Theory of turbocompressors

A course of lectures. 3 semester.
Group 63238/10

Department Assistant

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Theory of turbocompressors. *Lecture 7.*

CONTENT

Turbocompresor's stage theory basics

- Two general problems of the gasodynamic stage theory – "head" coefficient and efficiency definition
- Limitation of the calculating methods abilities
- Research of the physical basics of working process as a necessary condition for high efficiency of the flow passage creation.

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General goal of gasodynamic design is to provide necessary pressure ratio when it is the highest efficiency of the compressor.

$$\pi^* = \frac{p^*_{out}}{p^*_{in}} = \left[1 + \frac{k-1}{k} h_T \frac{1}{RT^*_{in}}\right]^{\frac{k}{k-1}\eta^*_p}$$

Two problems:

1. Definition of the h_t from the outlet velocity triangle. Difficulty – finding the flow delay angle δ_2.
2. h_r definition. Difficulty – complicated flow picture.

$$\eta^* = \frac{h_T - h_r}{h_T}$$

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Theory of turbocompressors. *Lecture 7.*

CONTENT

Gas moving in simple form canals.

- **Blade system of turbocompressor as a system of profiles or as channels system. Main channel types, which are forming flow passage of turbocompressor. Simple form channels as a model for flow peculiarities consideration in complicated canals.**
- **Gas moving in channels when the velocities are small. Inlet (acceleration) and basic sections in the tube of the uniform cross section. Velocity profiles when the flow is laminar and turbulent. Pressure losses calculation ("head") ant the main part of the tube by friction coefficient. Experimental dependences of friction coefficient from the Reynolds's number and relative roughness. Laminar and turbulent flow character.**
- **Self-semularity by Reynolds's number, hydraulic smoothness. Generalization of friction losses calculation for channels of optional form, abilities and limitation of such generalization.**
- **Straight axial diffuser flow. Conditions and criteria of the flow separation**
- **Losses in separated flow region, unsteadiness. Separation losses characterization. Boundary layer control. Experimental dates on plane and conic diffusers. Angle of expansion, cross section form, inlet inequalities and turbulence.**

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Main canal types, which are forming the flow passage of turbocompressor. Simple form compressors as models of the processes in complicated canals consideration.

- **1. Axially symmetric vaneless channels:** inlet sections of different systems
- **2. Radial and ring blade systems:** such blade systems are considered as a system of blade profiles or as system of between blade channels.
- **3. Complicated three-dimensional canals:** which don't have the axially symmetry – inlet and outlet equipments of compressors.

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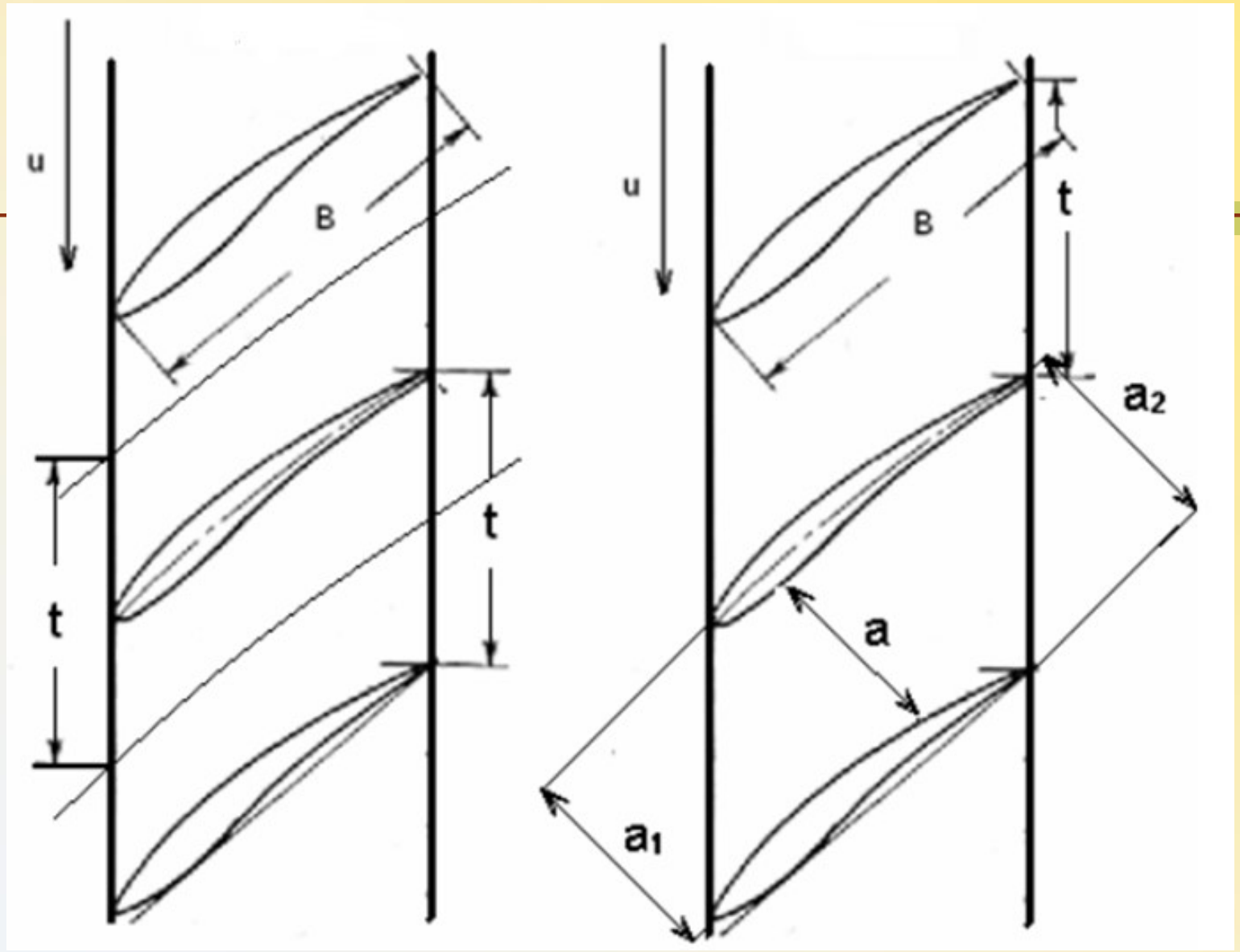

Blade system of axial compressor: left – as a system of between blades channels, right – as a system of profiles.

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Gas flow when the velocities are low. Inlet and main sections in the tube of the uniform cross section.

There are two types of sections in the channel where gas flows: **inlet (accelerate) and main section (section of stabilized flow).**

The flow could be of two types: **boundary layer and frictionless flow center.**

At the **boundary layer** , in accordance to **Newton's law** friction induced sheer stress is occurred, which is equal to the dynamic viscosity and velocity gradient by normal to the surface multiplication.

$$\tau = \mu \frac{\partial c}{\partial n} \quad (н / м^2)$$

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At the elementary channel section with the length *dl* the friction force is occurred:

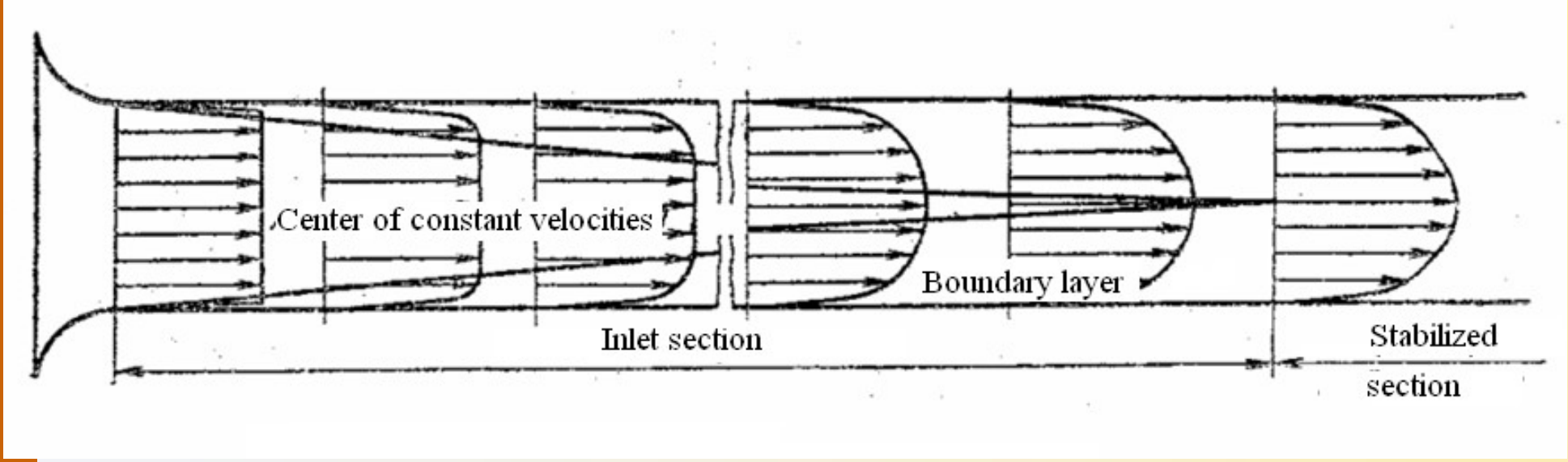
$$dR_w = \tau \times dl \times U$$

Where U – wetted perimeter of the channel, which is equal to πd *(m)* for cercular tube. Resistance force is overcome by means of mechanical gas energy.

At the cross section static pressure field stays uniform, as the flow doesn't deflect from the straight direction. That is why at the boundary layer total pressure is less then total pressure in the center of the flow.

$$p_{bl}^{*} = p + 0,5c^{2} \prec p + 0,5C^{2} = p_{c}^{*}$$

Gas flow when the velocities are low. Inlet and main sections in the tube of the uniform cross section.



Velocity change on the inlet section when the flow is in the straight tube

Reynolds's criteria for the developed flow analyze in the channels:

$$\mathrm{Re} = \frac{c d_h}{\nu} = \frac{c d_h}{\mu}\rho$$

where c – middle consumption velocity in the channel section, $d_h = 4f/U$ (f – section square, U – wetted perimeter of the channel) – hydraulic diameter, v – kinematic viscosity, μ – dynamic viscosity, ρ – density.

When the flow is **turbulent** the chaotic motion of the particles is observed, that is why the shearing strain is more than when the flow is laminar:

$$\tau = \mu \frac{dc}{dn} + \rho l^2 \left|\frac{dc}{dn}\right| \frac{dc}{dn}$$

Where l – length of runway

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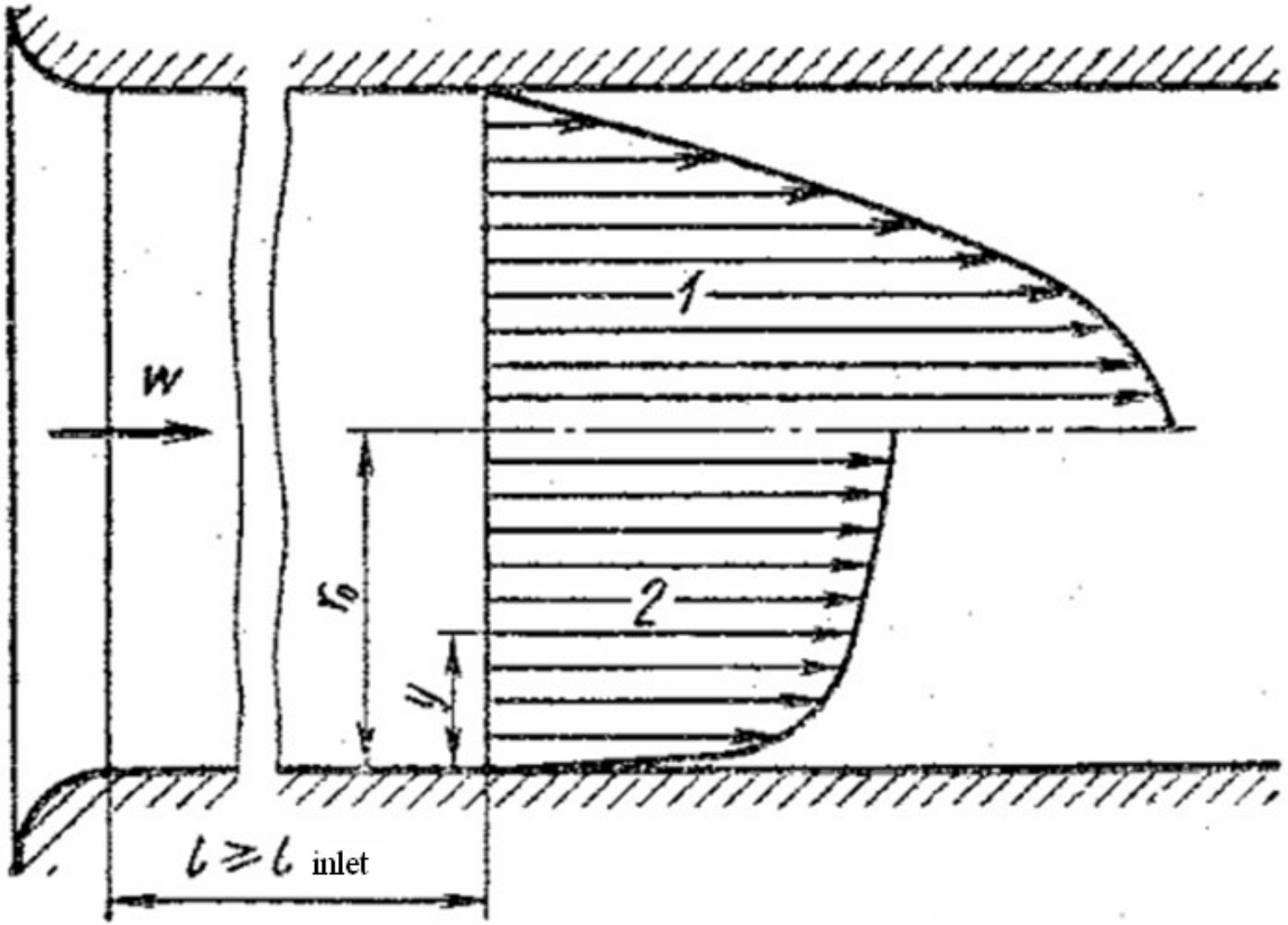

Velocity profiles on the basic section: when the flow is laminar (1) and turbulent (2) mode

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Gas flow when the velocities are low.
The calculation of the "head" losses on the main part of the tube by friction coefficient.

The highest loses are observed in the boundary layer, especially near walls. Shear strains on the wall channel surface :

$$\tau_0 = \lambda_1 \rho c_{mid}^{\ 2}/2$$

C_{mid} – middle consumption velocity,
λ_1 – dimensionless proportionality coefficient, friction coefficient, which depends on relative wall roughness and Reynolds number (sometimes is marked as c_f)

Elementary friction force, which is acting on the separated mass with volume *fdx*

$$dF_r = \tau_0 U_{wet} dx = \lambda_1 \rho (c_{mid}^2/2) U_{wet} dx ,$$

Where U_{wet} – wetted perimeter of the channel.
Here the friction losses are observed on all the walls of the channel, that is why U_{wet} is equal to perimeter of whole channel section.

Mass of separated volume $\rho f dx$, then the friction force, which comes on 1 kg per mass of gas:

$$F_r = \lambda_1 \rho \, (c_{\text{mid}}^2/2) \, \frac{U dx}{\rho f dx} = \lambda_1 \frac{U}{f} \frac{w_{\text{mid}}^2}{2}$$

In this formula the ratio $f/U=\rho_h$ is a hydraulic radius. For the circular section $\rho_h=r/2$, for rectangular cross sections with sides a and b $\rho_h=ab/(2(a+b))$.

The friction work ("head" loss) when gas is moving from section 1 to section 2, referred to 1 kg of mass:

$$h_r = \int_1^2 F_r\,dx = \int_1^2 \lambda_1 \frac{1}{\rho_h} \frac{c^2_{mid}}{2} dx$$

Or:

$$h_r = \int_1^2 \lambda \frac{1}{d_h} \frac{c^2_{mid}}{2} dx$$

Where resistance coefficient $\lambda=4\lambda_1$, and hydraulic diameter $d_h=4\rho h=4f/U$. For curved canal the integration is doing along curved middle line of channel.

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For circular cross section channel hydraulic diameter is equal to its diameter. With this consideration the pressure losses and losses coefficient of the main channel section:

$$\Delta p = \lambda \rho \frac{c_{mid}^2}{2} \frac{l}{d_h} \qquad \zeta_{fr} = \frac{\Delta p}{\rho \frac{c_{mid}^2}{2}} = \lambda \frac{l}{d_h}$$

So, losses coefficient of the tube is proportional to dimensionless friction coefficient, tube length and inversely proportional to hydraulic diameter. Pressure losses is proportional to losses coefficient and dynamic pressure $\rho c^2/2$

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Gas flow when the velocities are low.

Experimental dependence of the friction coefficient from the Reynolds number and relative roughness.

For turbulent flow in a tube with hydraulically smooth walls the resistance coefficient is:

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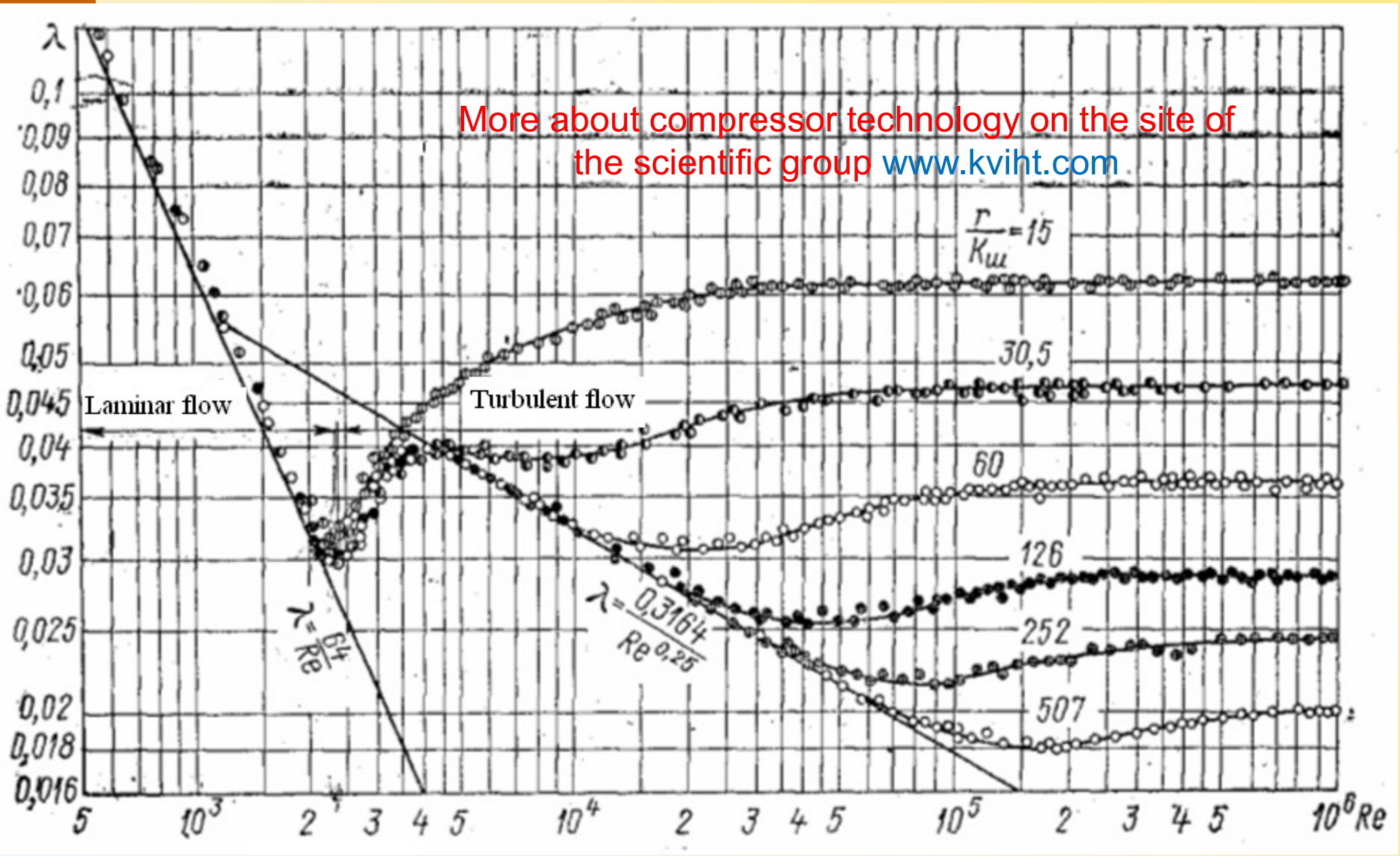
$$\lambda = 1/(1{,}8 \lg \mathrm{Re} - 1{,}5)^2$$

The sphere of flow modes, at which the resistance coefficient doesn't depend on Reynolds number is called the **sphere of self-semilarity by Reynolds number.**

Resistance coefficient λ of rough tubes at the sphere of self-semilarity for big enough Reynolds numbers could be determined by Nikuradze's curves or by formula:

$$\frac{1}{\sqrt{\lambda}} = 2{,}0 \lg(r/K_{\underset{\text{rough}}{\text{mid}}}) + 1{,}74$$



Resistance coefficient λ dependence for circular tubes from the Reynolds number and relative roughness by Nikuradze's dates

Gas flow when the velocities are low.
Experimental dependence of the friction coefficient from the Reynolds number and relative roughness.

Impact assessment of roughness on the flow character is assumed to make dactylic with the help of dimensionless value of roughness:

$$R_0 = cK_{\substack{\text{mid}\\ \text{rough}}} / \nu,$$

where $K_{\text{mid rough}}$ – middle value of roughness.

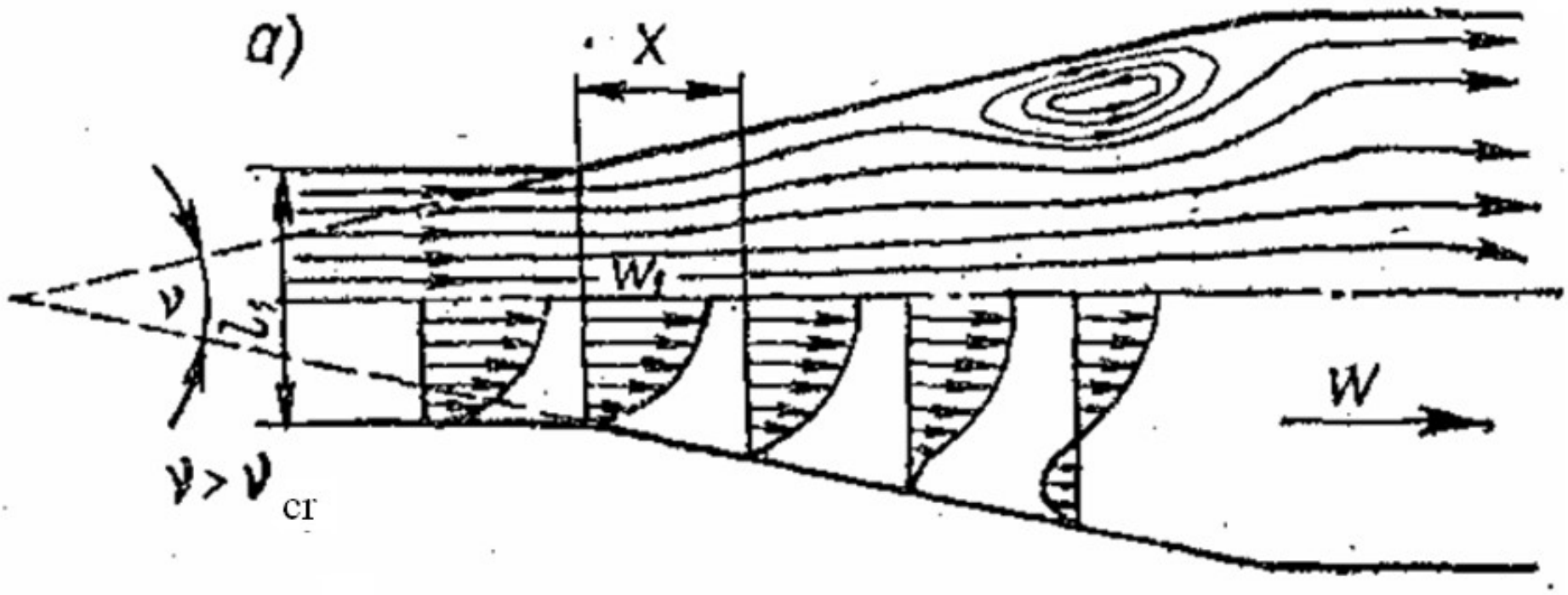
Roughness doesn't influence on resistance, as experience shows, if $R_0<100$. In this case they say about hydraulically smooth surface. Value $R_0=100$ could be used for row estimation of admissible roughness.

For adjusted calculation of the thickness of the laminar sublayer it is needed to determine by the length of studied surface or to use the formula:

$$K_{\substack{\text{mid}\\ \text{rough}}} / d_{\Gamma} \le (181 \lg \mathrm{Re} - 16{,}4)/\mathrm{Re}$$

Lecture 7.

Axial diffusers flow.
Conditions and criterions of the flow separation.



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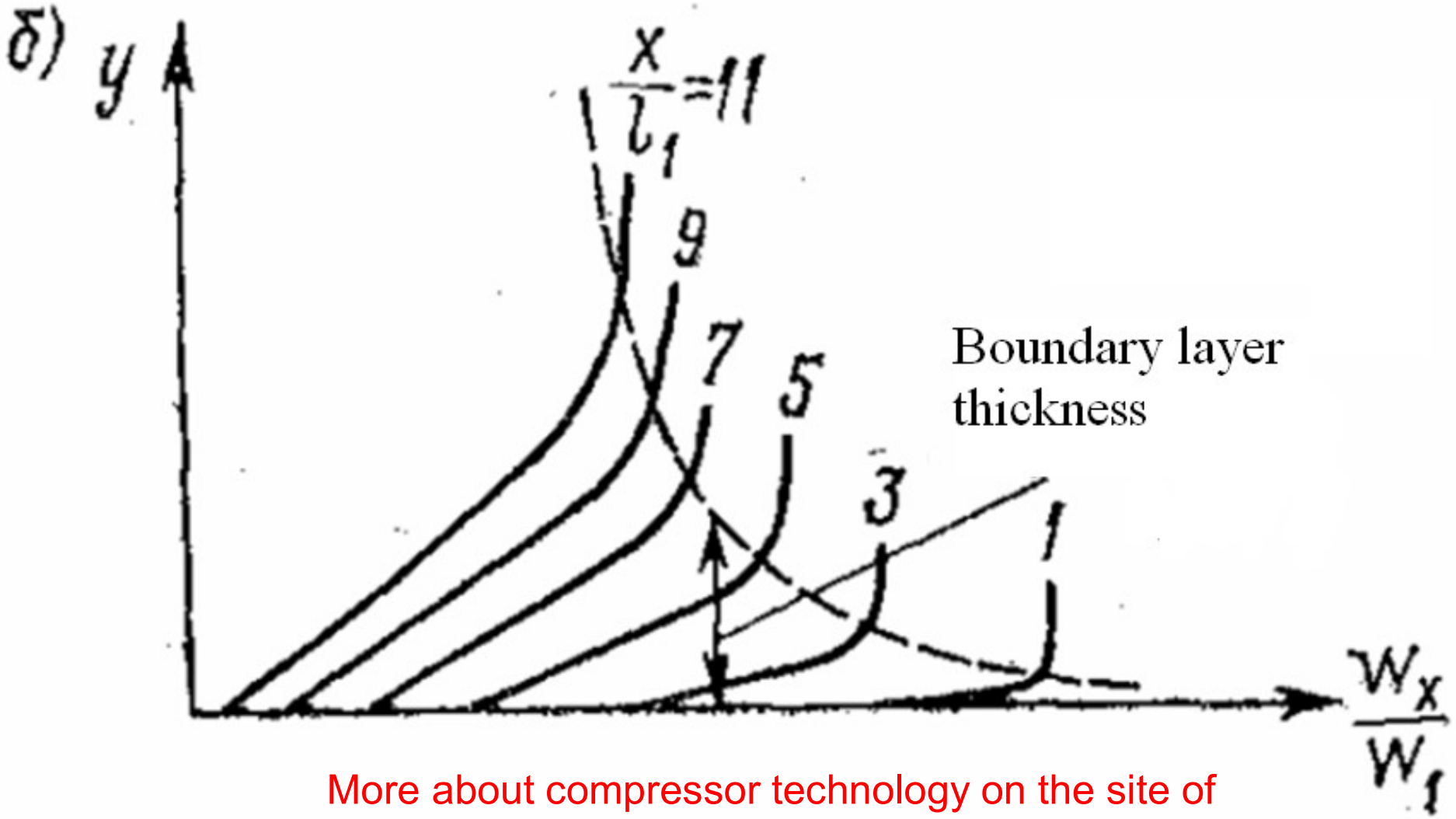

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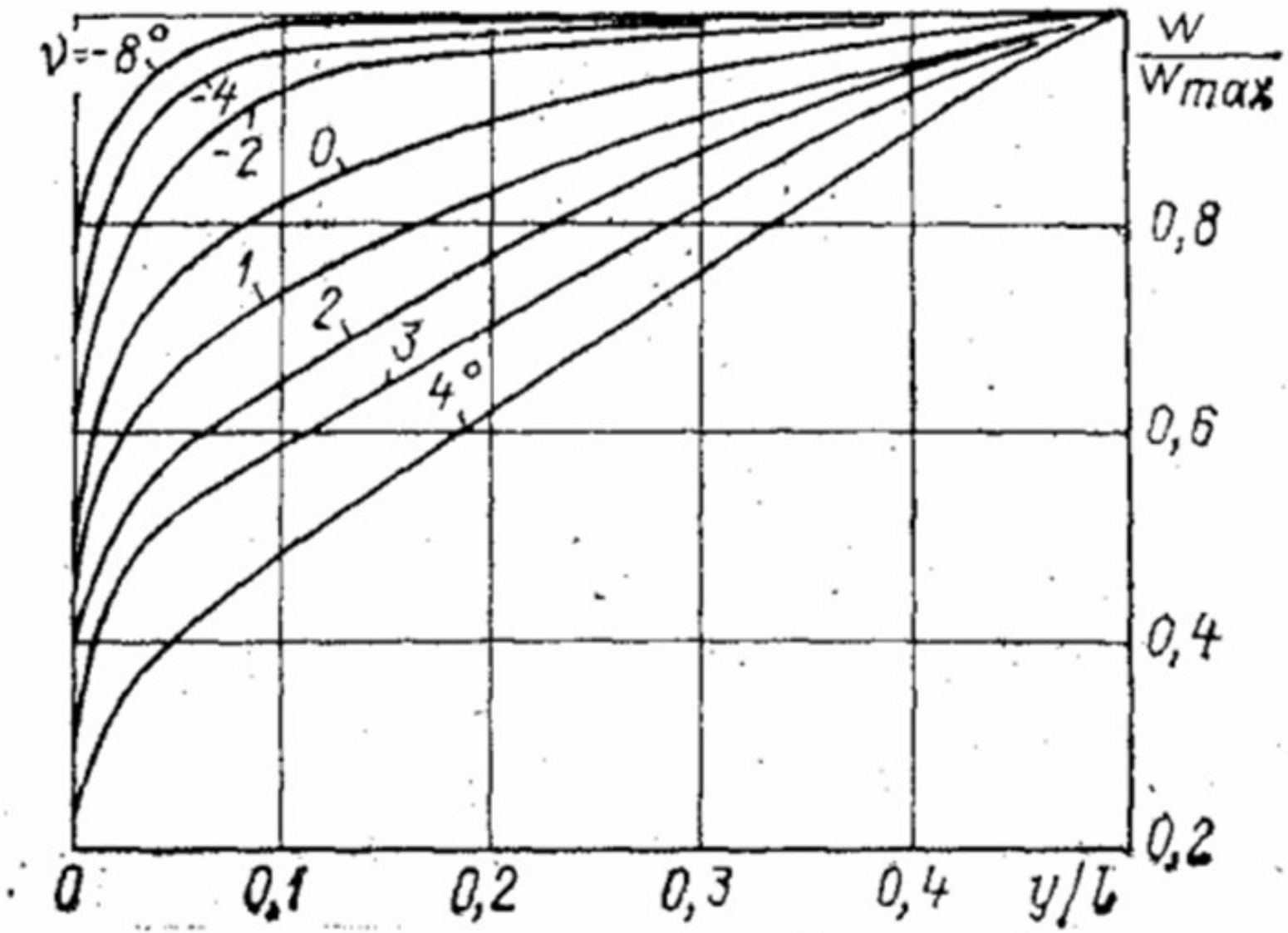
Velocity development at the boundary layer in different diffuser layers by length: w1- velocity of basic flow (center) on the inlet section of diffuser, y – distance from the wall of the channel.



Velocities profile when the angle of expansion of the channel is different: v>0 – diffuser; v<0 - confuser

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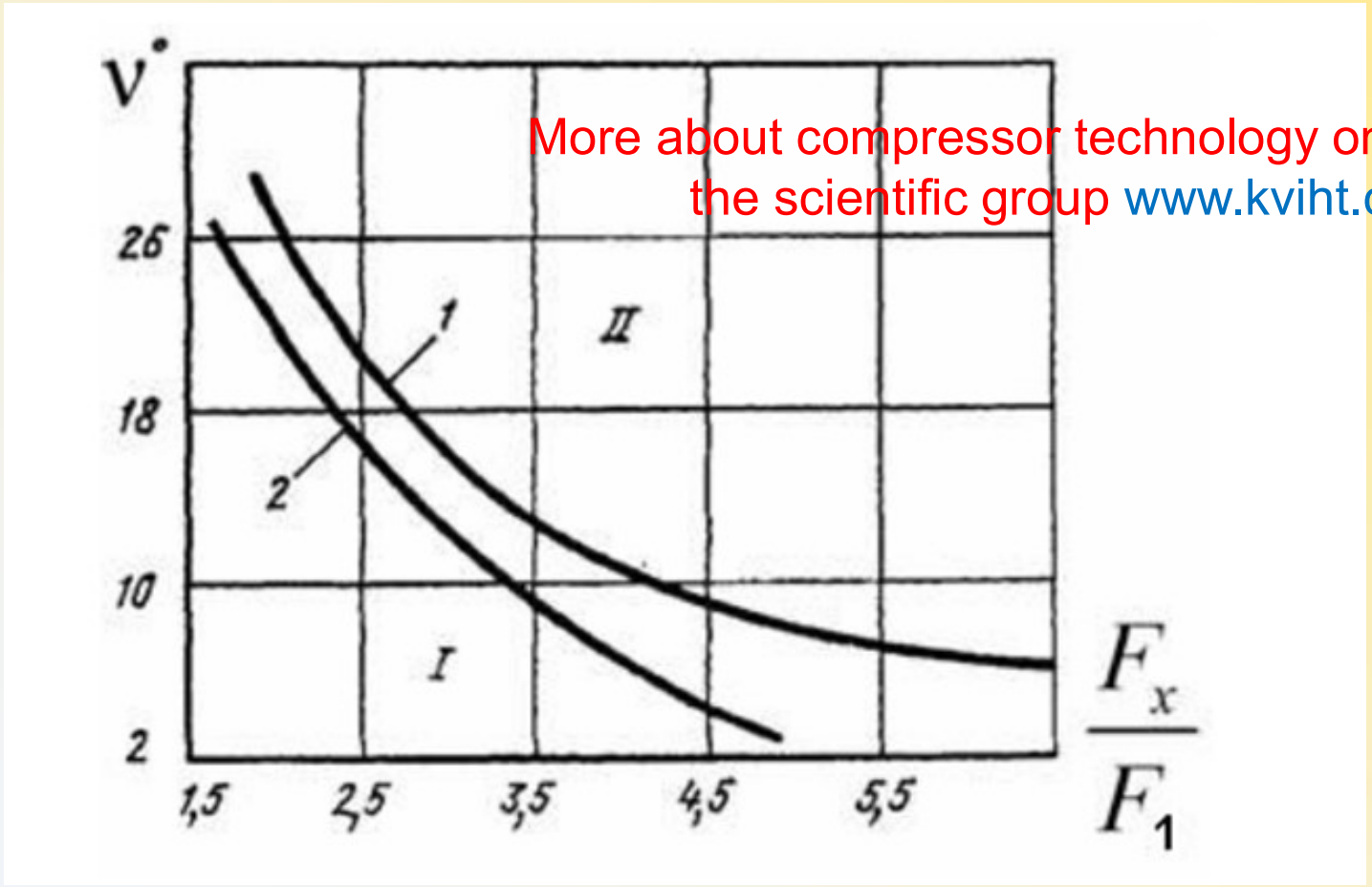
The beginning of the flow separation derivation corresponds to the flow stop near the wall, so the $dw/dy = 0$ when $y \to 0$, shear strains on the wall at the separation point $\tau_0 \sim dw/dy = 0$.

Separation leads to the vortex derivation, in which the direction of kinetic energy of the flow gets the vortex form. This energy as a mechanical energy is loosed – **vortex losses** appear or **separation losses**.



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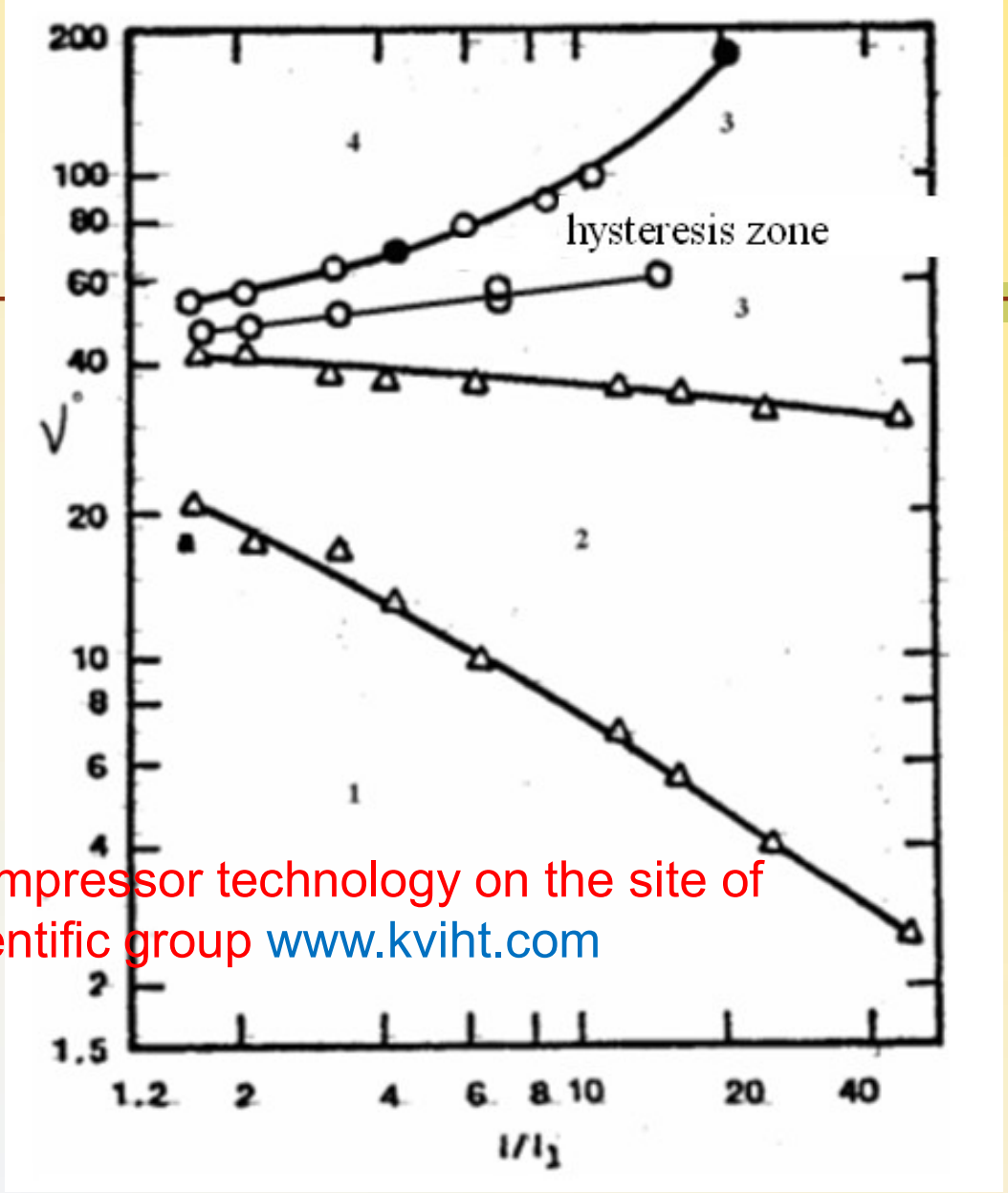
Flow separation sphere in diffusers

$$1 - \delta_1^* / D_1 \approx 0; 2 - \delta_1^* / D_1 \gg 0$$

Curves are dividing all the sphere of definitions v and F_x/F_1 on twp parts: diffusers without separation (sphere 1) and with separations (sphere 2). F_x – diffuser square by x coordinate. Curve 1 determines to more favorable inlet conditions (thickness of boundary layer displacement on the inlet section $\delta_1^* \approx 0$). Curve 2 corresponds to the case when the diffuser equipment situates behind the long inlet section, when $\delta_1^*/D_1 \gg 0$.



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Characteristic flow modes in plane diffusers: 1 – no flow separation; 2 – unsteady separation on one of the walls; 3 – flow separation on one of the walls; 4 – separation on two wall

For approximate losses in diffuser determination with high expansion angle ν, all the losses are provisionally divided on **friction losses** and **expansion losses**, by which in general means losses that are connected with vortex formation when the separation exists. The expansion losses are determined as experimentally calculated summary losses and friction losses difference.

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Expansion losses determine with the help of formula:

$$h_{exp} = \zeta_{exp} \quad c_1^2/2$$

Where the resistance coefficient to the expansion ζ_{exp} determines by formula:

$$\zeta_{exp} = K_1 \varphi_{exp} \quad (1 - c_2/c_1)^2$$

K_1 – coefficient, which take into account the velocity field state and boundary layer in the inlet

Coefficient ϕ_{exp} – the fullness of a blow coefficient, which depends on expansion angle and the form of the channel. For the angles v = 0 … 40°

$$\varphi_{exp} \approx 3,2K_2\left[\mathrm{tg}(v_e/2)\right]^{1,25}$$

Where K_2 – coefficient, which characterize the form of the cross section of diffuser, v_e – equivalent angle of expansion of the channel.

Reynolds number is also influence o the flow separation development. It is obvious that when the flow is laminar, because of the velocity profile is less filled, the boundary layer is thicker, the flow velocity near the wall will fall down to zero earlier then in the case of turbulent flow, that is why **the flow separation when the flow is laminar will start faster.**

The channels with difficult cross section forms are estimated with the help of **equivalent conic diffuser,** which is straight conic diffuser with inlet and outlet squares and middle line length l, equal to corresponding sizes of considerating diffuser.

Equivalent angle of expansion v_e for such a conic diffuser could be determined as:

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$$\mathrm{tg}(v_e/2) = \frac{\sqrt{f_2} - \sqrt{f_1}}{l\sqrt{\pi}}$$

For small angles of expansion the Fligner's formula is used:

$$\varphi_{exp} = K_2 \sin v_e$$

Friction losses coefficient ζ_{fr} could be found:

$$h_{fr} = \zeta_{fr} \frac{c_1^2}{2} = \int_0^s \lambda \frac{1}{d_h} \frac{c^2}{2} ds = \int_0^x \frac{\lambda}{4} \frac{c^2}{2} \frac{U}{f} \frac{dx}{\cos(\nu_e/2)}$$

where s – wall length of the diffuser $dx = ds \cos(\nu_e/2)$

Lecture 7.

Axial diffusers flow.

Diffuser losses estimation. Influence of expansion angle, cross section form, flow unsteadiness, equivalent and optimal expansion angle.

From the continuity condition when: $\rho = \text{const}$, $c = c_1 f_1 / f$,

$df = d(\pi D^2/4) = 0{,}5\pi D\, d(D)$ и $d(D/2) = dx\, \mathrm{tg}(\nu_e/2)$, then:

$$\zeta_{fr} = \int_1^2 \frac{\lambda}{4\sin(\nu_e/2)} (f_1^2 / f^3) df = \frac{\lambda}{8\sin(\nu_e/2)} (1 - \dot{c}^2)$$

Then for the diffuser:

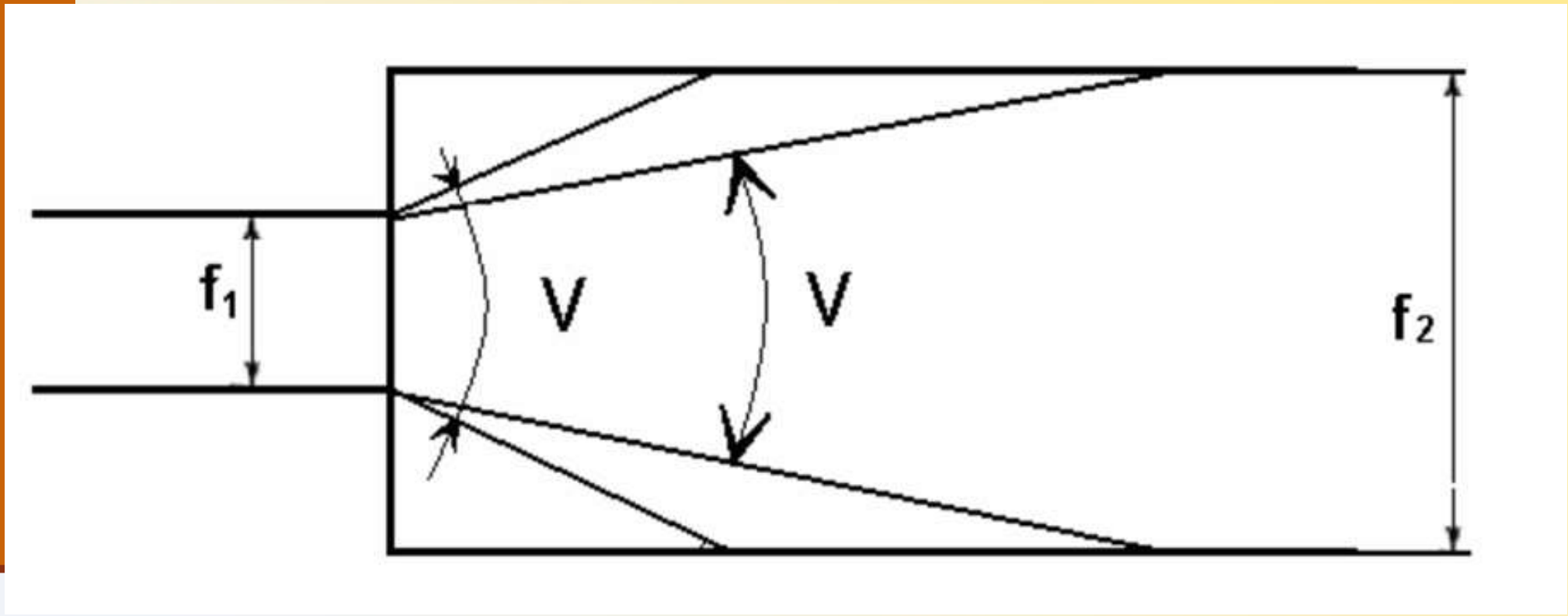
$$h_r = \zeta\, c_1^2/2 = (\zeta_{fr} + \zeta_{exp}\ ) c_1^2/2$$

where:

$$\zeta = \zeta_{fr} + \zeta_{exp} = \frac{\lambda}{8\sin(\nu_e/2)}(1-\dot{c}^2) + 3{,}2K_1K_2\left[\mathrm{tg}(\nu_e/2)\right]^{1{,}25}(1-\dot{c})^2$$

From the stated formulas it follows that minimum losses in diffuser correspond to approximately the highest expansion angle, when the boundary layer separation hasn't appeared yet.

Angle of expansion by which the losses coefficient is minimum is the optimal angle of expansion ν_{exp}.



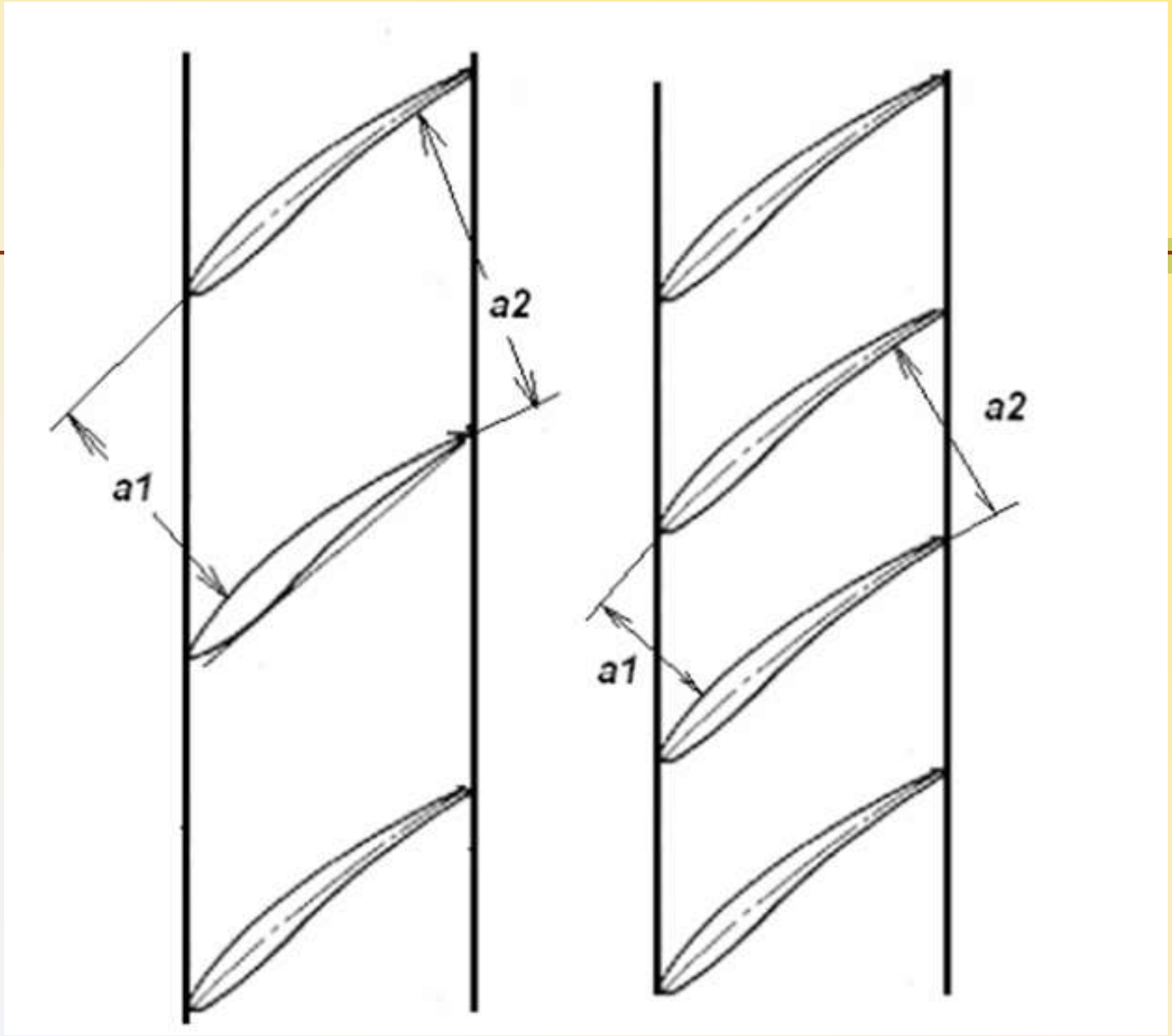
Angle of expansion influence on the diffuser's length when the cross sections are adjusted in the inlet and outlet

Optimal angle of expansion:

$$\nu_{opt} = 0{,}43\ [(\lambda/K_1)(n+1)/(n-1)]^{4/9}$$

Where $n=f_2/f_1$ – ratio of ending square of the cross sections of diffuser to the beginning, while using the assumption ρ=const we determine $n=1/c$. While ν_{opt} is low and $\sin(\nu_{opt}/2)\approx\nu_{opt}/2$, $\cos(\nu_{opt}/2)\approx 1$.

For plane diffuser (expansion takes place only in one plane) $\nu_{opt}\approx 10\ldots12°$.



Influence of relative step of diffuser blade system scheme on the equivalent angle of expansion

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Lecture 7.

Axial diffusers flow.

Diffuser losses estimation. Influence of expansion angle, cross section form, flow unsteadiness, equivalent and optimal expansion angle.

$$\operatorname{tg}\frac{\nu_{eqiv}}{2} = \frac{a_2 - a_1}{2B} = \frac{t}{B}\frac{\sin\beta_{bl2} - \sin\beta_{bl1}}{2}$$

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The aim of the boundary layer control– displace the point of flow separation downstream and to lower the losses in the diffuser while its axial dimensions are restricted

Methods: separative walls setting, guided vanes, the usage of the curved profiled walls of special form, suction and blowing of the boundary layer and others.

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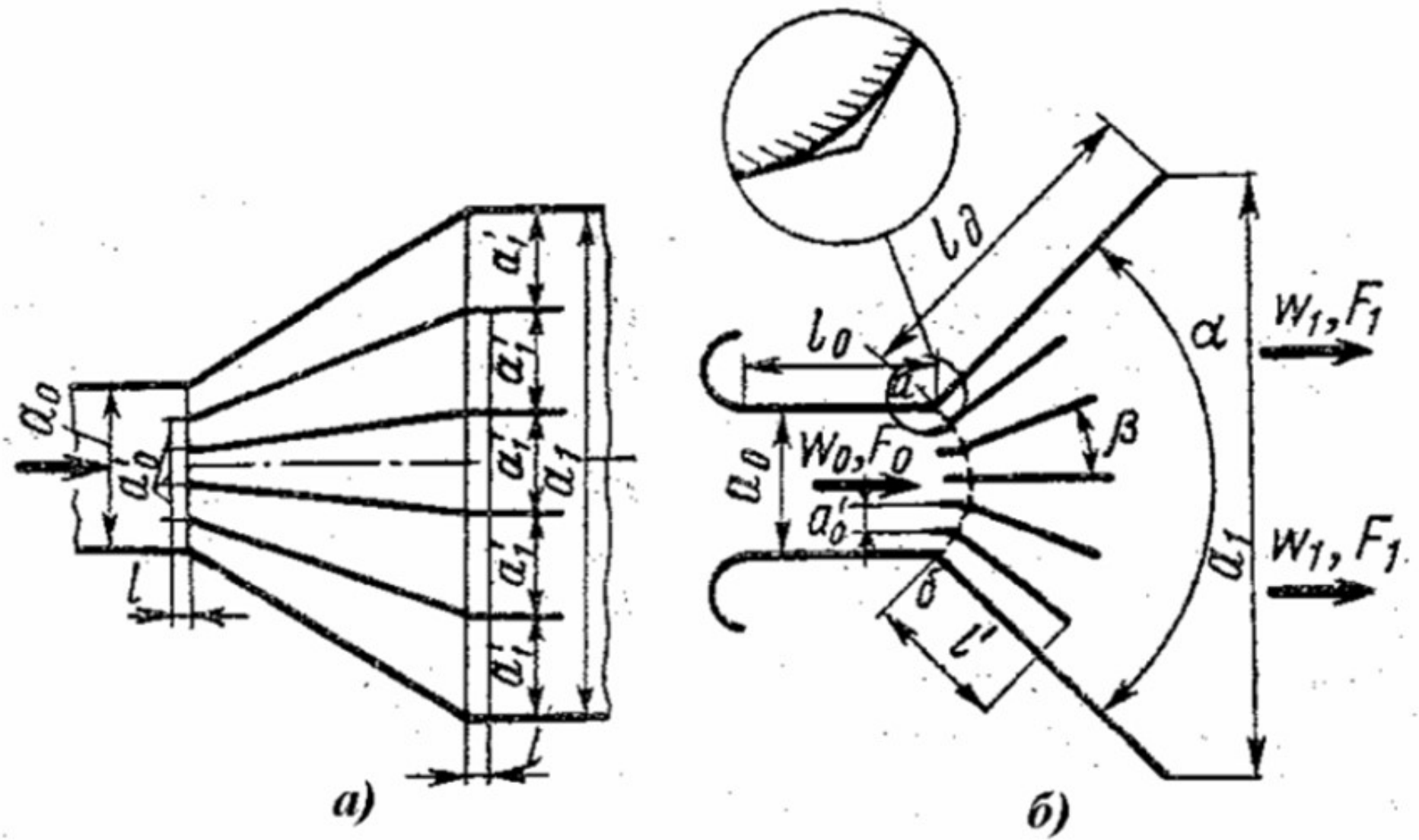

Losses decrease by separative walls setting: elongate (a) and shorten (б)

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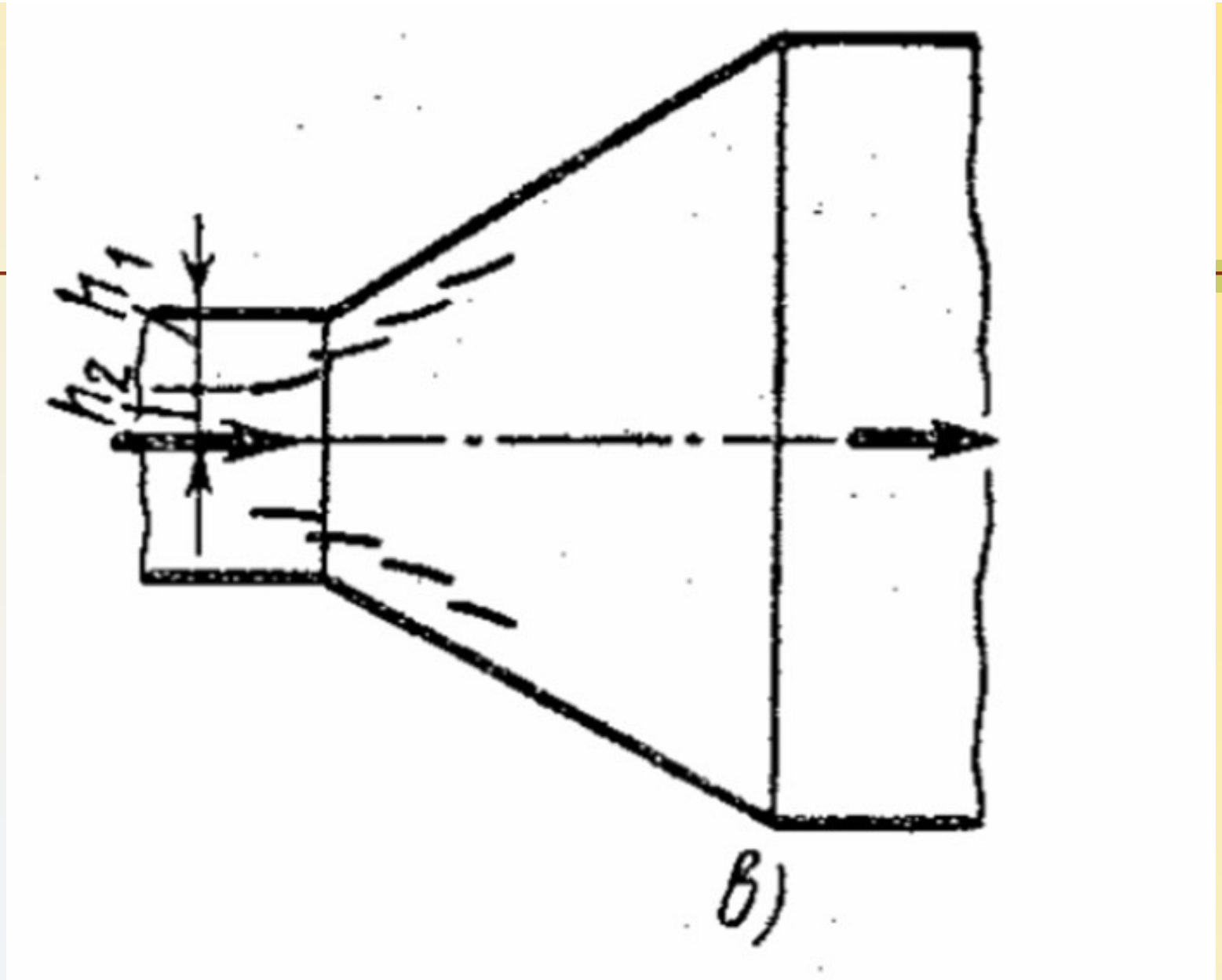

Losses decrease by guided vanes setting (deflectors) (в)

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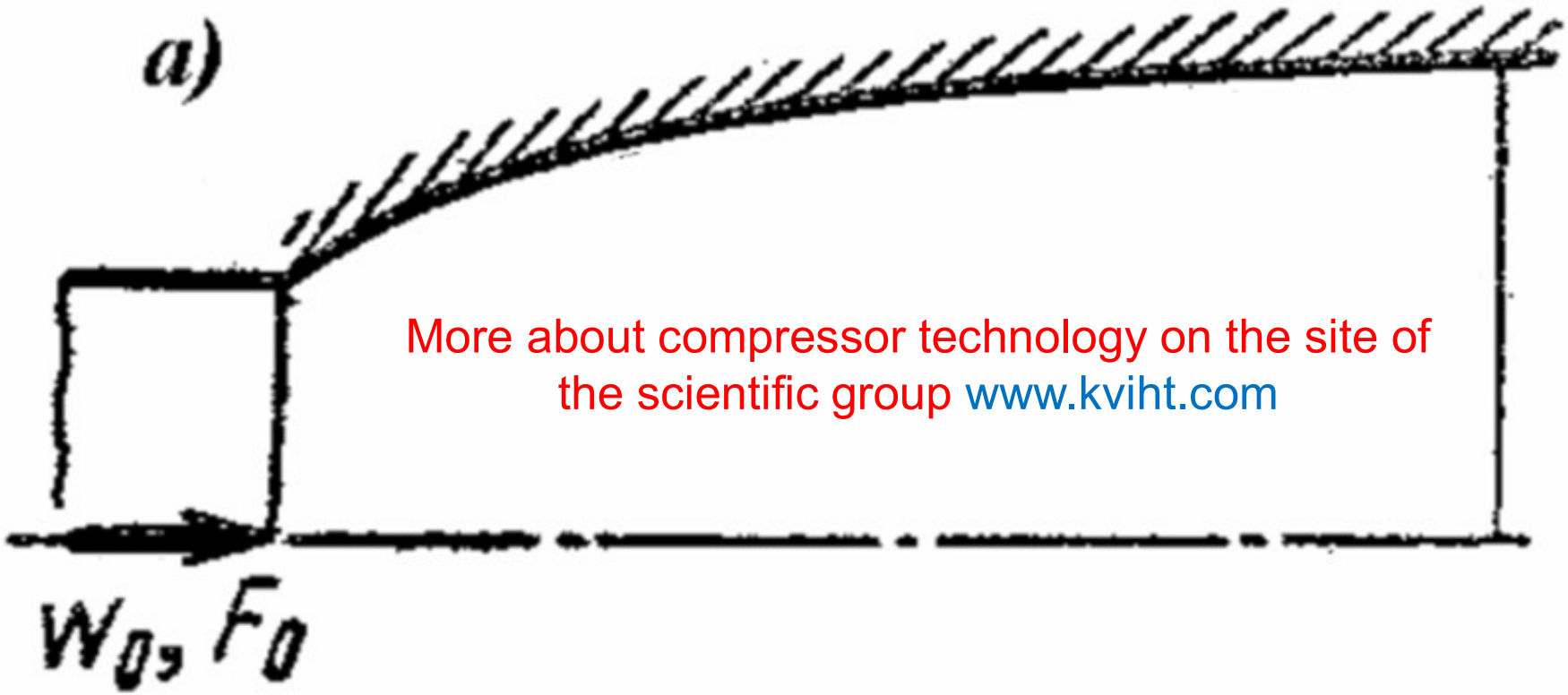

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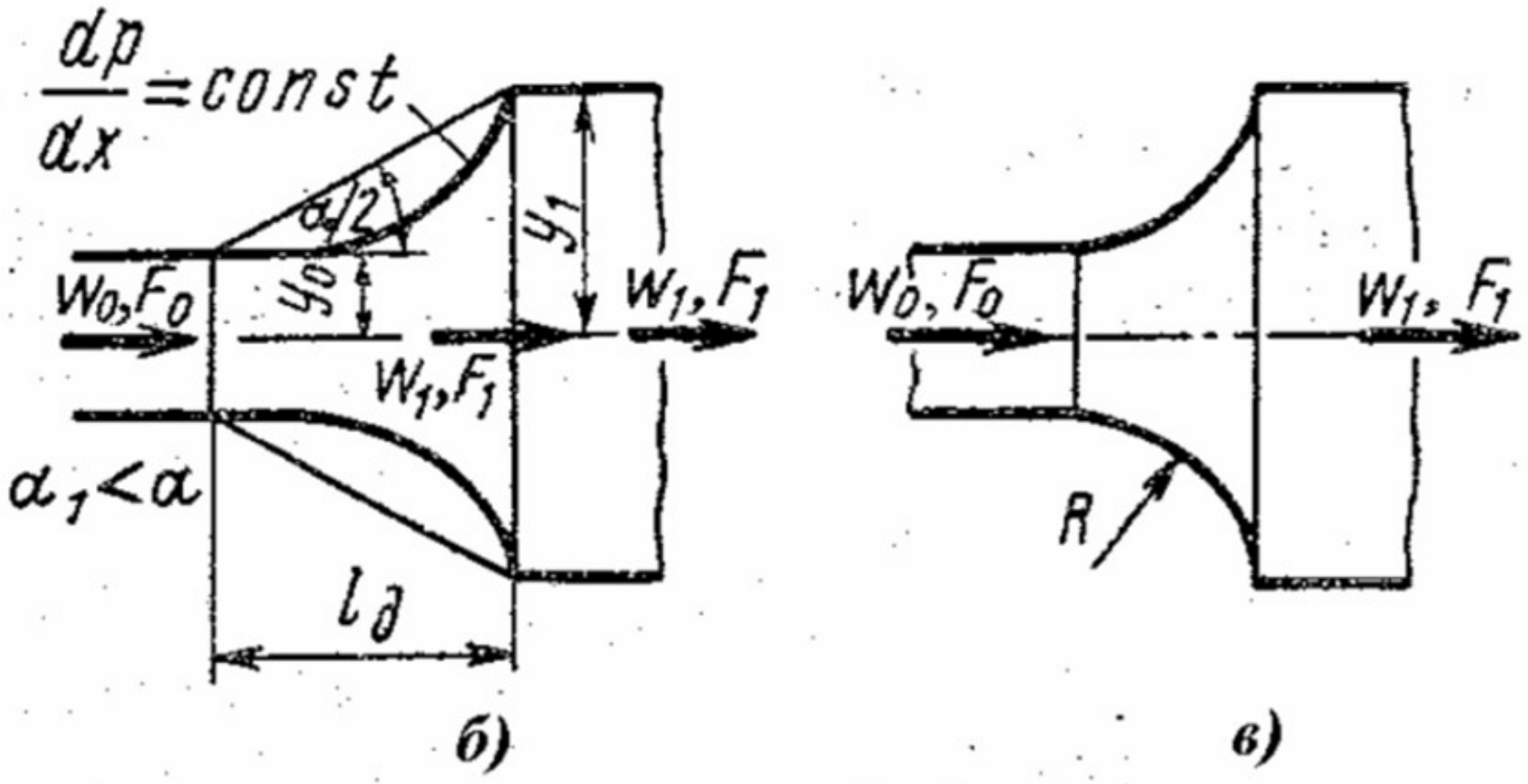
Losses decrease by means of special profiled walls of diffuser: Diffuser with before separation condition of the boundary layer



Losses decrease by means of special profiled walls of diffuser: diffuser with constant pressure gradient along the axis *dp/dx=const* (б); diffuser with formative by the curve circle totally (в)

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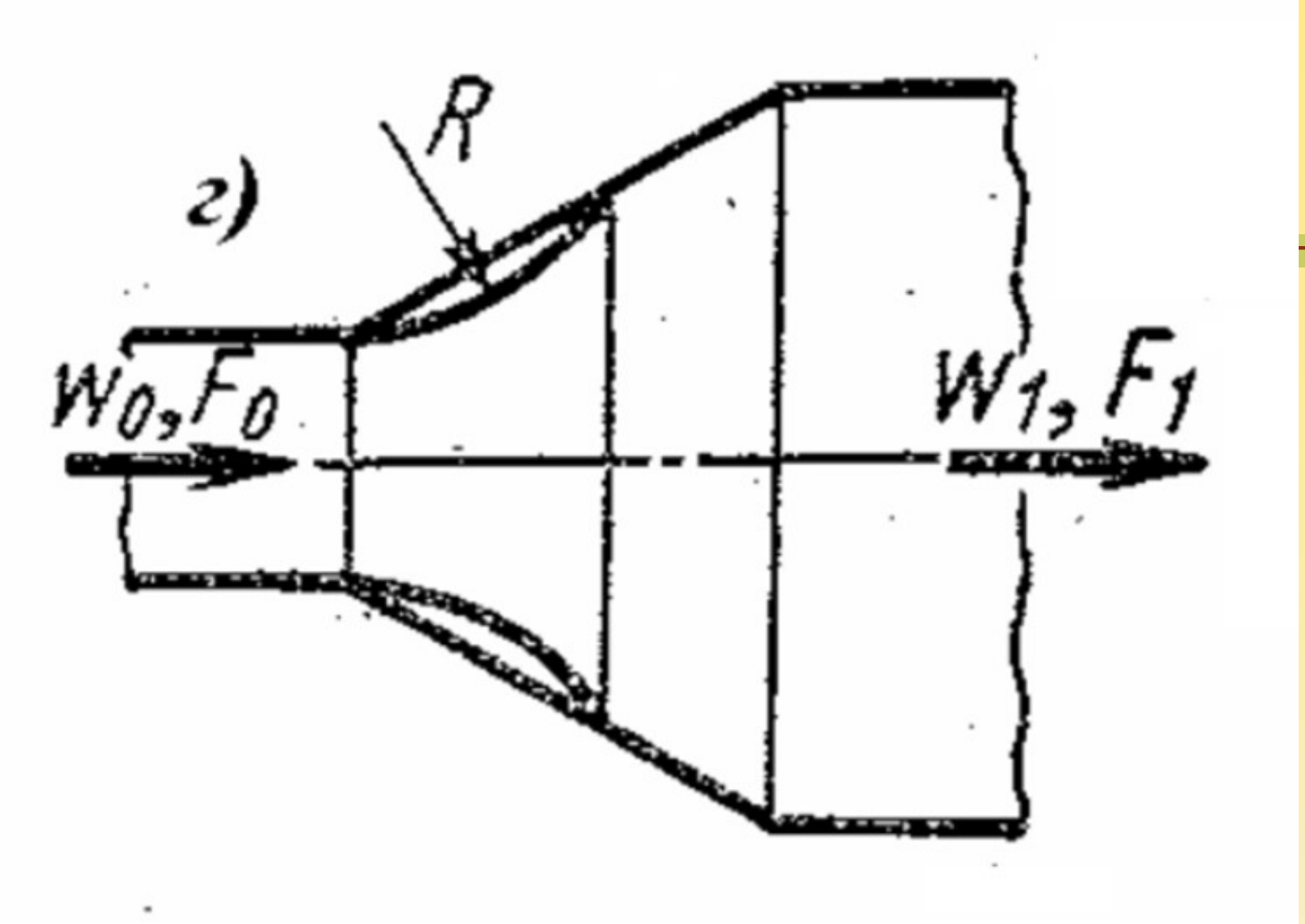

Losses decrease by means of special profiled walls of diffuser: diffuser with formative by the curve circle partly (г)

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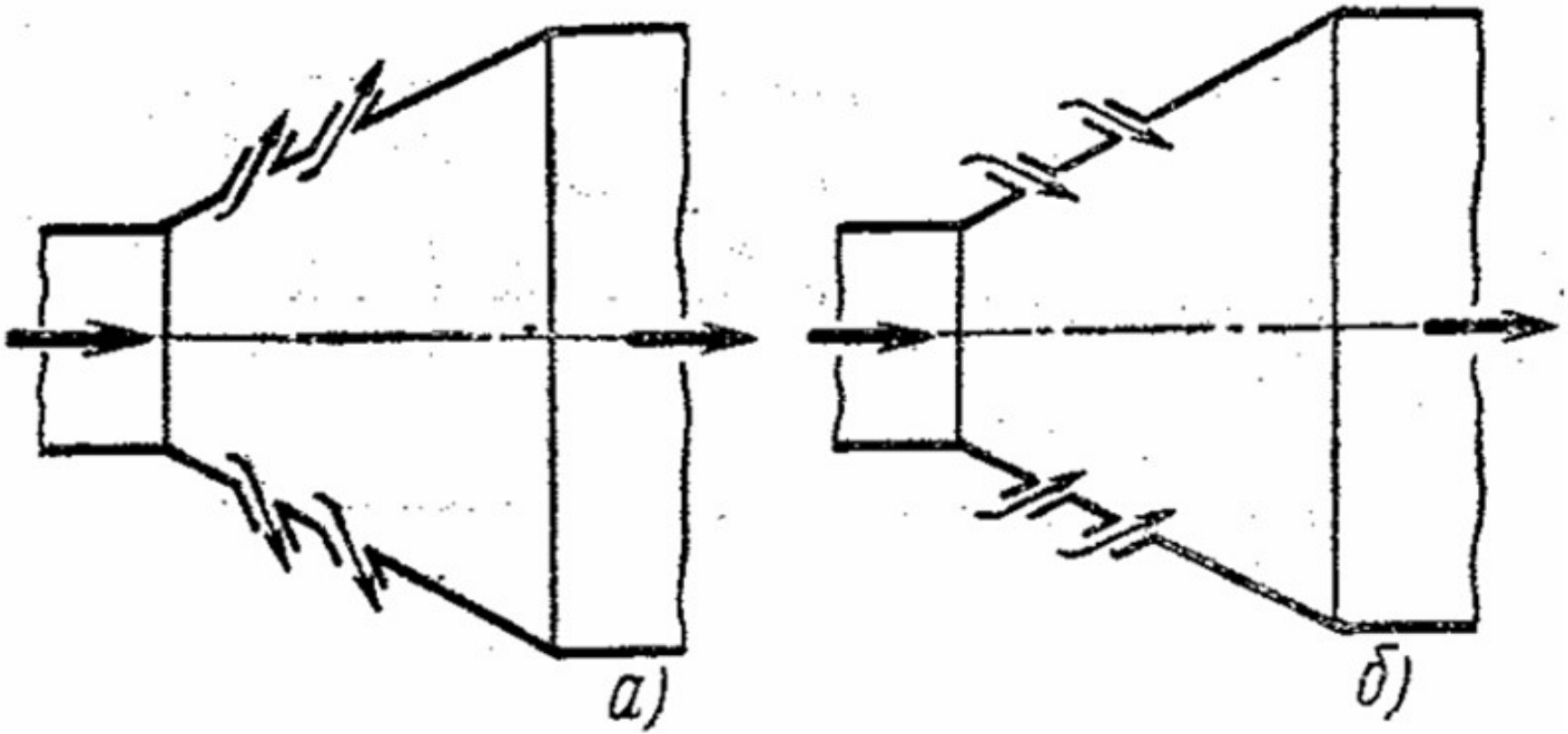

Losses decrease by means of suction (а) and blowing out (б) of the boundary layer

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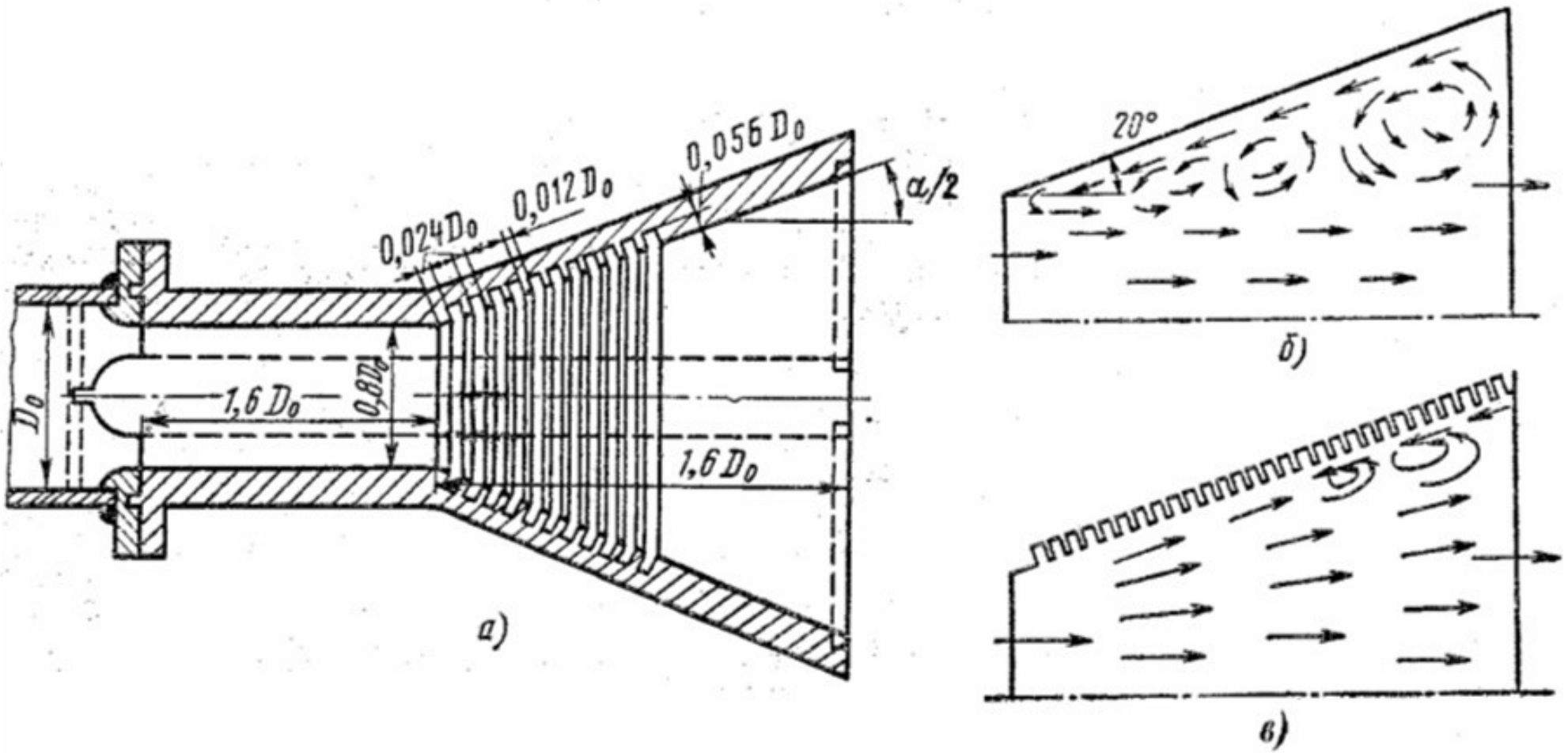

Losses decrease by means of cross finning of the walls of diffuser: optimal finning parameters (а); flow structure while plane (б) and finned (в) wall

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