achieved 500 MW; steam turbines for electrical stations have reached a capacity of 1200 MW and more at efficiency of 40 per cent. Simultaneously, under construction are highly effective steam turbines of small and average capacity not only for power stations, but also for drive of various mechanisms for ships and other objects.

1.2. Gas turbines

Gas turbines have appeared in industrial markets later – only in 1925-1940. Its occurrence was connected first of all to the needs of aviation, and in this area, gas turbines completely superceded the internal combustion engine. Gas turbines, which work on the fuel combustion products, represent basic and constantly improved type of equipment for pumping of natural gas large distances. Besides, it is applied for power production on the power stations and on ships. In various combinations with steam turbines, the gas turbines allow to design power units having a total capacity up to 1000 MW with efficiency up to 60 per cent.

1.3. Terms 'turbine' and 'unit'

The term 'turbine' is frequently applied as a synonym of the terms 'unit' or 'engine' because any turbine works in structure unit or engine: the hydraulic turbine is used in a hydraulic electrical station to drive an electrical generator; the steam turbine is included into the structure of steam power unit which includes a set of equipment ensuring effective work of this unit; at last, the gas turbine works in a structure of gas turbine unit or gas turbine engine. Besides the individual steam and gas turbines, combined steam and gas units can work together in a single structure in power stations, ships etc.

Main attention will be further paid to thermal (steam and gas) turbines and to the appropriate thermal turbine units.

The major advantages of turbines and turbine units are their high economy, compactness, reliability and the possibility to use them as component parts of power units with unit capacity from hundreds of watt to thousands of megawatt.

2. The common information on work of turbines of various types

2.1. Principles of the turbine work

Turbine is a prime mover with rotary motion of the working unit, namely the rotor, and with the continuous operating process of converting the potential energy of the working fluid supplied (steam, gas, air or their mixtures, or liquids such as water) into mechanical work on the rotor shaft. Turbines are bladed machines because the energy conversion occurs in the blading consisting of guide (nozzle) vanes mounted in the stationary casing, and moving blades fixed on the rotor and moving with the rotor. This combination of vanes and blades creates blade stage (cascades) that is a system of channels where the operating process of the turbine takes place. Turbine blading design is one of the most important and complicated problem arising in the process of designing a machine. The complexity is connected with the necessity of a complex solution of problems of gas dynamics, heat transfer, structural strength and technology of manufacturing blades.

In the guide (nozzle) vane cascades the flow of steam, gas or air is accelerated and twisted. In the guide vane cascades (guiding apparatus) of a turbine with liquid working fluid (for example water) the required flow direction is provided and the fluid flow rate is controlled.

In the rotating cascade of moving blades stationed downstream of the guide (nozzle) vane cascade the energy of the moving steam, gas or liquid is converted into mechanical work on the rotating rotor for overcoming the resistance forces of the driven machinery. This combination of rows of guide (nozzle) vanes and moving blades (arranged in series) is a turbine stage. The working fluid energy conversion may take place in one stage (in this case the turbine is referred to as a one-stage turbine) or in several stages in succession, the latter being placed one after another (in this case the turbine is referred to as a multistage turbine).

2.2. Classification of turbines

Turbines are classified as axial and radial (radial-axial) flow depending on the direction of the flow relative to the rotor axis (see Figure 1).

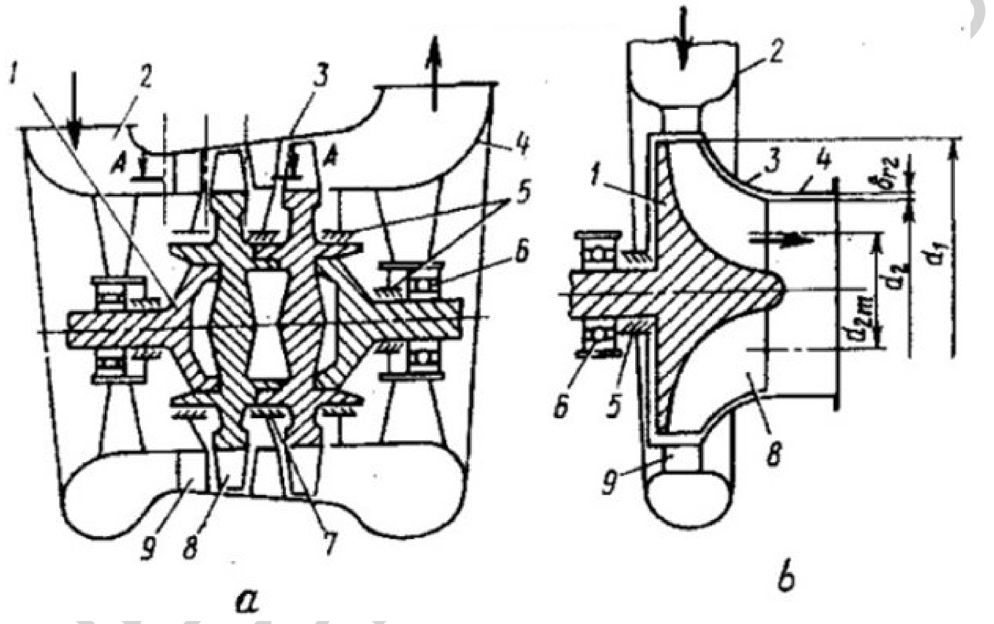

Figure 1. Schemes of the main types of turbines: *a* – axial turbine; *b* – radial-axial flow (centripetal) turbine. 1 – rotor; 2 – inlet; 3 – stator; 4 – outlet; 5,7 – labyrinth seals; 6 – bearing; 8 – moving (rotating) blade; 9 – nozzle (guide) blade (vane)

In axial turbines (Figure 1, a) the working fluid moves mainly along co-axial surfaces parallel to the turbine axis. The radial-flow turbine differs from the axial one in the fact that in the nozzle unit and in the greater part of the rotor the working fluid moves in the plane perpendicular to the axis of rotation. Depending on the flow direction to the circumference or to the axis of rotation, radial-flow turbines are divided into centrifugal (such turbines practically are not applied) and centripetal (Figure 1, b). Turbine rotors are set into rotation by the change in the moment of momentum of the working fluid as it flows through curved blade channels, which are formed by the moving blade surfaces. It is possible to choose the channels cross sectional area in the rows of guide (nozzle) vanes and in the rotor, which affects the stage operating process. On the basis of the operating principles, there are impulse and reaction turbine stages (and turbines). In impulse stages (turbines) the potential energy of the working fluid is converted into kinetic energy only in stationary guide (nozzle) vanes and this kinetic energy is used for creating useful work in the row of rotating blades. In reaction stages (turbines) a considerable part of the working fluid potential energy is converted into mechanical work in the rotor wheel blade channels. In steam and gas reaction turbines the circumferential force applied to the rotor is created not only due to the change of the working fluid flow direction (as in an impulse turbine) but also because of reaction force arising when the working fluid in the rotor blade channels expands.

2.3. Thermodynamic and kinematic parameters of the one-stage turbine

2.3.1. Change of parameters in a stage

In hydraulic reaction turbines the pressure of fluid flowing along gradually converging channels of the wheel decreases and its relative flow velocity increases. The pressure p_0 of the working fluid upstream of the nozzle vane cascade is higher than p_1 both in impulse and reaction stages (see Figure 2), therefore the flow in the nozzle accelerates: velocity $c_1 > c_0$.

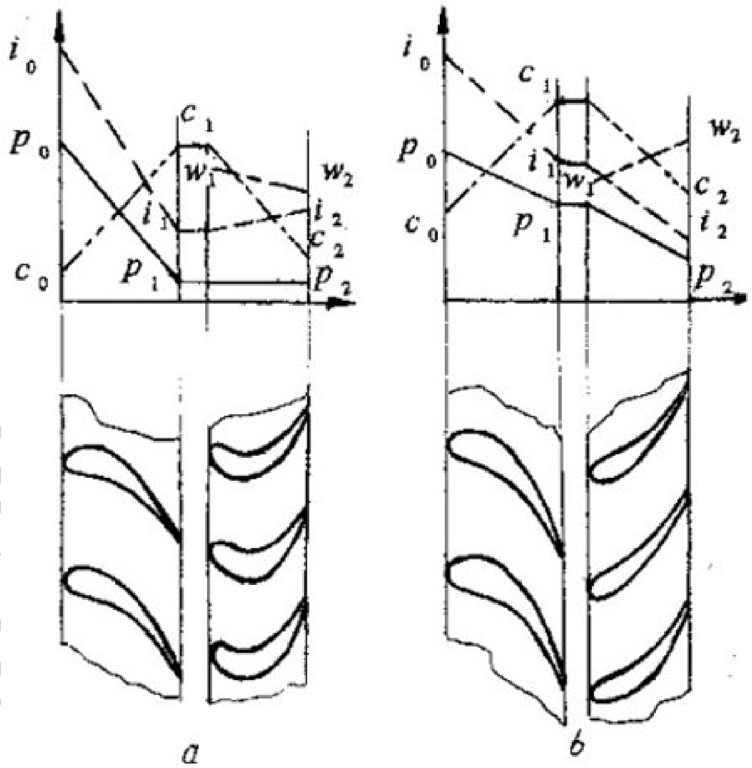

Figure 2. Changes of pressure p , absolute c and relative w speeds and enthalpy i in stages of the axial turbine: *a* – in impulse stage; *b* – in reaction stage

In an impulse stage, static pressure p_1 upstream of the rotor wheel is equal to p_2 downstream of it (Figure 2, a). In the reaction stage rotor wheel, $p_2 < p_1$. Absolute velocity decreases in the rotor wheel of the impulse and reaction stages. Figure 2 shows the variation of relative velocity w and enthalpy i in turbine stages.

2.3.2. Degree of reaction of a stage

The relationships between velocities and flow angles in turbine stages depend on the degree of reaction ρ of the stage, which is the ratio of the rotating blades theoretical temperature drop H_{0b}/c_p *to* the sum of the theoretical temperature drop of the nozzle vanes H_{0n}/c_p and rotating blades H_{0b}/c_p (i.e. to H_0/c_p) and is approximately equal to the temperature drop of the stage calculated using the stagnation parameters:

$$\rho = H_{0b}/\left(H_{0n}+H_{0b}\right) \approx H_{0b}/H_0 . \qquad (1)$$

The enthalpy (and temperature) drop in this relation may be determined from 'entropy – temperature' or 'entropy – enthalpy' diagrams as corresponding portions (see Figure 3); stagnation parameters are marked with asterisks. The word 'reaction' is used because the expansion of the working fluid occurs in the rotating blades row and additional force of reaction arises which rotates the rotor wheel.

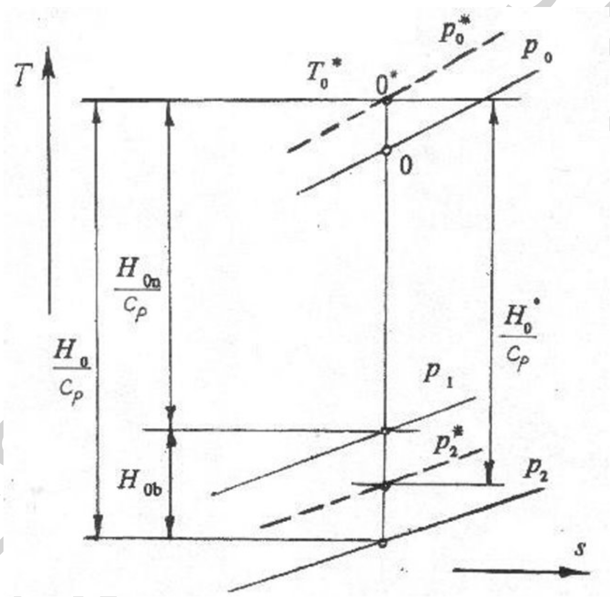

Figure 3. 'Entropy – enthalpy' diagram of process of a working fluid expansion in a turbine stage

The degree of reaction of stages at the mean diameter is usually chosen depending on the relative blade length l/D (D is the mean diameter of the flow passage), so that $\rho_r \geq 0.05...0.1$ at the blade root. In multistage turbines, the degree of reaction at the mean diameter gradually increases from the first to the last stage.

2.3.3. Specific work and efficiency of the turbine

The turbine is evaluated using two main parameters: specific work and efficiency.These values vary depending on the fact what turbine losses are taken into account. The working fluid specific work on the rotor wheel circumference may be determined from Euler equation:

$$\ell_u = c_{1u} u_1 + c_{2u} u_2, \tag{2}$$

where u_1, u_2, c_{1u} and c_{2u} are velocities at the turbine flow passage mean diameter, which are usually determined from velocity triangles at this diameter at the rotor wheel inlet and outlet (see Figure 4); u_1 is the circumferential velocity at the rotor inlet; u_2 is the circumferential velocity at the rotor outlet; c_{1u} is the circumferential projection of absolute velocity c_1 of the working fluid at the rotor inlet; c_{2u} is the circumferential projection of absolute velocity c_2 of the working fluid at the rotor outlet. [In the expression (2) both components are positive if opposite direction of pairs of angles α_1, β_1 and α_2, β_2 are assumed, see Figure 4].

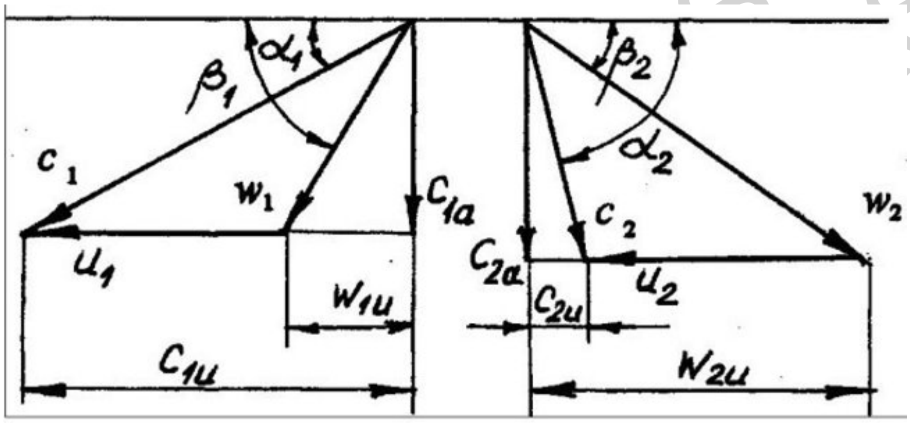

Figure 4. Triangles of speeds in the axial turbine

The other parameters of the velocity triangles are axial components c_{1a} and c_{2a} of absolute velocities c_1 and c_2, relative velocities of the working fluid w_1 and w_2 in the rotor wheel and their circumferential projection w_{1u} and w_{2u}. Specific work ℓ_u is less than the enthalpy drop H_0 by the amount of energy losses in the flow passage (it is the sum of specific losses h_n in the nozzle unit and h_b in the moving blades) and the working fluid kinetic energy h_e at the stage outlet. These losses may be evaluated using the efficiency at the wheel circumference

$$\eta_u = \ell_u / H_0 \approx 1 - \left(h_n + h_b + h_e\right) / H_0 \tag{3}$$

or

$$\eta_u = 1 - \xi_n - \xi_b - \xi_e, \tag{4}$$

where $\xi_n = h_n / H_0$, $\xi_b = h_b / H_0$, $\xi_e = h_e / H_0$ are corresponding relative losses. Usually there are radial clearances δ_n and δ_b between the blades and the casings in the nozzle cascade and the wheel, and the working fluid leakage occurs through their annular areas and thus the work ℓ_u at the wheel circumference decreases. If total specific losses in the radial clearances are h_c, the corresponding relative losses are $\xi_c = h_c / H_0$. Taking into account losses in the radial clearances the turbine specific work at the wheel circumference $\ell_{uc} = \ell_u - h_c$, power efficiency $\eta'_T = \eta_u - \xi_c$ and blade efficiency $\eta_b = \eta'_T + \xi_e$. If the kinetic energy of the flow issuing from the stage is used in the next stage, the losses may be estimated with the help of the stage efficiency from stagnation parameters, so that

$$\eta'_T = \ell_{uc} / H_0 = \eta'_T \left(H_0 / H_0^* \right), \tag{5}$$

where H_0^* is the total theoretical heat drop in accordance with stagnation parameters (see Figure 3).

To estimate a turbine work or shaft power besides the above losses it is necessary to determine relative losses ξ_{fv} caused by the friction of the disk and the working fluid and by the ventilation of the gas in the rotor wheel blade channels. Ventilation losses occur in partial admission turbines in which nozzle channels occupy only a part of the total circumference.

The degree of partial admission $\varepsilon = z_1 t_1 / (\pi D_1)$, where z_1 and t_1 are the number and the pitch of the nozzle vanes, D_1 is the mean diameter at the nozzle cascade outlet. The stage power efficiency, taking into account friction and ventilation losses, is $\eta_T = \eta'_T - \xi_{fv}$; the efficiency in accordance with stagnation parameters, taking into account these losses, is $\eta_T^* = \eta_T \left(H_0 / H_0^* \right)$, and the blade efficiency is $\eta_b = \eta_T + \xi_e$. Except for the considered losses at definition of power output of the turbine it is necessary to take into account losses of capacity due to friction in bearings both drive of pumps and regulators. These losses of energy are estimated in mechanical efficiency η_m.

Specific output of the one-stage turbine on the shaft in view of mechanical losses is

$$P_{Tsp} = H_0 \eta_T \eta_m . \tag{6}$$

Gross power of the one-stage turbine:

$$P_T = P_{Tsp} G, \tag{7}$$

where *G* is the flow per second of a working fluid through a flowing part of the turbine wheel.

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bases of designing of units and details of turbo-machines are stated].

Biographical Sketch

Eduard A. Manushin – Candidate of sciences (Ph.D.) (Technics) – 1964; Doctor of Sciences (Technics) – 1978; Professor – 1980; Full Member of the Russian Academy of Education (former – Academy of Pedagogical Sciences of the USSR) – 1989.

Graduated from Moscow Higher Technical School (now Moscow State Technical University – MSTU) named after N. Bauman (1956), specialty – engineer-mechanic in turbine building.

Engineer, lecturer, professor of MSTU (1956-1986). First vice-rector of MSTU (1986-1991). First vice-rector of the Russian Academy of Administration (1991-1994). General Director of the International System Research Center for Higher Education and Science (1995-1999). Academician-Secretary of the Higher Education Department of the Russian Academy of Education (1997 – 2002). Editor-in-Chief of the MAGISTER-PRESS Publishing House (1999 till now). Professor of MHTS (till now). Professor of MHTS (till now).

Research interests are concerned on energy-machine-building, specifically – in sphere of construction and computing of gas turbine and combined units, in particular – in cooling systems of high temperature gas-turbine engines.

Besides of this he is the specialist in the field of pedagogy of the higher school and in the sphere of informatization of education. From 1997 till now he is the Academician-Secretary of the Higher Education Department of the Russian Academy of Education.

He has 16 textbooks, more than 150 scientific articles. His recent publications include: *Theory and designing of gas turbine and combined units (*textbook, 2000, with co-authors); *Heat-transfer apparatus and cooling systems of gas turbine and combined units* (textbook, 2003, with co-authors); *Education and 21 st century: Information and communication technology* (with co-authors, 1999); *Thermotechnics* (with co-authors, 2003, forthcoming); *History of Humanity* (Russian editionof UNESCO Encyclopedia in 7 volumes – editor of 1-3 issued volumes).