THE NEW METHOD OF REGULATION OF CENTRIFUGAL FAN

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SUMMARY

The purpose of the article was to present new method of regulation of centrifugal fan with use of variable blade length concept. The authors proposed analytical formulas to calculate fan pressure rise curves in this case. Then verification on real object has been conducted. A fan performance curves calculated with proposed formulas fit to curves measured on real object. This shows that proposed analytical formulas can be useful tool to calculate fan pressure curves when proposed regulation method is implemented.

INTRODUCTION

The regulation of fan is very important in any installations, where flow parameters are changed, due to changing industrial processes or installations. The general rule is to ensure that fan will be operated in whole range with sufficient efficiency. According to literature and standardization minimal sufficient efficiency is 60 % [1-3]. On the other hand growing demands regarding environmental protection and decreasing power consumption of fans is underlined by political documents, e.g. UE Regulations [4]. Definitely regulation of centrifugal fans seems to be more problematic that axial fans.

Within previous years many methods of fan regulation have been developed. Generally, regulation of fan can be realized by change of characteristic of installation or by change of fan characteristics. First method can be realized by adding bypass or by adding resistance to the system [1, 5-9]. Change of fan characteristics can be for example realized by following methods [1-2, 5-19]:

1. Regulation by change of the fan rotational speed
2. Regulation by change of the impeller geometry
3. Regulation by inlet or outlet vane control system
4. Regulation by series or parallel connection of fans
5. Regulation that is realized by combination of few above method

From the point of view of energy losses, the most efficient regulation method is change of rotational speed, then regulation with use of inlet vane control system. On the other side is regulation with use of damper that causes huge energy loses and nowadays it seems to be useful only in case of small fans [1, 3, 5-14]. Although regulation with use of frequency inverter seems to be most preferably, this method is expensive in case of big fans and there are problems with large weight forces. In such cases regulation methods with use of variable geometry fans seems to be good solution, but unfortunately difficult to implement, what is especially visible in case of centrifugal fans. Regulation by change of impeller width with use of special discs is easy but less efficient than regulation by change of outlet blade angle, that on the other hand is much difficult to implement [1, 3, 5-7, 12, 15-18]. These methods are still rarely implemented but worth of interest [15, 17].

Basing on well know methods of centrifugal fan regulations, new method of regulation can be proposed. This method was a subject of Polish Patent no. 234339 [20]. The authors proposed there method of regulation by change of blade length l (figure 1). This results in change of outside impeller diameter D_2 and outlet velocity triangle. As a result change of fan pressure rise occurs and fan performance curves are changing. Such regulation can be realized during impeller motion or during standstill. The main difference comparing to standard centrifugal fan impeller is partition of blade for two parts: fixed placed between flange and backplate and movable that can be shifted up and down in order to change blade length and impeller diameter. Because of that, both impeller flange and backplate need to be parallel. Such solution seems to be other concept of regulation system with variable geometry and should ensure wider operating grange with higher efficiency. The goal of the article is to present and verify analytical formulas to recalculate fan pressure curve in case of this regulation method.

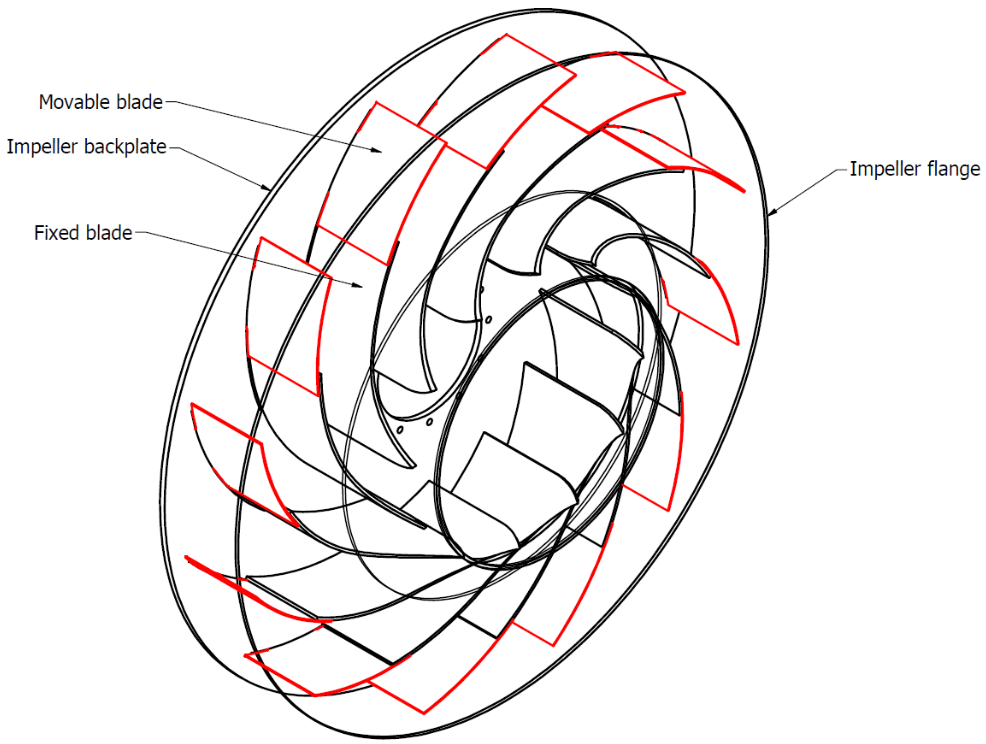

Figure 1: the concept of variable geometry impeller according to [20]

ANALYTICAL DESCRIPTION

From point of view of science as well as engineering demands, it is important to find formula that will be useful to calculate new performance curves for fan with proposed control system. New concept of centrifugal fan regulation will be described based on backward curved blades (figure 2). With such assumption blade length change causes change of impeller diameter D_2 and blade outlet angle β_2 (figure 2). Second assumption is that width of the impeller is constant. The symbol "M" in the following consideration denotes impeller with changed blade length.

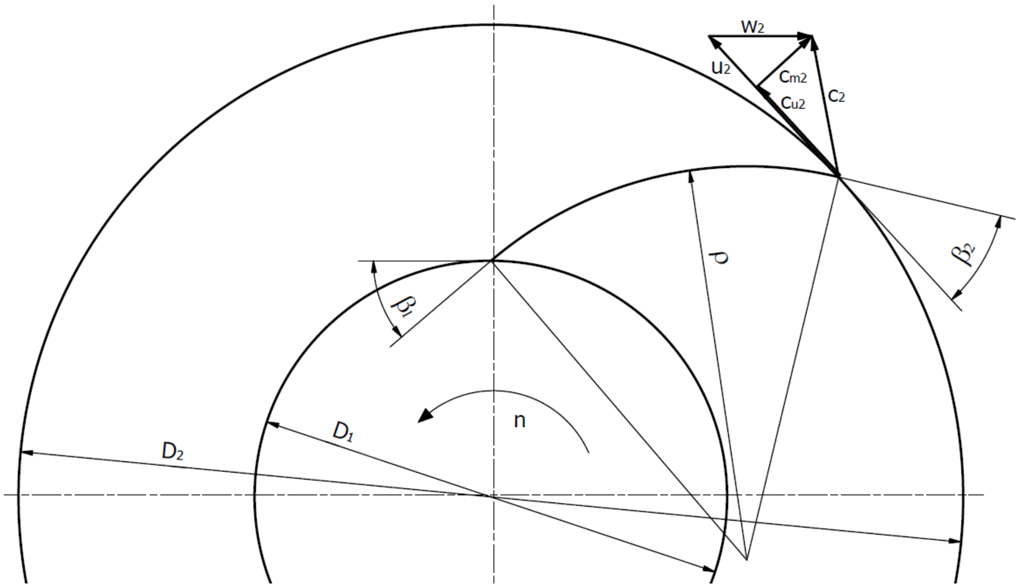

Figure 2: velocity triangle in case of backward curved blade (n - rotational speed, w_2 – relative fluid velocity, u_2 – tangential velocity, c_{u2} – peripheral component of fluid velocity, c_{m2} – radial component of fluid velocity, c_2 –absolute fluid velocity) [21]

Comparing theoretical pressure total rise formula in case of centrifugal fans it can be written [5, 8]:

$$\frac{\Delta P_{t,M}}{\Delta P_t} = \frac{\rho_{m,M} \cdot u_{2,M} \cdot c_{u2,M} \cdot \eta_{p,M} \cdot \mu_M}{\rho_m \cdot u_2 \cdot c_{u2} \cdot \eta_p \cdot \mu} \quad (1)$$

where:

ΔP_t – total pressure rise [Pa],

ρ_m – mean density of the gas [kg/m^3],

u_2 – tangential velocity at impeller outlet diameter [m/s],

c_{u2} – whirl velocity of the gas at impeller outlet diameter [m/s],

η_p – impeller hydraulic efficiency [-],

μ – slip factor [-].

Assuming that in both cases gas densities (ρ_m and $\rho_{m,M}$) and hydraulic efficiencies (η_p and $\eta_{p,M}$) are equal, it can be written:

$$\frac{\Delta P_{t,M}}{\Delta P_t} = \frac{u_{2,M} \cdot c_{u2,M} \cdot \mu_M}{u_2 \cdot c_{u2} \cdot \mu} \quad (2)$$

Science tangential velocity is:

$$u_2 = \pi \cdot D_2 \cdot n \quad (3)$$

where:

D_2 – impeller outlet diameter [m],

n – rotational speed of the fan [1/s].

Formula (2) can be written as:

$$\frac{\Delta P_{t,M}}{\Delta P_t} = \frac{\pi \cdot D_{2,M} \cdot n \cdot c_{u2,M} \cdot \mu_M}{\pi \cdot D_2 \cdot n \cdot c_{u2} \cdot \mu} \tag{4}$$

After simplification:

$$\frac{\Delta P_{t,M}}{\Delta P_t} = \frac{D_{2,M} \cdot c_{u2,M} \cdot \mu_M}{D_2 \cdot c_{u2} \cdot \mu} \tag{5}$$

Since peripheral component of fluid velocity of gas can be written as:

$$c_{u2} = u_2 - \frac{c_{m2}}{tg\beta_2} \tag{6}$$

Formula (5) can be rewritten:

$$\frac{\Delta P_{t,M}}{\Delta P_t} = \frac{D_{2,M} \cdot \left(\pi \cdot D_{2,M} \cdot n - \frac{c_{m2,M}}{tg\beta_{2,M}}\right) \cdot \mu_M}{D_2 \cdot \left(\pi \cdot D_2 \cdot n - \frac{c_{m2}}{tg\beta_2}\right) \cdot \mu} \tag{7}$$

Where radial velocity of gas is given by formula:

$$c_{m2} = \frac{V}{\pi \cdot D_2 \cdot b_2} \tag{8}$$

where:

V – volumetric flow [m^3/s],

b_2 – blade width at impeller outlet [m].

Substituting (8) into (7) and simplifying, general formula for pressure rise in case of impeller with changed diameter D_2, can be written as:

$$\Delta P_{t,M} = \Delta P_t \frac{\left(\pi^2 \cdot D_{2,M}^2 \cdot n - \frac{V_M}{b_{2,M} \cdot tg\beta_{2,M}}\right) \cdot \mu_M}{\left(\pi^2 \cdot D_2^2 \cdot n - \frac{V}{b_2 \cdot tg\beta_2}\right) \cdot \mu} \tag{9}$$

Thanks to above formula there is a possibility to calculate change of the pressure rise in the fan when impeller outlet diameter is changed. The only unknown value is volumetric flow rate V_M after change of D_2. Change of D_2 causes change of the velocity triangle at trailing edge of the blade. Assuming that velocity triangles in case of both impellers are similar, the ratio $c_{m2,M}/c_{m2}$ can be written as:

$$\frac{c_{m2,M}}{c_{m2}} = \frac{tan\beta_{2,M} \cdot (u_{2,M} - c_{u2,M})}{tan\beta_2 \cdot (u_2 - c_{u2})} \tag{10}$$

Substituting (3) and (8), then assuming that $c_{u2,M}/c_{u2} = u_{2,M}/u_2 = D_{2,M}/D_2$ and rewriting, formula for V_M calculation can be written as:

$$V_M = V \cdot \left(\frac{D_{2,M}}{D_2}\right)^2 \cdot \frac{tan\beta_{2,M}}{tan\beta_2} \tag{11}$$

From (11) it is well visible, that flow rate of fan with proposed regulation system is changed with square of ratio of both diameters and ratio of trailing edge angle. Both formulas (9) and (11) give an analytical tool to recalculate performance curves of fan with regulation method by change of impeller blade length. Application of these formulas will be verified by measurements on the real object.

VERIFICATION ON THE REAL OBJECT

In order to verify formulas (9) and (11) to calculate characteristic of the centrifugal fan when change of blade length and diameter D_2 occurs, test on the real object have been conducted. For this purpose fan type TES14-200 have been chosen. Basic data of this fan is presented in table 1. Basic impeller size is designated as LR2. The second impeller has larger outlet diameter D_2 by 10 %, and third impeller has this diameter smaller by 10 %. Each impeller is showed on figure 3. Measurements has been carried out according to Standardization DIN 24163 with inlet nozzle to measure volumetric flow rate and pressure rise measurements on the outlet duct [22]. Prepared test stand is showed on figure 4. Each impeller has been measured with the same scroll size. This was scroll designed for basic impeller (LR2). As a result fan performance curves in case of three different impellers have been measured (figure 5). With use of formulas (9) and (11) data from basic impeller size has been used to calculate fan pressure curve in case of larger and smaller impeller (L3 and LR4 respectively). Both curves are showed on figure 5. Shaft power and efficiency curves shows that small increase of outlet diameter can result in lower energy consumption but decrease gives the efficiency drop. It is result of not only change of diameter but also trailing edge angle. When D_2 is smaller, angle β_2 increases in case of backward curved blade and in such case it is normal that impeller hydraulic efficiency decreases. On the other hand larger diameter D_2 results in decrease of β_2 angle what can cause better hydraulic efficiency.

Table 1: geometrical data and slip factor regarding different size of impellers

<table>
<tr><th>Parameter</th><th>Description</th><th>LR2</th><th>LR3</th><th>LR4</th></tr>
<tr><td>D_1 [mm]</td><td>blade inlet diameter</td><td colspan="3">172</td></tr>
<tr><td>D_2 [mm]</td><td>blade outlet diameter</td><td>362</td><td>398,2</td><td>325,8</td></tr>
<tr><td>l [mm]</td><td>blade length</td><td>185</td><td>224,7</td><td>149,4</td></tr>
<tr><td>β_1 [°]</td><td>blade angle of attack</td><td colspan="3">23</td></tr>
<tr><td>β_2 [°]</td><td>trailing edge angle</td><td>29</td><td>25,45</td><td>31,51</td></tr>
<tr><td>b_1 [mm]</td><td>blade width at the inlet diameter</td><td colspan="3">75</td></tr>
<tr><td>b_2 [mm]</td><td>blade width at the outlet diameter</td><td colspan="3">55</td></tr>
<tr><td>Z [-]</td><td>number of blades</td><td colspan="3">12</td></tr>
<tr><td>μ [-]</td><td>slip factor</td><td>0,804</td><td>0,819</td><td>0,787</td></tr>
</table>

Figure 3: three impellers with different blade length and outlet diameter (from left side: LR3, LR2 and LR4 impellers)

Figure 4: test stand for fan TES14-200

CONCLUSIONS

In the article new method of regulation of centrifugal fan has been presented. In the next chapter, the authors proposed analytical method to calculate fan performance curves when change of blade length and impeller outlet diameter occurs. Tests on real object showed that formulas (9) and (11) can be used to calculate fan pressure rise. When both formulas are applied, the fan pressure rise curve lay below measured curve when D_2 diameter is increased and above real curve when diameter is decreased. That is well visible on figure 5. The differences are caused by:

1) The fact that formulas to calculate pressure rise (9) and volumetric flow rate (11) are based on assumption that velocity triangle are similar in both cases.
2) These theoretical velocity triangles are based on geometrical and flow data of basic fan. In fact, trailing edge angle differs from actual stream angle at trailing edge. This deflection is not included in presented formulas and when D_2 diameter is changed, actual stream angle could be changed in another way than assumed proportion.
3) Proposed formulas are based on consideration of flow through a impeller in planar middle section. Actually flow pattern in this case is spatial in nature what causes that velocity triangles changes on blade width.
4) Each of the impellers are working in the same volute casing which was designed only for basic impeller. Operation of the impellers with other geometry is affected by volute geometry, what also means that velocity triangle at trailing edge differs from assumed.

These facts are not easy to evaluate with use of analytical methods and would make formulas much more complicated. As showed results, differences are within margin of few percent. Since tested impellers were built with some constructional tolerances, differences between recalculated and measured values are within range from 2.5 % to 5 %. It fits to fan manufacturing accuracy category 1 or 2 [5, 8]. Generally proposed formulas are simple to use and suitable to calculate pressure rise curve in considered regulation method.

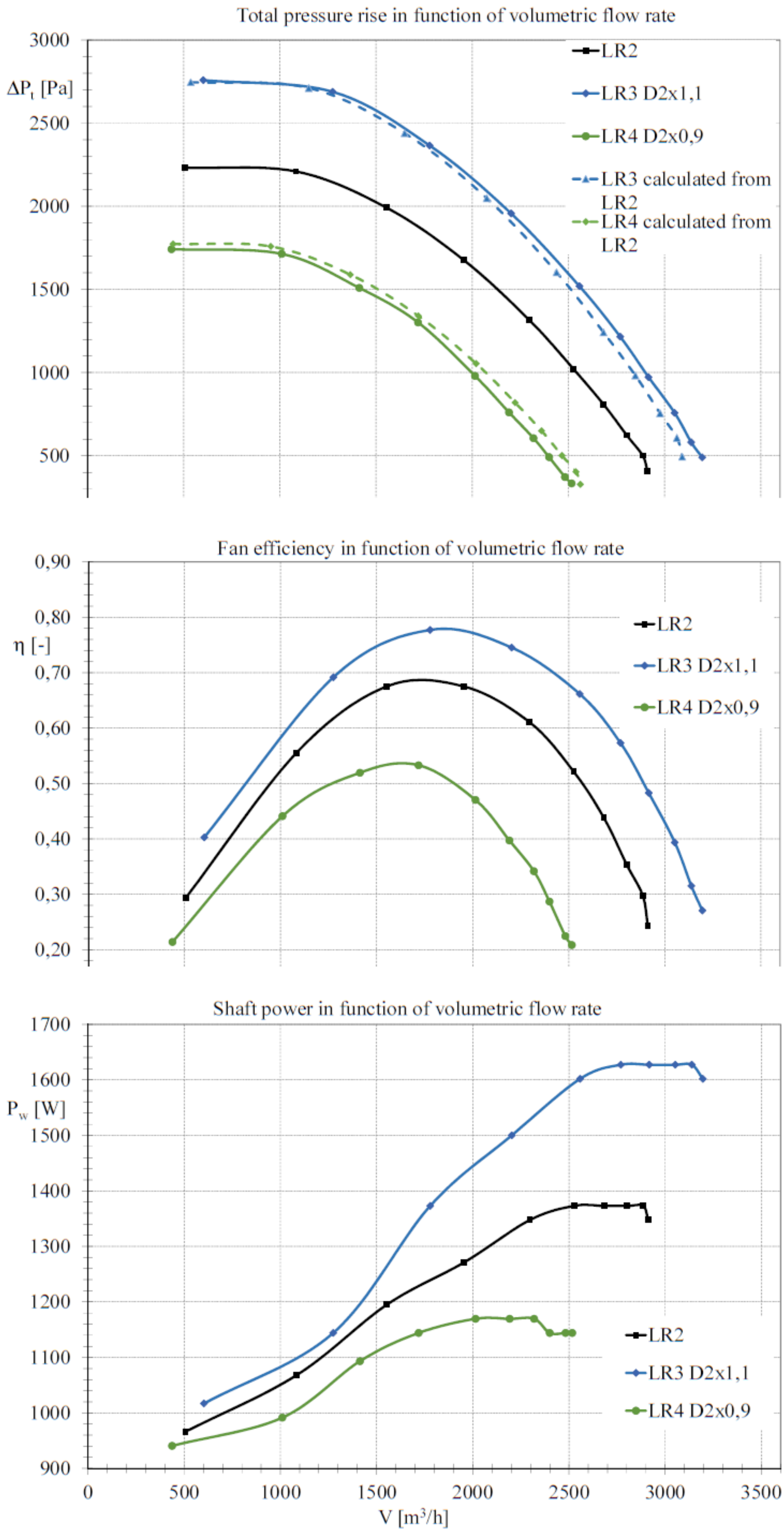

Figure 5: fan performance curves in case of three different impeller sizes: LR2, LR3 and LR4

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