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EXPERIMENTAL VERIFICATION OF THEORETICAL WORK OF ROTOR

SUMMARY

The article presents results of the flow tests of a fan without casing based on calculations of increase of energy in rotor, efficiency and power on the drive shaft as well as thermoanemometric measurements behind the rotor. Results of the tests were compared with whirl work calculated analytically aiming at improvement of analytic calculations.

Keywords*: rotor work, flow test of a fan, analytically calculated work, energy losses in rotor*

EKSPERYMENTALNA WERYFIKACJA TEORETYCZNEJ PRACY WIRNIKOWEJ

W artykule przedstawiono wyniki badań przepływowych wentylatora bez obudowy, polegających na eksperymentalnym wyznaczeniu przyrostu energii w wirniku, sprawności i mocy na wale oraz pomiarów termoanemometrycznych za wirnikiem. Rezultaty badań porównano z pracą wirnikową wyznaczoną analitycznie w celu udoskonalenia analitycznych metod obliczeń.

Słowa kluczowe*: użyteczna praca wirnikowa, testowanie wentylatora, analityczne określenie pracy, straty energii w wirniku*

NOMENCLATURE

A_t – area of the volute
a' – factor in Ekert's formula
C – coefficient of flow of the orifice
f – coefficient of power
p_b – barometric pressure
p_s – pressure in suction
N_u – useful power
N_i – shaft power
$N_i = N_w$
n' – experimental speed of rotation
S – rotor number
T_1 – temperature before the orifice
u_1, u_2 – blade speed on the rotor inlet and outlet
$\dot{V}_n$ – nominal capacity
$\dot{V}_s$ – fan capacity
α_{3t}, y – factor on Sentek's formula
β – orifice throat
η_i – inner fan efficiency
η_{sil} – efficiency of motor
κ – isentropic factor
μ – slip factor
ξ_k – loss coefficient in the set chamber-cone-pipe
ξ_n – coefficient of non-tangential inflow
ξ_t – coefficient of friction loss in the rotor
ρ_{cm} – manometric fluid density
ρ_{alk} – alcohol density
ρ_t – air density
φ – flow coefficient
χ_1, χ_2 – reduce coefficient of area of rotor
Ψ' – factor in Pfleiderer's formula
w_{1a} – tangential component of relative velocity
w_{1b} – nontangential component of relative velocity

1. INTRODUCTION

Theoretical work of rotor is defined for inviscid fluid flow in a real radial rotor. It may be determined analytically applying angular momentum equation, assuming radial inflow to blades and contact of speed vectors with blade curvature, taking into account power reduction value.

Generally known formulas for calculating the value of power reduction usually depend on three geometrical parameters, that is, number of blades, blade outlet angle, and diameter ratio. These formulas define the numerical value only for the nominal point understood as the design point.

New formulas have been developed in recent years to improve precision of calculation results, taking into account further geometrical parameters, such as width of the rotor and variable output along the characteristic curve. Thanks to these calculations, analytically determined theoretical work of the rotor may be closer to "reality". It should be verified with balance measurements and measurements of flow structure behind the rotor. The first type of measurements allows determination of inner operation corresponding with the theoretical operation of the rotor at the working part of the characteristics. The second type of measurements with a three-fibre thermoanemometric sensor allows determination of absolute peripheral velocity field components at the rotor outlet. The averaging velocity value of this field allows calculation of rotor operation. The thermoanemometric measurements allow determination of various turbulence parameters in connection with the efficiency of a machine.

The paper deals with the procedure of analytical determination of theoretical rotor operation, results of balance thermoanemometric examinations, compared to each other in order to improve the analytical method.

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2. LIST OF GEOMETRICAL DATA AND ROTOR DIAGRAM

Aerodynamic diagram of the rotor (Fig. 1) is based on geometrical data of the rotor shown in Table 1.

3. DETERMINATION OF ROTOR CHARACTERISTICS

Measuring position for examination of characteristics is shown at Figure 2.

Values listed in Table 2 were measured at the position.

Table 1. List of geometrical dimensions of the rotor

d_s = 225 mm	pipe diameter
d_1 = 203 mm	cone diameter
d_w = 208 mm	inlet chamber diameter
D_o = 225 mm	internal rotor diameter at the side of inlet chamber
D_1 = 225 mm	internal rotor diameter taking into account thickness of blades
D_2 = 315 mm	external rotor diameter taking into account thickness of blades
D_3 = 315 mm	external rotor diameter at the side of casing
s = 2,5 mm	slot at the inlet
β_1 = 8°	blade angle at the inlet
β_2 = 45°	blade angle at the outlet
γ = 13°	front disk inclination angle
b_1 = 75 mm	rotor width at the inlet
b_2 = 54 mm	rotor width at the oulet
H = 97 mm	rotor sleeve diameter
L = 98 mm	blade length
g = 1,5 mm	blade thickness
z = 9	number of blades
α_V = 0,57	slot coefficient
$k_{V=}$ 0,8	pressure ratio at the slot
A = 225 mm	outlet window height
B = 180 mm	casing width
n = 2880 r.p. m	speed of rotation
ω = 301,44 rd/s	rotor angular velocity

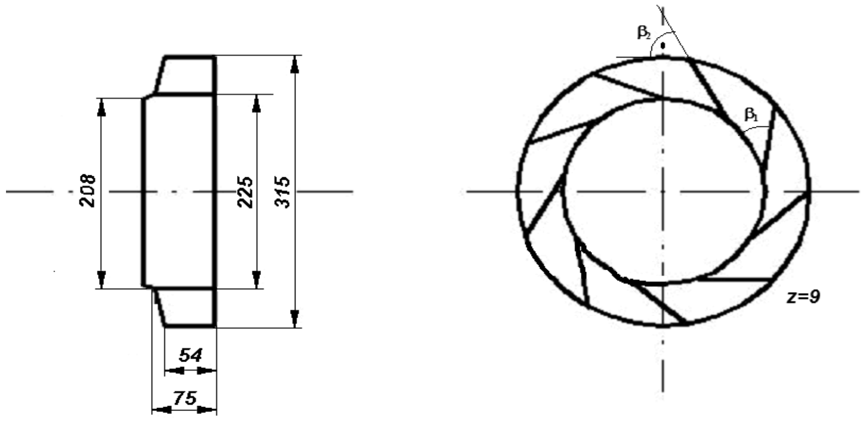

Fig. 1. Rotor Diagram

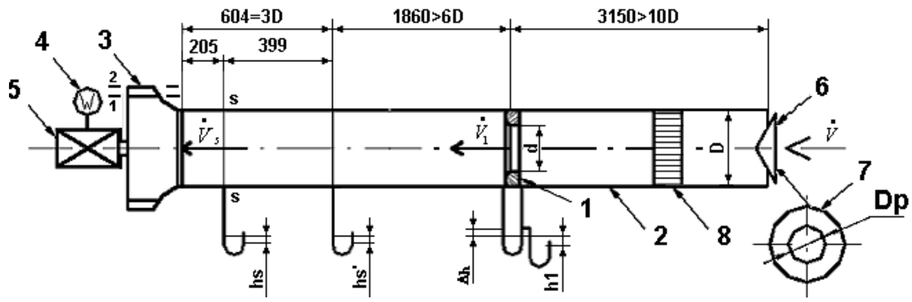

Fig. 2. Diagram of a laboratory measurement position – disk measurement [2]: 1 – measuring orifice ISA D 220/d 155, 2 – pipeline, 3 – fan, 4 – wattmeter, 5 – motor drive (direct drive), 6 – reducer, 7 – shutter, 8 – straightener

Table 2. List of results of laboratory measurements

Description	Unit	1	2	3	4	5
Negative pressure height ahead of orifice	mm alcohol	24	76	105	110	119
Differential pressure height Δh	mm alcohol	158	90	41	26	14
Negative pressure height at suction h_s	mm alcohol	110	120	126	126	126
Wattmeter reading	scale interval	40.5	37	33	30	27

4. ALGORITHM OF ROTOR FLOW PARAMETERS CALCULATION

Calculation of rotor capacity $\dot{V}_s$ at the inlet

Capacity is calculated according to the formula because

$$\dot{V}_s = \dot{V}_1 \cdot \frac{\rho_1}{\rho_s} \tag{1}$$

$$\dot{m} = \dot{V}_1 \cdot \rho_1 = \dot{V}_s \cdot \rho_s$$

where:

$\dot{V}_1$ – capacity ahead of the orifice,

$\dot{V}_s$ – rotor efficiency,

p_s, ρ_s – absolute pressure and air density in A_s section,

p_1, ρ_1 – absolute pressure and air density ahead of orifice.

Capacity at the reducer is calculated according to the formula

$$\dot{V}_1 = K \cdot \sqrt{\Delta h} \quad \left[\frac{m^{2.5}}{s}\right] \tag{2}$$

where:

$$K = \frac{C}{\sqrt{1-\beta^4}} \cdot \varepsilon_1 \cdot \frac{\pi d^2}{4} \cdot \frac{\sqrt{\rho_{alk} \cdot g \cdot 10^{-3}}}{\sqrt{\rho_1}} \cdot \sqrt{2} \quad \left[\frac{m^3}{s}\right] \tag{3}$$

data are presented in Δh [alcohol mm].

$$C = 0.5959 + 0.0312\beta^{2.1} - 0.184\beta^8 + +0.0029\beta^{2.5} \cdot \left(\frac{10^6}{\mathrm{Re}_D}\right)^{0.75} \tag{4}$$

$$\varepsilon_1 = 1 - \left(0.41 + 0.35\beta^4\right) \cdot \frac{\Delta p}{\kappa \cdot p_1} \tag{5}$$

$$\Delta p = \Delta h \cdot \rho_{cm} \cdot g \tag{6}$$

$$\rho_1 = \rho_N \cdot \frac{(p_1 - \varphi \cdot p_p'') \cdot T_N}{p_N \cdot T_1} + \varphi \cdot \rho_p'' \tag{7}$$

$$p_1 = p_b - h_1 \cdot \rho_{cm} \cdot g \tag{8}$$

$$p_s = p_b - h_s \cdot \rho_{cm} \cdot g \tag{9}$$

C – fluid flow coefficient,

ε_1 – expansion number,

d – orifice hole diameter.

Calculation of total pressure difference Δp_c

Total pressure difference is calculated according to the formula

$$\Delta p_c = \Delta p_{st} + \Delta p_d \tag{10}$$

where:

Δp_{st} – static pressure difference,

Δp_d – dynamic pressure difference,

$$\Delta p_{st} = \frac{h_s \cdot \rho_{mm} \cdot g}{10^3} \text{ [Pa]} \tag{11}$$

$$\Delta p_d = \frac{1}{2} \rho_{śr} \cdot \dot{V}_s^2 \cdot \left(\frac{1}{A_t^2} - \frac{1}{A_s^2}\right) \text{[Pa]} \tag{12}$$

$$\rho_{śr} = \frac{\rho_s + \rho_t}{2} \tag{13}$$

$$\rho_t = \rho_1 \cdot \frac{p_t}{p_1} \text{ – air density in } A_t \text{ section} \tag{14}$$

$$\rho_s = \rho_1 \cdot \frac{p_s}{p_1} \text{ – air density in } A_s \text{ section} \tag{15}$$

$p_t = p_b$ – exists at the measuring position

$$A_t = \pi \cdot D_2 \cdot b_2 = 0.0534 \text{ m}^2 \tag{16}$$

Calculation of effective power N_u

$$N_u = \Delta p_c \cdot \dot{V} \cdot f \tag{17}$$

where $f = 1$.

Calculation of power at rotor shaft N_i

$$N_i = N_{el}\,\eta_{sil} \tag{18}$$

Calculation of rotor inner efficiency η_i

$$\eta_i = \frac{N_u}{N_i} \tag{19}$$

Conversion of total pressure difference and power into conventional equal density 1.2 kg/m^3:

$$\Delta p_{c1,2} = \Delta p_c \cdot \frac{1.2}{\rho_{śr}} \tag{20}$$

$$N_{w1,2} = N_w \cdot \frac{1.2}{\rho_{śr}} \tag{21}$$

Conversion of capacity, total pressure difference, and rotor power into rated rotations n = 2880 rev/min:

$$\dot{V}_s = \dot{V}_s{}' \cdot \frac{n}{n'} \tag{22}$$

$$\Delta p_c = \Delta p_c{}' \cdot \left(\frac{n}{n'}\right)^2 \tag{23}$$

$$N_w = N_w{}' \cdot \left(\frac{n}{n'}\right)^3 \tag{24}$$

Results of rotor examinations are shown in Table 2, results of calculations in Table 3 and diagram (Fig. 3).

5. THERMOANEMOMETRIC MEASUREMENTS

Thermoanemometric measurements behind the rotor have been widely described in publications [1, 3]. Results of measurements performed for five different rotations described in those publications were converted into rotations 2880 [rev/min], and provided with flow characteristics of the rotor shown in Figure 3.

Table 3. List of results of calculations and conversion of characteristics into rated rotations and conventional density

Measurement	Unit	0	1	2	3	4
Rotary speed of the rotor	rev/min	2830	2875	2890	2900	2920
Capacity $\dot{V}_s$	m^3/s	0.5979	0.4526	0.3062	0.2439	0.1791
Power at shaft N_i	kW	0.887	0.810	0.723	0.657	0.591
Total pressure difference Δp_c	Pa	817.6	969.8	1000.6	1007.6	1013.2
Power at shaft $N_{i\rho1,2}$	kW	0.898	0.818	0.728	0.661	0.595
Total pressure difference $\Delta p_{c\rho1,2}$	Pa	827.4	978.6	1007.2	1013.9	1018.8
Fan efficiency η_i	–	0.551	0.542	0.424	0.374	0.307
Capacity $\dot{V}_s$ for n = 2880 and ρ = 1.2	m^3/s	0.6085	0.4534	0.3051	0.2423	0.1767
Power at shaft N_i For n = 2880 i ρ = 1.2	kW	0.946	0.822	0.720	0.648	0.570
Total pressure difference Δp_c for n = 2880 and ρ = 1.2	Pa	856.9	982.0	1000.3	999.9	991.0
Fan efficiency η_i for n = 2880 and ρ = 1.2	–	0.5519	0.5418	0.4237	0.3737	0.3060
Re′	–	956742	971955	977026	9804407	987168
Re for n = 2880	–	973645	973645	973645	973645	973645

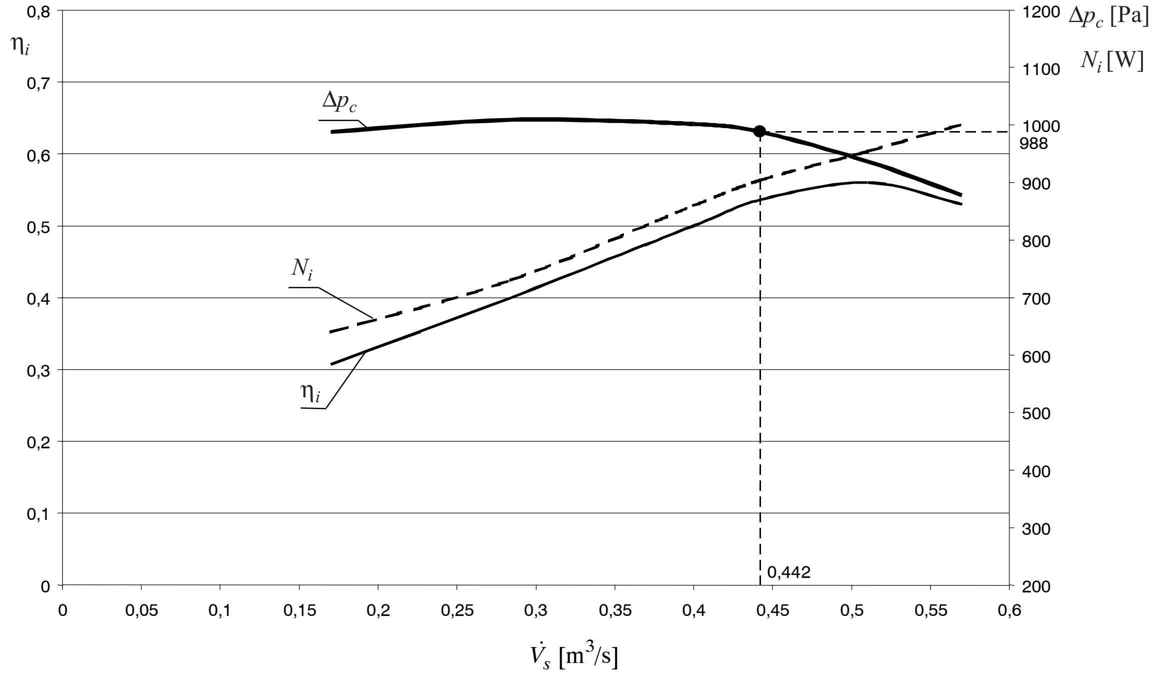

Fig. 3. Flow characteristics of the rotor

Table 4 presents results of conversion of capacity $\dot{V}_s$, effective rotor operation l_{u12} determined according to components c_{2m} i c_{2u} measured with an thermoanemometer and the obtained average value.

Table 4. List of results of conversion of capacity and operation determined with thermoanemometric measurements

It.	n' rev /min	n rev/min	$\frac{n}{n'}$	$\left(\frac{n}{n'}\right)^2$	n = 2880 rev/min	
					$\dot{V}_s$ m^3/s	l_{u12} m^2/s^2
1	439	2880	6.56	43.04	0.439	1024
2	897	2880	3.21	10.31	0.435	912
3	1368	2880	2.10	4.43	0.429	902
4	2038	2880	1.41	2.00	0.467	921
5	2260	2880	1.27	1.62	0.439	992
Average					0.442	950

6. CALCULATION OF POWER REDUCTION VALUE ACCORDING TO PARTICULAR FORMULAS

Calculation of power reduction value according to Eck's formula:

$$\mu = \frac{1}{1+\dfrac{1.5+1.1\cdot\dfrac{\beta_2}{90^\circ}}{2z\cdot\left(1-\dfrac{r_1}{r_2}\right)}} = 0.71,$$

for br = const; $\beta_2 = 30\div50$.

Calculation of power reduction value according to Eckert's formula

$$\mu = \frac{1}{1+\dfrac{a'\cdot\pi\cdot\sin\beta_2}{a\cdot 2z\cdot\left(1-\dfrac{r_1}{r_2}\right)}} = 0.75,$$

for $\frac{a'}{a} = 0.8$; br = const.

Calculation of power reduction value according to Pfleiderer's formula:

$$\mu = \frac{1}{1+\dfrac{2\psi'}{z\cdot\left[1-\left(\dfrac{r_1}{r_2}\right)^2\right]}} = 0.59\div0.66,$$

$$\psi' = 0.65\cdot\left(1+\frac{\beta_2}{60^\circ}\right) \text{and } \psi' =$$

$$= 0.85\cdot\left(1+\frac{\beta_2}{60^\circ}\right) \text{for } \beta_2 < 90^\circ.$$

Calculation of power reduction value according to Stodola's formula:

$$\mu_0 = 1-\frac{\pi}{z}\cdot\sin\beta_2,$$

$$\mu_0 = 0.75,$$

Calculation of power reduction value according to Strachowicz's formula:

$$\mu = \frac{1}{1+\frac{2\pi \cdot \sin 2\beta_2}{3z}} = 0.81,$$

$$\beta_2 \le 45°.$$

Calculation of power reduction value according to Sentek's formula

$$\lg\mu = -2.78 \cdot 10^{-4}\left[-\lg\left(\mathrm{tg}\alpha_{3t}\right)\right]^{0.95-4.22\frac{b_1}{D_1}} \cdot 10^{y}\left(\lg z\right)^{-2.2} =$$

$$= -0.1073 \Rightarrow \mu = 0.70,$$

where:

$$\mathrm{tg}\alpha_{3t} = \frac{\frac{\dot{V}\chi_2}{\dot{V}_n}\mathrm{tg}\beta_2}{\left(S - \frac{\dot{V}}{\dot{V}_n}\right)} = \frac{\frac{0.44}{0.252}\cdot 1 \cdot 0.98}{11.4 - 1.75} = 0.1777,$$

$$\dot{V}_n = \pi \cdot D_1 \cdot b_1 \cdot u_1 \cdot \mathrm{tg}\beta_1 = 0.252 \text{ m}^3/\text{s}.$$

$$y = \left(8.07 - 5.87 \cdot \frac{D_1}{D_2}\right) \cdot \frac{D_1}{D_2} - 0.39 \cdot \frac{b_1}{D_1} = 2.639.$$

Calculation of μ value is shown in Table 5.

Table 5. List of power reduction values calculated according to various formulas

It.	μ value	According to
1	0.71	Eck's formula
2	0.75	Eckert's formula
3	0.59÷0.66	Pfleiderer's formula
4	μ_0 = 0.75	Stodola's formula
5	0.81	Strachowicz's formula
6	0.70	Sentek's formula

7. ANALYTICAL DETERMINATION OF EFFECTIVE WHIRL WORK

Work in Euler's rotor is determined according to the formula

$$l_{ut\infty} = u_2 \cdot \left(u_2 - \frac{\dot{V}}{\pi \cdot D_2 \cdot b_2 \cdot \mathrm{tg}\beta_2}\right) \tag{25}$$

At nominal point this work equals

$$l_{ut\infty n} = u_2 \cdot \left(u_2 - \frac{\dot{V}_n}{\pi \cdot D_2 \cdot b_2 \cdot \mathrm{tg}\beta_2}\right) = \tag{26}$$

$$= 2033 \text{ m}^2/\text{s}^2$$

where

$$\dot{V}_n = \pi \cdot D_1 \cdot b_1 \cdot u_1 \cdot \mathrm{tg}\beta_1 = 0.252 \text{ m}^2/\text{s}.$$

Work reduction resulted from finite number of blades calculated with component method is

$$l_\mu = l_{ut\infty n} \cdot (1-\mu).$$

Calculations are based on μ values presented in Table 5:

$$l_{\mu 1} = 2033 \cdot 0.29 = 590 \text{ m}^2/\text{s}^2,$$

$$l_{\mu 2} = 2033 \cdot 0.25 = 508 \text{ m}^2/\text{s}^2,$$

$$l_{\mu 3} = 2033 \cdot (0.41 \div 0.34) = 833 \div 691 \text{ m}^2/\text{s}^2,$$

$$l_{\mu 4} = 2033 \cdot 0.25 = 508 \text{ m}^2/\text{s}^2,$$

$$l_{\mu 5} = 2033 \cdot 0.19 = 386 \text{ m}^2/\text{s}^2,$$

$$l_{\mu 6} = 2033 \cdot 0.30 = 610 \text{ m}^2/\text{s}^2.$$

At the characteristics point with capacity

$$\dot{V}_s = 0.442 \text{ m}^3/\text{s}.$$

Work $l_{ut\,\infty} = u_2\,(u_2 - 0.442/\pi D_2\, b_2 \,\mathrm{tg}\,\beta_2) = 1863\ (\text{m/s})^2$.
Theoretical whirl work is calculated with component based method according to the formula

$$l_{ut} = l_{ut\infty} - l_\mu \tag{27}$$

or factor based method according to the formula

$$l_{ut} = l_{ut\infty} \cdot \mu \tag{28}$$

Work l_{ut} for determined power reduction values is shown in Table 6.

Table 6. Theoretical whirl work calculated for various power reduction and loss values

<table>
<tr><td rowspan="4">It.</td><td colspan="2">Work</td><td colspan="7">Losses</td></tr>
<tr><td>$l_{ut\infty} - l_{\mu}$</td><td>$\mu\ l_{ut\infty}$</td><td>l_k</td><td>l_t</td><td>l_n</td><td>l_w</td><td>l_v</td><td>l_b</td><td>Σ</td></tr>
<tr><td colspan="9">[m^2/s^2]</td></tr>
<tr><td>1</td><td>2</td><td>3</td><td>4</td><td>5</td><td>6</td><td>7</td><td>8</td><td>9</td></tr>
<tr><td>1</td><td>1273</td><td>1322</td><td rowspan="6">57</td><td>221</td><td>140</td><td>210</td><td>70</td><td>15</td><td>713</td></tr>
<tr><td>2</td><td>1355
1030÷1342</td><td>1400
1100÷1230</td><td colspan="5" rowspan="5">Sum $l_t + l_n + l_w$</td><td rowspan="5">571</td></tr>
<tr><td>3</td><td>42</td><td>0</td></tr>
<tr><td>4</td><td>1525</td><td>1400</td></tr>
<tr><td>5</td><td>1647</td><td>1509</td></tr>
<tr><td>6</td><td>1423</td><td>1304</td></tr>
</table>

The following internal losses should be calculated to obtain the effective work:

- loss in cone and inlet chamber of the rotor according to formula

$$l_k = \frac{1}{2} \cdot \xi_k \cdot c_s^2 \tag{29}$$

- friction loss in rotor

$$l_t = \frac{1}{2} \cdot \xi_t \cdot w_{1a}^2 \tag{30}$$

- non-tangent inflow loss

$$l_n = \frac{1}{2} \cdot \xi_n \cdot w_{1b}^2 \tag{31}$$

- outlet loss for free space behind the rotor

$$l_w = \frac{1}{2} \cdot \xi_w \cdot c_3^2 \tag{32}$$

- volumtric loss [3]

$$l_v = \frac{\dot{V}_v}{\dot{V}_2} \cdot l_{ua} \tag{33}$$

- fanning loss [3]

$$l_b = \frac{K_b \cdot u_2^3 \cdot r_2^2}{2 \cdot \dot{V}} \tag{34}$$

Factor $\xi_k = 0.82$ is selected according to Standard [5].
Further formulas for loss calculations are as follows:

- speed

$$c_s = \frac{4 \cdot \dot{V}_s}{\pi \cdot d_s^2} = \frac{4 \cdot 0.442}{\pi \cdot d_s^2},$$

- factor of friction loss in the rotor

$$\xi_t = \lambda \cdot \frac{L}{d_h},$$

- Reynold's value

$$\mathrm{Re} = \frac{w_{1a} \cdot d_{h1}}{\upsilon},$$

- hydraulic diameter of the blade passage

$$d_{h1} = \frac{4 \cdot a_1 \cdot b_1}{2 \cdot (a_1 + b_1)} = \frac{4 \cdot (t_1 \cdot \sin\beta_1) \cdot b_1}{2 \cdot (t_1 \cdot \sin\beta_1 + b_1)},$$

- tangent component of relative speed

$$w_{1a} = \frac{\dot{V}}{\pi \cdot D_1 \cdot b_1 \cdot \chi_1 \cdot \sin\beta_1},$$

- non-tangent inflow factor is adopted according to tests [4] $\xi_n = 0.23$,
- outlet loss factor is adopted according to Tuliszka's tests [3] $\xi_w = 0.25$,
- speed c_3 is determined according to the formula

$$c_3 = \sqrt{c_{3m}^2 + c_{3u}^2},$$

where:

$$c_{3m} = \frac{\dot{V}}{\pi \cdot D_2 \cdot b_2},$$

$$c_{3u} = u_2 - c_{3m} \cdot \mathrm{ctg}\beta_2,$$

speed $w_{1b} = \frac{u_1}{c_{1n}} \cdot (c_1 - c_{1n})$.

Calculations have been made for capacity $\dot{V} = 0.442\ m^3/s$, results are presented in Table 6.

8. COMPARISON OF METHODS APPLIED TO DETERMINE WHIRL WORK

Comparison was made for one point of the characteristics corresponding to capacity $\dot{V}_s$ = 0.442 m^3/s, whirl works in proper sections of the machine and energy losses between these sections were calculated. The comparison was based on results obtained analytically and from balance and thermoanemometric experiments.

Balance measurement was made between sections 1 and 2. Thermoanemometric measurements were made in section 2. If we assume that angular momentum of the rotor inlet in section 1 equals zero, then component c_2u measured with a fibre sensor allows to calculate increase of energy between sections 1 and 2. Table 7 presents measurement sections of the machines, values of work and losses.

Arrangement of sections is presented in Figure 2 (part 1).

Strachowicz's formula gives the value of power reduction which is the closest to the real value obtained from the ratio of internal work l_i to l_{ut8}.

9. SUMMARY AND CONCLUSIONS

Proper identification of thermoanemometric and balance measurements allows to obtain the same values of effective work calculated for one point of flow characteristics. This conclusion results from the data listed in Table 7 line 7.

Comparison of both measurements allowed to determine losses between section "s-s" and "1-1", and arrive at the conclusion that tests aiming to determine the factor of total loss in cone and inlet chamber, volumetric loss included, should be carried out.

Table 7. Comparison of whirl work values obtained analytically (1), from balance measurement (2) and with thermoanemometer (3) for n = 2880 rev/min and $\dot{V}$ = 0,442 m^3/s

It.	Work and losses in machine sections	Unit	Results of		
			1	2	3
			Theoretical calculations	Energy balance	Thermo-anemometer
1	$l_{ut\infty}$	m^2/s^2	1863	–	–
2	l_{ut}		1509	–	–
3	l_{u12}		–	–	950
4	l_k		–	57	57
5	l_v		–	70	70
6	$l_{s1} = l_k + l_v$		–	127	127
7	l_{s2}		796	821	823
8	η		–	0.535	–
9	l_i		–	1534	1538
10	$l_t + l_n + l_w$		–	571	571
11	$l_b + l_v$		–	85	85
12	$\sum l_{3\text{-}8}$		713	713	713
13	μ_{rzecz}	–	–	0.82	0.82

Literatura

[1] Fortuna S.: *Zaprojektowanie odpowiednich wirników o zmiennej szerokości i ilości łopatek w celu badania strug wypływających do wolnej przestrzeni.* Kraków, 1995 (praca niepublikowana)

[2] Fortuna S., Gumuła S., Młodawski J., Prync-Skotniczny K.: *Prędkość przepływu i turbulencja w strudze powietrza wypływającego z koła łopatkowego wentylatora promieniowego.* XII Krajowa Konferencja Mechaniki Płynów, Zbiór Prac tom II, Rzeszów Jawor, 1996

[3] Fortuna S., Karpiel T.: *Badanie strat w elementach wentylatora.* Kraków, 2002 (praca niepublikowana)

[4] Fortuna S.: *Wyznaczanie pracy wirnikowej na podstawie wyników pomiarów termoanemometrycznych, cz. 1.* Prace Naukowe. Monografie. Konferencje, VI Międzynarodowa Konferencja „Wentylatory przemysłowe", Pol. Śląska w Gliwicach, 2003

[5] Fortuna S.: *Wyznaczanie pracy wirnikowej na podstawie wyników pomiarów termoanemometrycznych, cz. 2.* Prace Naukowe. Monografie. Konferencje, VI Międzynarodowa Konferencja „Wentylatory przemysłowe". Pol. Śląska w Gliwicach, 2003

[6] Fortuna S., Młodawski J.: *Eksperymentalna weryfikacja teoretycznej pracy wirnikowej.* IX Międzynarodowa Konferencja „Przepływowe maszyny wirnikowe", Pol. Rzeszowska, 2003

[7] Fortuna S.: *Calculation of rotor work on the basis of thermoanemometric measurement results, part 1.* Mechanics, vol. 24, No 1, 2005

[8] Fortuna S.: *Calculation of rotor work on the basis of thermoanemometric measurement results, part 2.* Mechanics, vol. 24, No 1, 2005

[9] PN-76/M-34034. *Zasady obliczania strat ciśnienia*

[10] Norma ISO 5801. *Industrial Fans. Performance Testing Using. Standarized Airways.* 1987 (E)