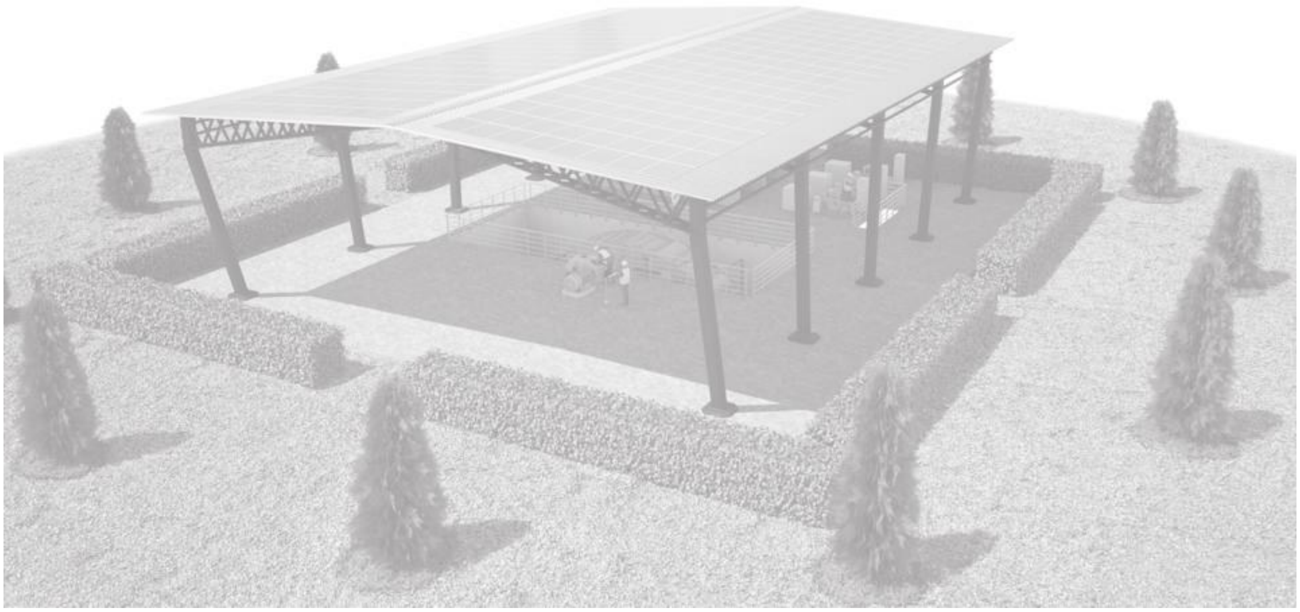
ESTIMATED ASSESSMENT OF THE MAIN PARAMETERS OF THE POWER GENERATION SYSTEM



Completed:

Ph.D. in mechanical engineering

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ANALYTICAL ANALYSIS OF TURBINE ELEMENTS

The task of this item is the analytical calculation of the parameters of the turbine, which affect the design of the product. This turbine should work well in a pool of water (fig. 1).

While working on this item, the number of blades will be analyzed.

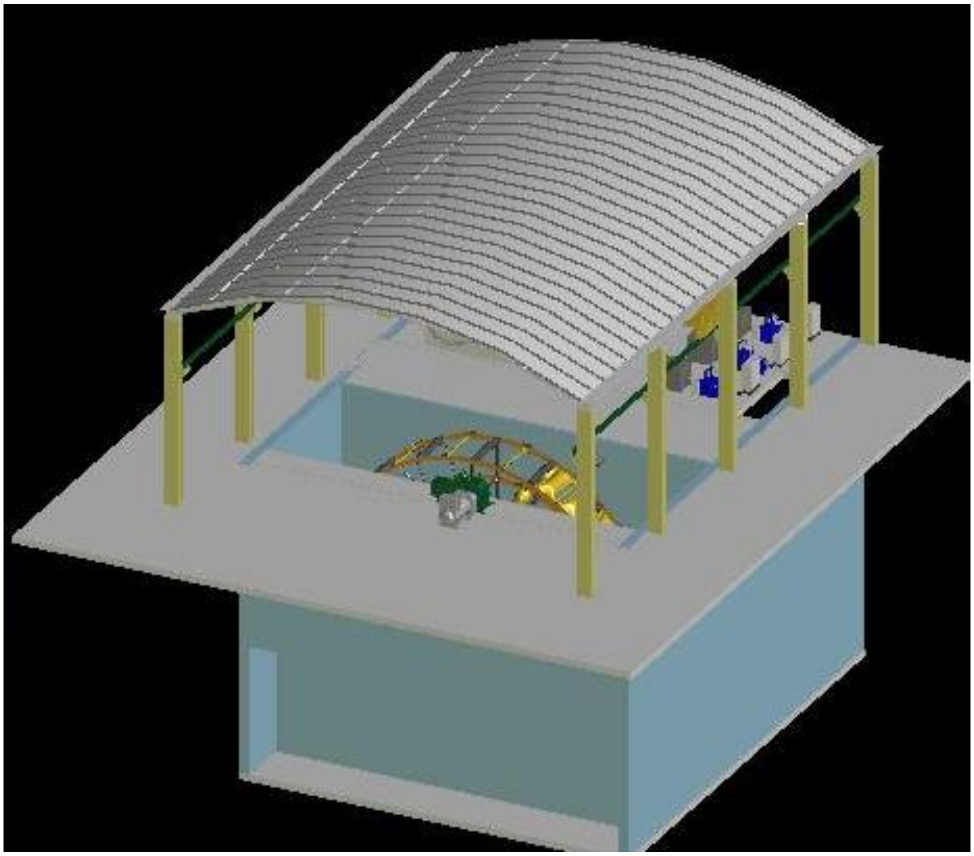

Fig. 1

The operation of the considered turbine plant is based on the principle of Archimedes' law.

$$F = \rho g V,$$

where F - buoyant force (Archimedes force), N;

ρ - water density, ρ =1000 kg/m^3;

g - acceleration of gravity, $g = 9.81$ m/s^2;

V - The volume of the extruded body. In this case, the volume of inflated balloons with air, m^3.

The desired power received at the generator terminals Np is

$$Np = 1 \text{ MW}.$$

Total number of balloons on turbine n_{Tot_b} is

$$n_{Tot\_b} = 64.$$

Initial wheel size (turbine wheel diameter) D_{in} is

$$D_{in} = 13 \text{ m}.$$

Initial area of one balloon A_{in_b} is 80 m^2.

$$A_{in\_b} = 80 \text{ m}^2.$$

Initial volume of one ballon V_{in_b} is

$$V_{in\_b} = 25.32 \text{ m}^3.$$

Assume that the number of inflated balloons in the water is ~ 1/3 of the total number of balloons on the turbine wheel.

Then initial number of inflated ballons n_{inf} is

$$n_{inf} = 16/3 \sim 5 \text{ ballons}.$$

Total volume of inflated ballons V_{Tot_b} is

$$V_{Tot\_b} = n_{inf} \cdot V_{in\_b} = 132.93 \text{ m}^3.$$

Then the pushing force (buoyant force or Archimedes force) created by the inflated balloons is

$$F = \rho g V_{Tot\_b} = 1000 \cdot 9.81 \cdot 132.93 = 1304043.3 \ N.$$

Torque moment M_{Torq} is determined by the formula

$$M_{Torq} = F \cdot L,$$

where

L - force arm or 1/2 turbine wheel diameter, $L=0.5 \cdot D_{in}=6.5\ m$. Then

$$M_{Torq} = 1304043.3 \cdot 6.5 = 8476281.45\ Nm$$

Let us determine the minimum rpm of turbine required to achieve the electric power at the terminals of the turbogenerator of 1 MW or 1000 kW according to the formula

$$n = \frac{9549 \cdot 1000 \cdot Np}{M_{Torq}} = 1.13\ rpm$$

In the project, we take as the nominal value n = 5 rpm, then the power will be equal to

$$Np = \frac{M_{Torq}}{n \cdot 9549 \cdot 1000} = \frac{8476281.45}{5 \cdot 9549 \cdot 1000} \cong 4{,}44\ MW$$

Let us determine the required minimum moment of resistance of the turbine shaft W according to the formula

$$W = \frac{\sqrt{M_b^2 + 0.45 \cdot M_{Torq}^2}}{[\sigma_b]}$$

where $[\sigma_b] = 90\ \ MPa$ - allowable bending stress [1].

M_b - bending moment from balloon, N·m.

Let's determine the maximum bending moment that occurs on the turbogenerator shaft according to the formula

$$M_{b1} = \frac{3 \cdot F \cdot L}{8} = 3178605.5\ Nm$$

$$M_{b2} = \frac{F \cdot L}{4} = 2119070.4\ Nm$$

For further calculations, we will use the maximum bending moment $M_b = M_{b1}$.

Then we have

$$W = \frac{\sqrt{3178605.5^2 + 0.45 \cdot 8476281.45^2}}{90} = 72380\ \text{cm}^3$$

Having the value of the moment of resistance of the turbine shaft *W*, we find the required minimum value of the shaft diameter D_{min} according to the formula

$$W = \frac{\pi D_{min}^3}{32} \rightarrow D_{min} = \sqrt[3]{\frac{32 \cdot W}{\pi}} = 90.3\ cm\ or\ 0.903\ m$$

<u>Fasteners</u>

Calculation of fasteners for the turbine-generator connection. Having a shaft diameter of 0.903 m, we see that the diameter of the mounting bolts D_f is 120 cm or 1.2 m. Let us determine the required minimum diameter of each bolt assuming the use of 24 bolts.

We have

$$M_{Torq(Dmin)} = M_{Torq(Df)}$$

$$\updownarrow$$

Let us determine the value of the force acting on the diametr of the bolts according to the formula

$$M_{Torq} = F_{Df} \cdot \frac{D_f}{2} \rightarrow \; F_{D_f} = \frac{2 \cdot M_{Torq}}{D_f}$$

In this case, the force acting on one bolt will be equal to

$$F_{D_{f\_b}} = \frac{F_{D_f}}{24} = \frac{M_{Torq}}{12 \cdot D_f} = \frac{8476281.45}{12 \cdot 1.2} = 588630.7\ N$$

Determine the required minimum bolt diameter D_{f_b} according to the formula for determining shear stresses τ_{sh}

$$\tau_{sh} = \frac{F_{D_{f\_b}}}{S_b} = \frac{4 \cdot F_{D_{f\_b}}}{\pi \cdot {D_{f\_b}}^2}$$

$$\updownarrow$$

$$D_{f\_b} = \sqrt{\frac{4 \cdot F_{D_{f\_b}}}{\pi \cdot \tau_{sh}}},$$

where $\tau_{sh} \leq [\tau]_{sh} = (0.2 \div 0.3) \cdot \sigma_\tau$.

Bearing for fastener steel $\sigma_\tau = 680\ MPa \rightarrow \tau_{sh} = 0.25 \cdot 680 = 170\ MPa$. Then

$$D_{f\_b} = \sqrt{\frac{4 \cdot 588630.7}{3.14159 \cdot 170}} \sim 66\ mm.$$

Thermodynamic calculation of a screw compressor

Initial data

$p_{in\_com} = 0.11\ MPa\ (absolute\ pressure)$;

$p_{out\_com} = 0.1\ MPa\ (overpressure)$;

the absolute pressure used in the calculations will be

$p_{out\_com} = 0.2\ MPa\ (absolute\ pressure)$;

$V_{Tot\_b} = \mathrm{n}_{\mathrm{Tot\_b}} \cdot Vin_b \cdot \frac{n}{60} = 33.76 \frac{m^3}{s}$;

$t_{air} = T_{in\_com\_1} = 293$ К $= 20$ °C;

$\beta_1 = 0.04,\ \beta_2 = 0.05$ (suction and discharge pressure losses);

$\eta_{mech} = 0.96,\ \eta_{eng} = 0.98$ (mechanical and engine efficiency);

$\sigma_1 = 1.1, \sigma_2 = 0.75$ (coefficients);

$\overline{L} = 1$ (relative rotor length);

$m_t = 0.8k$ (polytrophic temperature index);

$C = 0.003$ (temperature coefficient);

$\nu_{\mathrm{infl}} = 0.08$ (relative inflows);

$\delta = 50\ microns$ (allowable clearance);

Compressor type - screw compressor.

- ## Preliminary thermodynamic calculation

Total Compressor Pressure Ratio

$$\varepsilon_{general} = p_{out\_com}/p_{in\_com} = 0.2/0.11 = 1.818.$$

Let us determine the average design pressures in the working chamber during the actual cycle:

$$p'_{in_{com}} = p_{in\_com} \cdot (1 - \beta_1) = 0.1 \cdot (1 - 0.04) = 0.096\ MPa;$$

$$p'_{out\_com} = p_{out\_com} \cdot (1 + \beta_2) = 0.2 \cdot (1 + 0.05) = 0.21.$$

Preliminary pressure ratio

$$\lambda'_p = 1 - \beta_1 = 1 - 0.04 = 0.96.$$

Let us determine the ratio of pressures in the working chamber of the compressor during the actual cycle:

$$\varepsilon'_{general} = p'_{out_{com}} / p'_{in_{com}} = 0.21/0.096 = 2.188.$$

The temperature index of the compression polytrope with respect to finite parameters m_t:

$$m_t = 0.8 \cdot k = 0.8 \cdot 1.4 = 1.12.$$

Heating factor:

$$\lambda'_T = 0.98 - C \cdot \left(\varepsilon'_{general} - 1\right),$$

where $C = 0.003,$ we get

$$\lambda'_T = 0.98 - C \cdot \left(\varepsilon'_{general} - 1\right) = 0.98 - 0.003 \cdot (2.188 - 1) = 0.976.$$

Let us determine the temperature of the injected gas:

$$T_{out_{com}} = \frac{T_{in_{com}}}{\lambda'_T} \cdot \varepsilon'_{general}{}^{\frac{m_t - 1}{m_t}} = \frac{293}{0.976} \cdot 2.188^{\frac{1.12-1}{1.12}} = 326.48\ K.$$

Relative value of external leaks:

$$\nu'_{e.l.} = 0;$$

Relative value of internal inflows:

$$\nu'_{inf.} = 0.08;$$

Preliminary utilization rate:

$$\nu'_{util.} = 0.035;$$

Let's define a preliminary performance factor:

$$\lambda' = \lambda'_p \cdot \lambda'_T \cdot (1 - \nu'_{util.}) - \nu'_{inf.} - \nu'_{e.l.} = 0.96 \cdot 0.976 \cdot (1 - 0.035) - 0.08 = 0.8242.$$

Find the outer diameters of the driving d'_1 and driven d'_2 screws:

$$d'_1 = d'_2 = \sqrt{\frac{\pi \cdot V_{Tot\_b}}{\lambda' \cdot z_1 \cdot u''_1 \cdot \overline{L} \cdot (k_1 + k_2)}},$$

where V_{Tot_b} – compressor performance; $z_1 = 40$ – number of teeth of the lead screw; $u''_1 = 200\ \frac{m}{s}$ – linear speed at the outer diameter of the lead screw

$$d'_1 = d'_2 = \sqrt{\frac{3.14 \cdot 33.76}{0.8242 \cdot 40 \cdot 200 \cdot 1 \cdot (0.06943 + 0.049673)}} = 0.367\ m.$$

Find the preliminary number of revolutions of the leading n'_1 and driven n'_2 rotors:

$$n'_1 = \frac{60 \cdot u''_1}{\pi \cdot d'_1} = \frac{60 \cdot 200}{\pi \cdot 0.367} = 10413\ rpm;$$

$$n'_2 = \frac{n'_1}{1.5} = 6942\ rpm.$$

- **Selection and calculation of basic geometric parameters and characteristic angles**

To select and calculate the main geometric parameters, we use the quantities d_1' and $\overline{L}$ known from the thermodynamic

mic calculation, we choose the following parameters:

$d_1' = d_2' = 0.367\ m -$ external diameters of the driving and driven screws;

$d_{1s} = 0.64 \cdot d_1' = 0.235\ m -$ pitch circle diameter of the lead screw;

$d_{2s} = 0.96 \cdot d_1' = 0.352\ m -$ diameter of the starting circle of the driven screw;

$d_{1int} = d_{1int} = 0.22\ m -$ diameter of the inner circle of the driven and leading screws;

$A = 0.8 \cdot d_1' = 0.2936\ m -$ center distance;

$L = d_1' = 0.367\ m -$ screw length;

$H_1 = 1.2 \cdot d_1' = 0.4404\ m -$ axial lead screw pitch;

$H_2 = 1.5 \cdot H_1 = 0.6606\ m -$ axial pitch of driven screw;

$r = 0.18 \cdot d_1' = 0.066\ m -$ height of the head (leg) of the leading (slave) screw;

$h_0 = 0.02 \cdot d_1' = 0.0073\ m -$ height of the head (leg) of the driven (leading) screw;

$i_{12} = 1.5 -$ gear ratio;

$\tau_{13} = 300^0 -$ lead screw angle;

$\tau_{23} = \frac{\tau_{13}}{i_{12}} = 200^0 -$ angle of twist of the driven screw;

$\beta_{init} = 59^0 2917.208^{"} (1.03\ radian) -$ angle of inclination of the helix on the initial cylinders of the screws. Let's define characteristic angles:

1) β_{02} – the angle between the line of centers and the beam drawn from the center of the driven screw to the point of intersection of the initial circle of the driven screw with the outer circle of the lead screw:

$$\beta_{02} = arccos\frac{4\cdot A^2 + {d_{2s}}^2 - {d_1'}^2}{4\cdot A\cdot d_{2s}} = \arccos\left(\frac{4\cdot 0.2936\ ^2 + 0.352^2 - 0.367^2}{4\cdot 0.2936\cdot 0.352}\right)$$
$$= \arccos(0.808) = 36°\,5'\,56.459'' = 36.1\,° = (0.63004669\,radian).$$

2) β_{01} – the angle between the line of centers and the beam drawn from the center of the lead screw to the point of intersection of the initial circle of the driven screw with the outer circle of the lead screw:

$$\beta_{01} = arccos\frac{4\cdot A^2 + {d_1'}^2 - {d_{2s}}^2}{4\cdot A\cdot {d_1'}^2}$$
$$= \arccos\left(\frac{4\cdot 0.2936\ ^2 + 0.367^2 - 0.352^2}{4\cdot 0.2936\cdot 0.367}\right)$$
$$= \arccos(0.825) = 34°\,24'\,41.431'' = 34.4\,°$$
$$= (0.60059413\,radian).$$

3) α_2 – the central angle of the trailing edge of the cavity of the driven screw:

$$\alpha_2 = 28°\,15' = 28.07493\,° = (0.49\,radian).$$

4) α_{01} – angle of the end of the output of the tooth of the driven screw from the cavity of the leading screw on the pressure side:

$$\alpha_{01} = (\beta_{02} - \alpha_2)\cdot i_{12};$$

$$\alpha_{01} = (36.1 - 28.1)\cdot 1.5 = 12\,° = (0.20944\,radian);$$

5) τ_{1tw_lim} – limit angle of twist:

$$\tau_{1tw\_lim} = 2\cdot\pi - \frac{2\cdot\pi}{z_1} - (\beta_{01} + \alpha_{01}) = 2\cdot 180\cdot(1 - 0.025) - 46.4 = 304.6\,°.$$

6) angle ratio τ_{13} *and* τ_{1tw_lim}:

$$\Delta\tau = \tau_{13} - \tau_{1tw_{lim}} = 300 - 304.6 = -4.6\,°;$$

Let's determine the refined utilization factor:

$$\nu'_{util.} = 1 - \frac{V_{max}}{V_T} = \frac{k_3}{\overline{L} \cdot (k_1 + k_2)},$$

where V_T – theoretical volume of the steam cavity, due to its geometric parameters;

V_{max} – the maximum possible useful volume of the steam cavity with certain parameters of the screws and their profiles at the time of the start of compression;

k_1, k_2 – coefficient of use of the area of the leading and driven screws, for an asymmetric profile $k_1 = 0.06943, k_2 = 0.049673,\ k_3 = 0.00512$;

$$\nu'_{util.} = \frac{0.00512}{0.06943 + 0.049673} = 0.043.$$

7) φ_1 – the angle of rotation of the lead screw from the beginning of suction in the steam cavity to the achievement of its maximum volume V_{max}

$$\varphi_1 = \beta_{01} + \frac{2 \cdot \pi}{z_1} - 0.5 \cdot \Delta\tau = 34.4 + 0.157 + 2.3 = 36.857\,°;$$

8) opening angles of the suction port from the side driving α_{dw1} and driven α_{dw2} rotors:

$$\alpha_{dw1} = 0.5 \cdot \tau_{13} + \frac{\pi \cdot (z_1 - 1)}{z_1} = 0.5 \cdot 300 + \frac{180 \cdot (40 - 1)}{40} = 325.5\,°;$$

$$\alpha_{dw2} = \frac{\alpha_{dw1} + \frac{2 \cdot \pi}{z_1}}{i_{12}} - \alpha_1 = \frac{325.5 + 9}{1.5} - 13 = 210\,°;$$

9) Cavity volume at the beginning of compression V_{max}:

$$V_{max} = \overline{L} \cdot {d'_1}^3 \cdot (k_1 + k_2) \cdot (1 - \nu'_{util.})$$
$$= 0.367^3 \cdot (0.06943 + 0.049673) \cdot (1 - 0.043) = 0.01535\,m^3;$$

10) We determine the geometric degree of compression of the stage εg:

$$\varepsilon_{geom} = (\frac{P_2}{P_1})^{\frac{1}{k}} = (\frac{\sigma_2 \cdot p_{out\_com}}{\sigma_1 \cdot p_{in\_com}})^{\frac{1}{k}} = (\frac{\sigma_2}{\sigma_1} \cdot \varepsilon_{general})^{\frac{1}{k}} = (\frac{0.75}{1.1} \cdot 1.818)^{\frac{1}{1.4}} = 1.1658;$$

11) Determine the filled volume of the steam cavity:

$$V_{fill} = V_{max} \cdot \left(1 - \frac{1}{\varepsilon_{geom}}\right) = 0.01535 \cdot \left(1 - \frac{1}{1.1658}\right) = 0.00218\, m^3;$$

12) Discharge cavity volume:

$$V_2 = V_{max} - V_{fill} = 0.01535 - 0.00218 = 0.01317\, m^3;$$

The pressure of the beginning of polytrophic compression in the steam cavity:

$$P_1 = p_{in\_com} \cdot \sigma_2 = 0.1 \cdot 1.1 = 0.11\, MPa;$$

Internal compression end pressure:

$$P_2 = P_1 \cdot \left(\frac{V_{max}}{V_2}\right)^k = 0.11 \cdot \left(\frac{0.01535}{0.01317}\right)^{1.4} = 1.239\, MPa;$$

Compression angle φ_{lc}

$$\varphi_{lc} = 265\,° = (4.62\, radian).$$

- **Construction of the indicator diagram of the steam cavity**

We build a schematized indicator diagram using the following parameters:

Along the abscissa:

$V_{max} = 0.01535\, m^3$;

$V_2 = 0.01317\, m^3$;

$\nu'_{util.} = 0.043$;

$V_T = \overline{L} \cdot {d'_1}^3 \cdot (k_1 + k_2) = 0.367^3 \cdot (0.06943 + 0.049673) = 0.016\, m^3.$

Along the y-axis:

$p_{in\_com} = 0.1\, MPa;$

$P_1 = 0.11\, MPa;$

$P_2 = 1.239\, MPa;$

$p_{out\_com} = 0.2\, MPa;$

$\Delta p_{in\_com} = \beta_1 \cdot p_{in_{com}} = 0.04 \cdot 0.1 = 0.004\, MPa;$

$\Delta p_{out\_com} = \beta_2 \cdot p_{out_{com}} = 0.05 \cdot 0.2 = 0.01\, MPa.$

The compression line is an adiabatic and the points on it are determined based on the following equation:

$$P_i = P_1 \cdot \left(\frac{V_{max}}{V_i}\right)^k;$$

See Fig. 2 for the indicator diagram.

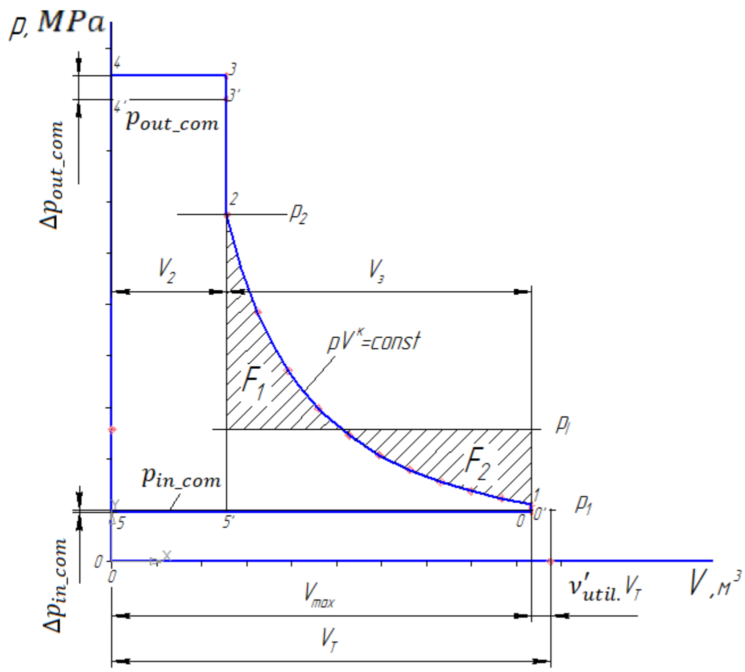

Fig. 2

The average integral pressure was found graphically:

$$P_I = 0.058\ MPa;$$

- **Calculation of power consumption. Drive selection.**

The power supplied to the compressor unit with a screw compressor is spent on compressing and moving gas (indicative power), on overcoming mechanical friction in the compressor elements (mechanical power), and on driving auxiliary mechanisms (oil and water pumps, fan, multiplier, etc.).

The indicator power is determined from the indicator diagram:

$$A_{\text{ind}} = V_{fill} \cdot P_I + p_{out_{com}} \cdot V_2 - V_{max} \cdot p'_{in_{com}} = 0.00218 \cdot 0.058 + 0.2 \cdot 0.01317 - 0.01535 \cdot 0{,}096 = 1287 \text{ J},$$

At the same time, the indicator power

$$N_{\text{ind}} = A_{\text{ind}} \cdot z_1 \cdot n_1/600 = 1287 \cdot 40 \cdot \frac{10413}{600} = 893.4\ kW.$$

- Power consumed by the compressor

$$N_{com} = \frac{N_{ind}}{\eta_{mech}} = \frac{893.4}{0.96} = 930.7kW,$$

where

$\eta_{mech} = 0.96 -$ mechanical efficiency, taking into account friction losses.

- Useful power gain

$$\Delta N = Np - N_{com} = 4440 - 930.7 = 3509.3kW = 3.51MW.$$

$$\eta_{\Delta N} = \Delta N \cdot \frac{100}{Np} = \frac{3.51 \cdot 100}{4.44} = 79.1\%.$$

Turbine mass determination

The turbine is made of aluminum (Fig. 3). The mass of the material was determined using the SolidWorks software package

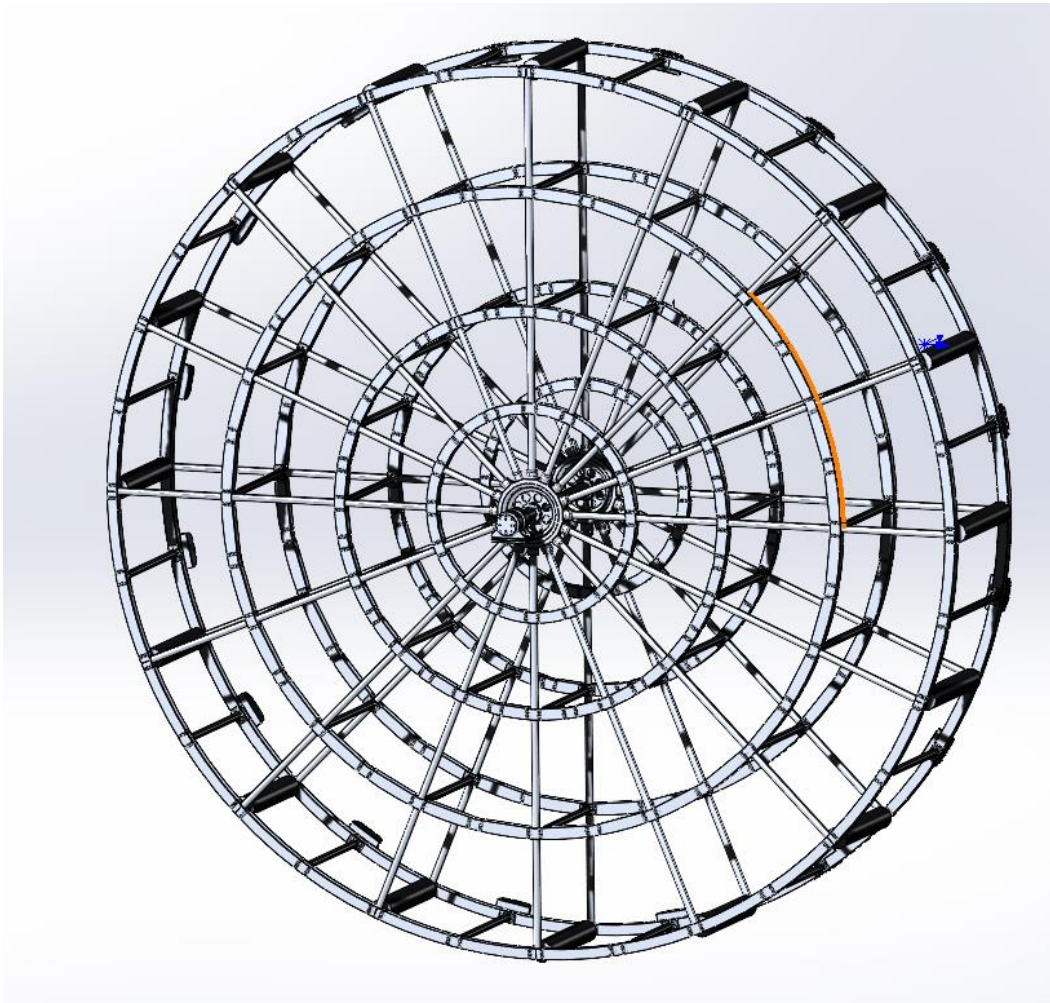

Fig. 2

Material density – 2700 kg/m^3;

Volume – 5.85 m^3;

Turbine surface area – 347.49 m^2;

The mass of the turbine is 15.79 tons.

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