# COMPRESSORS

**V. S. Beknev**
*Department of Power Engineering, Moscow Bauman State Technical University, Moscow, 107005, Russia*

**Keywords:** compressor, volume, piston, screw, gas-dynamic, axial, radial, rotor, stator, cascade, blade, vane, vaned diffuser, vaneless diffuser, inlet duct, outlet duct, stage, losses, head, circulation, efficiency, degree of reaction, modeling, controlling, rotating stall, surge, incidence angle, deviation angle, stagger angle.

## Contents

**Summary**

The basic principles of theory and design of the axial and radial compressors for the gas turbine engines and units are considered. The main schemes of the combined compressors for different applications are discussed. The basic equations of the fluid mechanics and thermodynamics for the compressor theory and design are considered. The axial flow compressor design is based on the cascade theory including some experimental data for profile and end wall losses. The radial flow compressor design is based on the 3D theory and some experimental data for the losses. The inlet and outlet ducts design is discussed too. The compressors performances, similarity problems, modeling and controlling are considered.

## 1. Compressor Definition, Types of Compressors

Compressors are the technical devices for compressing and transferring the compressed gas in the flow path of the engine or any unit. There are three types of compressors due to working process character: the volume compressors which compress and transfer the gas by the volume changing of the working part of the machine, the gas dynamic compressors which compress and transfer the gas by interacting it with the rotating and stationary parts of the machine as the rotating and stationary cascades of blades and vanes, and heat compressors which work on the external heat sources. The schematic arrangement of a one stage volume piston (reciprocating) compressor is shown in Figure 1, a, where 1 – the working volume, 2 – the suction volume, 3- the exhaust volume.

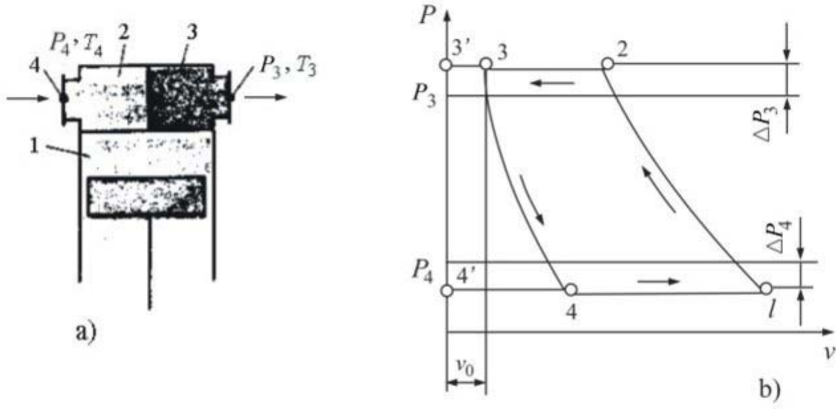


Figure 1. Scheme of one-stage volume piston compressor (a) and its ideal indicator diagram (b)

The corresponding valves are between these volumes.

The ideal indicator diagram of this compressor on the p-v plane is shown in Figure 1, b, (4-1) – the suction process, (1-2) - the compression, (2-3) – the transferring, (3-4) – the converse expansion.

There are several types of these multistage compressors: V – shaped, L – shaped and opposite. The schematic arrangement of the volume screw (rotary) compressor is shown in Figure 2.

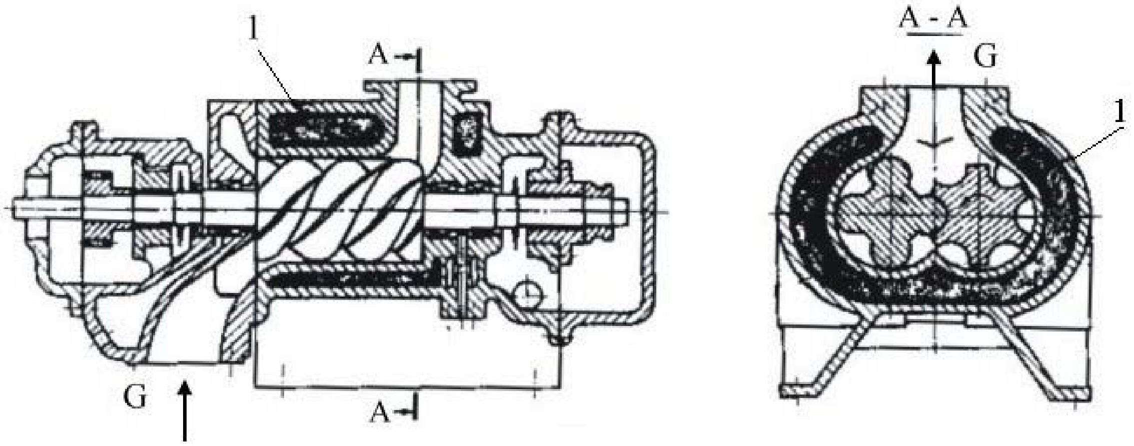


Figure 2. Scheme of volume screw compressor

It has two screw-shaped rotors – leading and led, which are placed in the special part of the casing.

These rotors for the oil-less screw compressor have the clearances between them and the casing during operation less than 0.1 mm, where 1 – the water cooling space. However the oilfilled screw compressors are more useful, they allow the contact for rotors and casing.

The volume compressors, including 2 or 3 lobe and sliding-vane ones, are widely used in chemical and civil industry. In energetics and in transport machinery of any kind, the gas dynamic compressors are used in most cases [1, 5].

There are two main types of the gas dynamic compressors: the axial flow compressors (AFC), which transfer the compressed gas practically along the axis of the rotor rotation,

Figure 3 (where 1 – rotor, 2 – casing, 3 – inlet duct, 4 – outlet duct, 5 – R, 6 – S, 7 – support bearing) and the radial flow compressors (RFC) or the centrifugal compressors which transfer the compressed gas practically normal to the compressor axis of its rotation, Figure 4 (where a, b, 1 – 4 the same as for Figure 3, 5 – VLD, 6 – VD, 7 – IGV, 8 - IND).

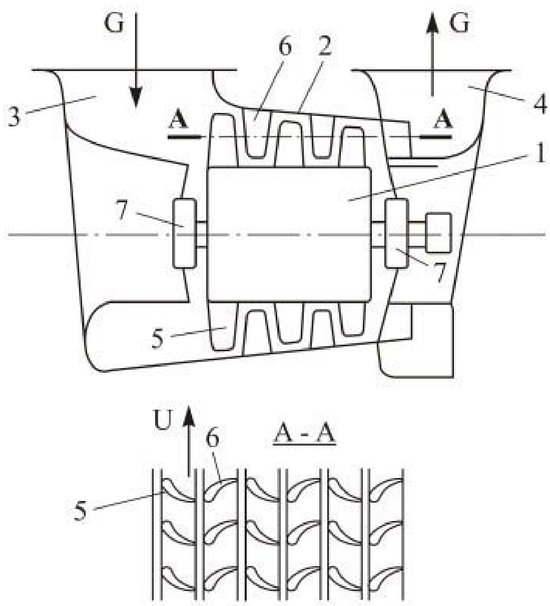


Figure 3. Scheme of MAFC

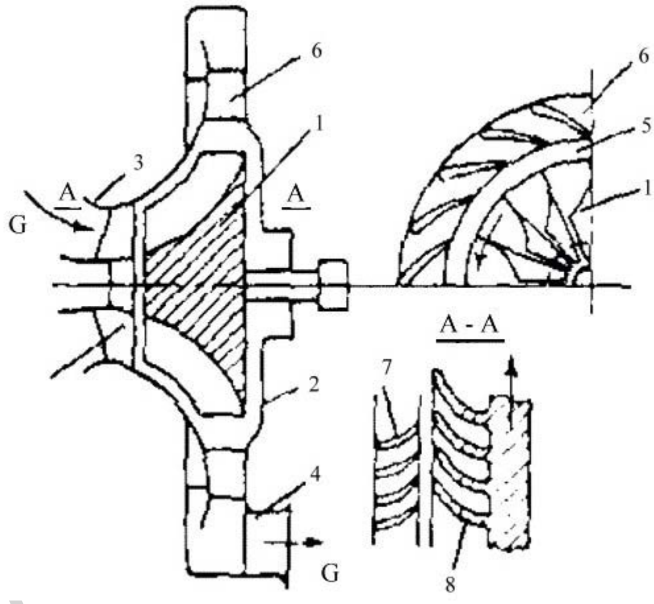


Figure 4. Scheme of RFC stage

Some intermediate design has the diagonal compressor which compresses and transfers the gas in the diagonal direction, Figure 5 (where 1 – 4 – the same as for Figure 4). For the modern AFC of the open cycle GTE or GTU the air mass flow rate is from several (kg)s$^{-1}$ to hundreds (kg)s$^{-1}$, the values of $\pi_C$ are up to 30 and higher for the MAC. The isentropic efficiency of the MAC reached the high level of 0.85 … 0.90 for the high mass flow rates and $\pi_C$. The RFC are used mostly for the transport GTE for the mass flow rates from several (kg)s$^{-1}$ to tens (kg)s$^{-1}$, the values of $\pi_C$ are up to 4…6 per one stage, the efficiency $\eta_{is}$ is 0.79 … 0.82 per stage. The differences between compressors for the aircraft and for other fields are vanishing.

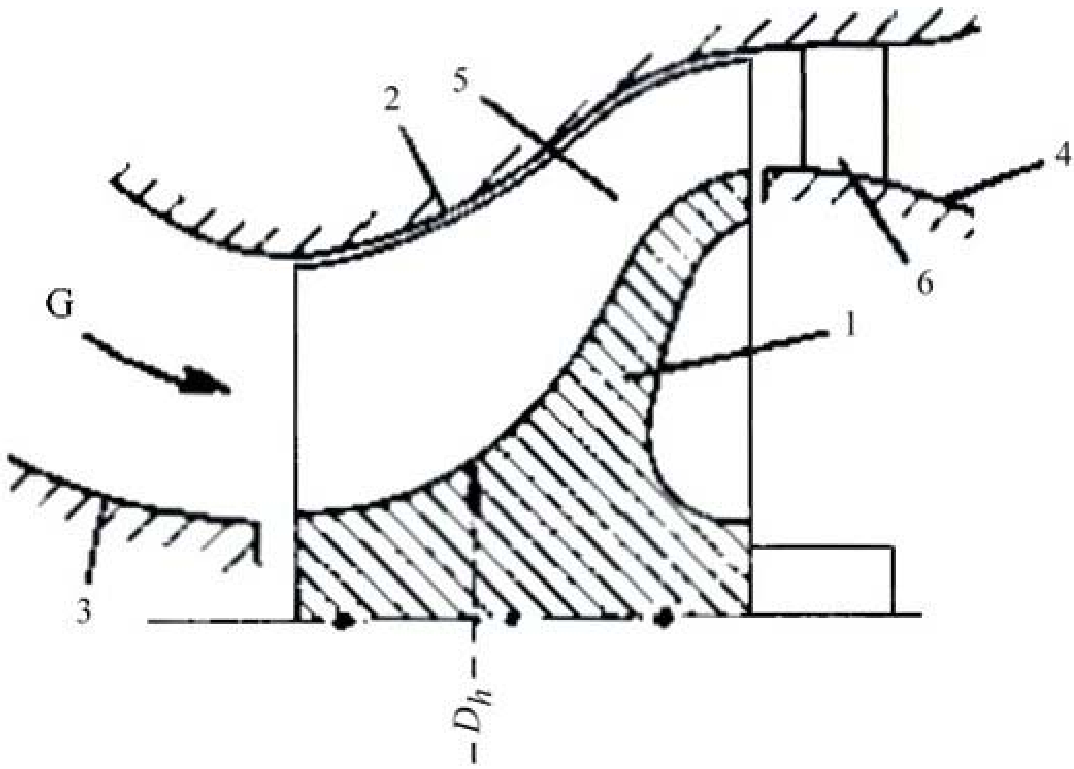


Figure 5. Scheme of diagonal flow compressor stage

### 1.2. **Axial flow compressor (AFC)**

The predecessor to the AFC is the axial flow fan invented in Russia by staff captain Teplov in 1854 and that was applied to metallurgy and used in the navy. The fan blades of the rectangular shape were fixed on the strap with some angle. The strap was put on the wheels, one of which was the leader.

During this wheel rotation the blades were moving in front of the rectangular channel inside of which the air was moved ventilating the room. These blades on the wheel create the fan for the room ventilating through the annulus channel.

The initial principle of action of the axial flow fan or compressor is connected with the inertia process acting on the gas mass particle *dm* which is forced to move along the moving blade surface liberating the room for the neighboring gas particle on the left side and starting the gas flow from right to left, Figure 6, a.

The plain diffuser cascade of the moving blades in Figure 6, a shows the velocity triangles in this flow, where $\overline{w}$ is the relative gas velocity through the moving channel, $\vec{u}$ is the speed of the blade itself, $\overline{c}$ is the absolute gas velocity, $\overline{c} = \overline{w} + \overline{u}$. The angle $\alpha$ corresponds to the absolute velocity $\overline{c}$, the angle $\beta$ - to the relative velocity $\overline{w}$.

Two diffuser cascades (RC and SC), Figure 7, produce the elementary stage of the AFC on the layer *dr* with their velocity triangles.

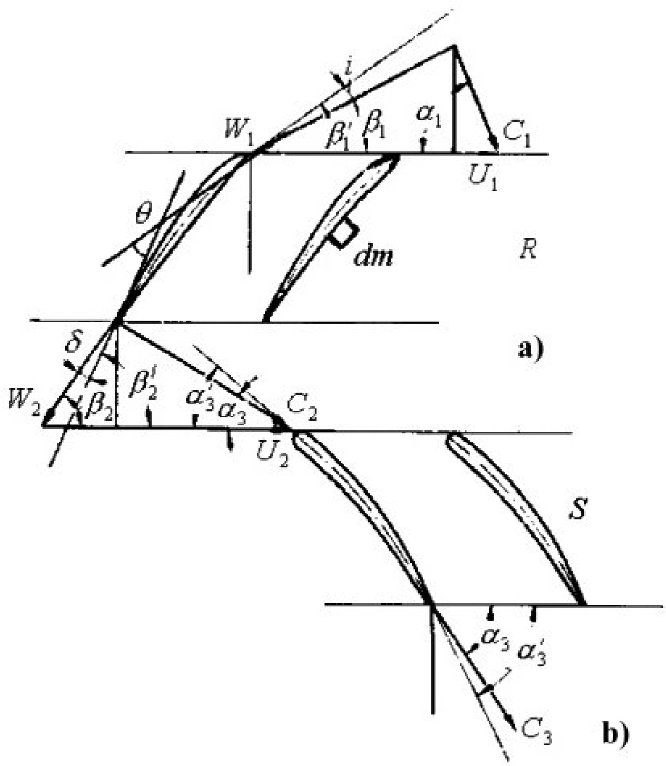


Figure 6. AFC rotor (R) and stator (S) cascades and velocity diagrams

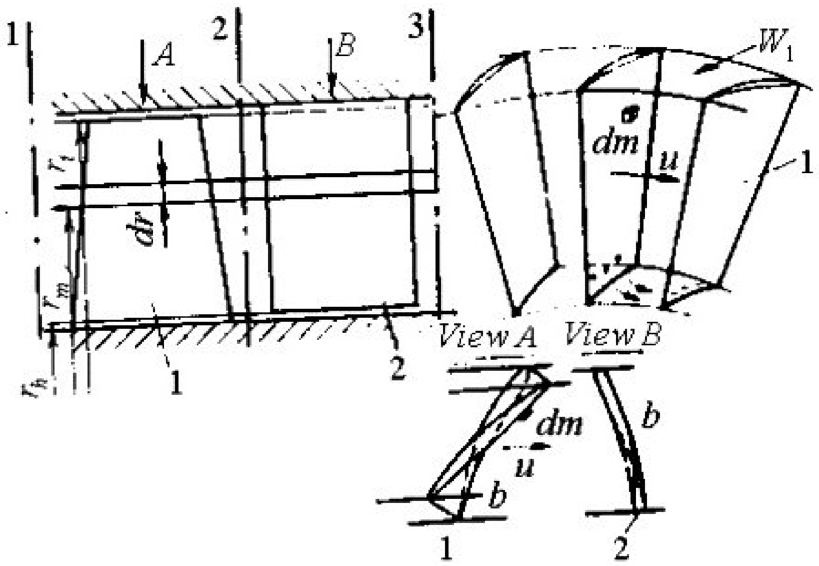


Figure 7. AFC stage rotor (R) and stator (S) blades

### 1.2.1. **AFC stage**

The stage of the AFC consists of the rotor (R) and the stator (S). In front of the first stage the IGV can be placed to create some special conditions for the first stage rotor design. Along the height of the stage in radial direction there are elementary stages, mentioned above. The stage efficiency depends on the degree of the concordance of the performances of its rotor and stator cascades.

### 1.1.2. Stage parameters

There is a difference between the elementary stage parameters and the stage parameters. For the elementary stage there are the local flow rate coefficient $\overline{c_a}^{/} = (c_a^{/})u^{-1}$, where $c_a^{/}(r)$ and $u(r)$, the local head coefficient $\overline{H_T^{/}} = \left(H_T^{/}\right)u^{-2}$, where $\overline{H_T^{/}}(r)$ and $u(r)$, the local isentropic efficiency $\eta_{is}^{/}(r) = \left[H_{is}^{/}(r)\right]\cdot\left[\overline{H_T^{/}(r)}\right]^{-1}$. For the stage there are the flow coefficient $\overline{c_a} = \left(c_a\right)_{av}\left(u_t\right)^{-1}$, where $\left(c_a\right)_{av}$ is the mass flow rate averaged value of the axial velocity, the head coefficient $\overline{H_T} = \left[(H_T)_{av}\right]\cdot\left[U_t\right]^{-2}$, where $\left(H_T\right)_{av}$ is the mass flow rate averaged head of the stage, the isentropic stage efficiency $\left(\eta_{is}\right) = \left[H_{is}\right]_{av}\cdot\left[H_T\right]_{av}^{-1}$, where $\left(H_{is}\right)_{av}$ is the mass flow rate averaged isentropic head of the stage. The degree of reaction ($R_d$) is the elementary stage parameter which shows the part of the compression work of the rotor to the full compression work of the stage. For the stage the $R_d$ on the mean radius is usually shown. The stage total pressure ratio $\pi_{C_{st}} = \left(p_3^*\right)\cdot\left(P_1^*\right)^{-1}$, where $p_3^*$ is the mass rate averaged total pressure behind the stage, $p_1^*$ - the same before the stage.

### 1.1.4. Multistage axial flow compressor (MAFC)

The MAFC is the succession of the axial flow stages together with the inlet and outlet ducts, Figure 3. These stages form the blading apparatus (BA) of the MAFC, which can be considered separately from the ducts. The $(\pi_C)_{BA}$ is the product of the $(\pi_C)_{st}$, the general head of the MAFC is the sum of the stage heads, $H_{T\Sigma} = \sum_{i=1}^{n} H_{T_i}$. Taking into account that for the stage "i" the value of

$$\left(H_{is}\right)_i = C_p T_{0_i}\cdot\left[\pi_{C_i}^{(k-1)k^{-1}} - 1\right],\; H_{T_i} = \left[\left(H_{is}\right)_i\right]\cdot\left(\eta_{is}\right)_i^{-1},$$

$$T_{0(i+1)}^* = T_{0i}^*\left\{1 + \left[\pi_{C_i}^{(k-1)k^{-1}} - 1\right]\cdot\left(\eta_{is}\right)_i^{-1}\right\},$$

so

$$\left(\eta_{is}\right)_{C\Sigma} = \left(H_{is\Sigma}\right)\cdot\left(H_{T\Sigma}\right)^{-1} = \left\{C_p T_0^*\left[\pi_{C\Sigma}^{(k-1)k^{-1}} - 1\right]\right\}\cdot\left[\sum_{i=1}^{n}\left(H_{is}\right)_i\left(\eta_{is}\right)_i^{-1}\right]^{-1} \quad (1)$$

The expression (1) shows the important thing that for the case when $(\eta_{is})_i = const$, the value of $(\eta_{is})_{C\Sigma} < (\eta_{is})_i$ because always $H_{is\Sigma} < \sum_{i=1}^{n} (H_{is})_i$ [see 2.2.3. Figure 19,b]. It should be underlined that $(\eta_{is})_{C\Sigma} \neq \prod_{i=1}^{n} (\eta_{is})_i$ in contrary to the case of transferring the given value of power through it.

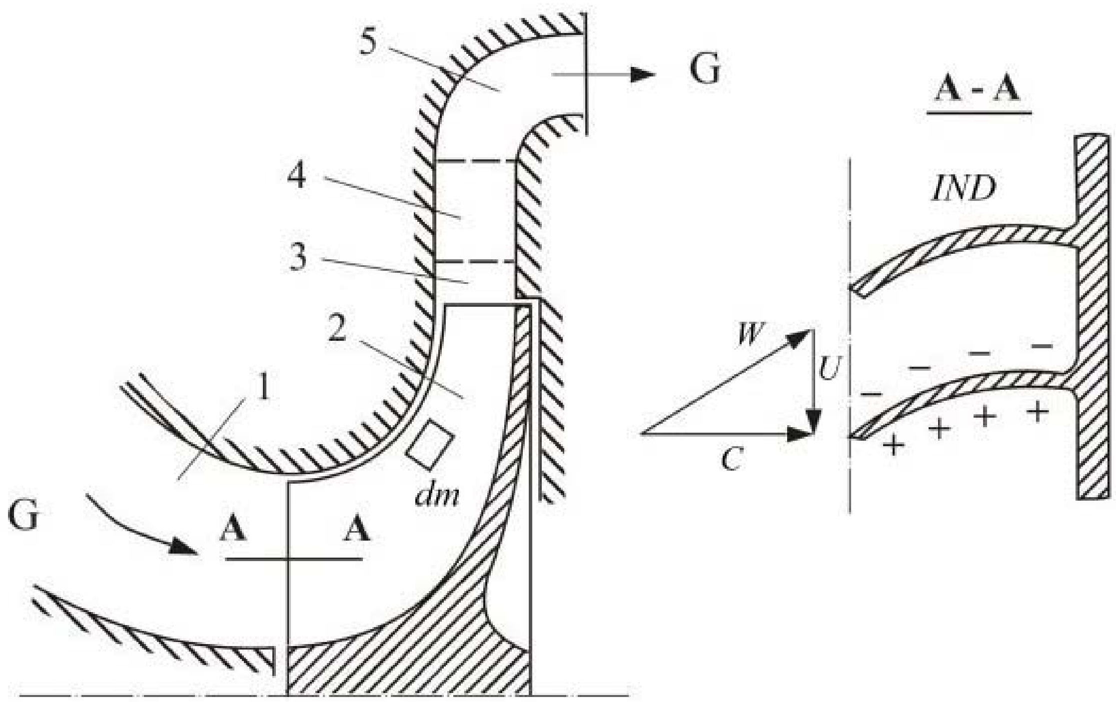


Figure 8. RFC stage

-
-
-



**Bibliography**

Beknev V.S. et al. (1983). Turbomachines and MHD-generators of the gas turbine and combined units, 392 pp. Moscow, Machinostrojenie, Russia. [This presents the basic equations for turbomachines design, the performances, modeling and controlling the gas dynamic compressors too]. [in Russian].

Beknev V.S. et al. (1973). Gas dynamics of the gas turbine and combined units, 392 pp. Moscow, Machinistrojenie, Russia. [This presents the basic equations of gas dynamics for the fluids in the one-dimensional, the 2D and 3D approaches. The method of singularities is considered too]. [in Russian].

Beknev V.S. et al. (1999). End wall phenomena at axial flow compressor design. Vestnik MGTU. Machinostrojenie, N3, p. 50-58. [This presents some design data for the radial distribution of losses at the stages of the multistage axial flow compressor. The deviation angle can be estimated by the method of the integral equations for any blade shape]. [in Russian].

Beknev V.S. (1967). Axis-symmetrical duct with inner ring vanes design. Teploenergetica, N4, p. 60-65. [This presents the approach for the diffuser duct design with several ring vanes to decrease the separation losses]. [in Russian].

Cohen H. et al (1987). Gas turbine theory, 3rd edition, 414 pp., Longman scientific and technical, England [This presents the basic materials for the gas turbine cycles, compressors, turbines, etc., and their performances].

Manushin E.A. et al. (1993). Nuclear gas turbine and combined units, 272 pp. Moscow. ISBN 5.283-03568-9. [in Russian]. [This presents the helium turbomachines design] .[in Russian].

Roberts W.B. et al (1988). Design point variation of 3-D Loss and Deviation for axial Compressor Middle Stages, ASME paper 88-GT-57, 16 pp. [This presents the experimental data for the stage losses in axial compressors].

**Biographical Sketch**

**Victor S. Beknev** graduated from the Moscow Bauman State Technical University (1951) and the Lomonosov Moscow State University (1954). He is the author of more than 160 publications in the field of fluid mechanics, turbomachinery and gas turbine engines of different applications. Dr. Beknev has delivered a number of papers at ASME meetings.