Variable geometry centrifugal fan - challenges in identification and control of dynamics

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Abstract
The paper presents the investigations of the dynamic characteristics as well as the operational dynamic performance of the centrifugal fan with dynamic adjustment of the outer diameter of the impeller. This is new solution that has never been used in centrifugal fans. As a result, it is possible to increase the range of high-efficiency fan operation by about 40% compared to existing solutions without such feature. The scientific basis of this idea is that a relatively small change in the diameter of the fan impeller significantly affects the flow rate and pressure rise of the fan. Therefore, the idea uses the variable length of the rotor blades, dividing them into a fixed part and a moving part. The moving part of the blade is controlled by a special mechanical system that has the ability to change the position of the movable blade during the fan operation. This causes changes in mass distribution, geometry, stiffness, and operational load / flow that can generate dynamic phenomena during operation, which are investigated in the paper.

1 Introduction

The dynamics of rotary machines is a significant design and operational challenge. In the case of axial and radial fans, dynamic phenomena are responsible for their damage, manifested mainly in the form of fatigue cracks. This type of damage is difficult to detect and can lead to serious failures, resulting in high repair costs and downtime [1]. Examples of such phenomena are shown in Fig. 1.

Industrial axial and radial fans are found in every area of industry, ranging from small units with a power of few kW, to large machines with a power of several MW. They are usually the core equipment that has to be reliable. In addition, they consume significant amounts of energy, which translates into operating costs. These machines consume approximately 300 TWh of electricity per year, which rank them as third largest group in terms of energy consumption among all industrial devices in EU [2]. In 2013, the EU issued a directive on minimum requirements for electrically driven fans, which has been in force since 2013 [3]. As a result of this, the saving of drive energy and improvement of efficiency have become primary drivers for this technology. Therefore, the market expects further development of existing fans in terms of efficiency [4]. The key parameter of an axial or radial fan is efficiency , which should be maintained at a high level in terms of adjusting the flow parameters of the fan.

Currently there are two main ways of centrifugal fan regulation [5]. First way is by change of the characteristic of the installations. The second way is by change of the fan characteristic. This work is focused only on the second group. Within this scope, there are few regulation methods of centrifugal fan well described in literature. These methods are [5]:

- Fan regulation by inlet or outlet vane control system
- Fan regulation by change of the fan rotational speed
- Fan regulation by change of the impeller geometry
- Fan regulation by series or parallel connection of fans
- Fan regulation that is realized by combination of few above method

Figure 1. Fatigue failures of industrial fans

The most popular is regulation by change of the fan rotational speed. Currently in most cases it is executed with use of frequency inverter. This method is efficient and easy to implement but requires additional electrical equipment that is responsible for fan regulation [5]. The disadvantage of this method is that frequency inverters for fans with large power consumption above 750 kW are custom special order what causes high investment costs. The second problem regards to large inertia forces in case of such fans when rotational speed is changed. These forces causes problem not only onto fan structure and strength but also on electrical motor of the fan. In these cases Radial Vane Inlet Control (RVIC) systems are implemented. Unfortunately efficiency of such regulation is lower than in case of rotational speed change [6]. Regulation of centrifugal fan by change of the impeller geometry is rarely used [8, 13-15].

The authors of the paper dealt with the development of a new method of regulating the flow parameters of centrifugal fans. This method is focused on increasing the range of effective fan operation by changing length of the impeller blade, which results in outer diameter change of the impeller. The theory of centrifugal fan design describes the relationship between outer diameter of the fan impeller and flow rate as well as pressure rise. These equations are presented below:

$$V_2 = V_1(D_2/D_1)^3 \tag{1}$$

$$\Delta p_2 = \Delta p_1(D_2/D_1)^2 \tag{2}$$

where:

V1—flow rate of fan with impeller of D1 outer diameter
V2—flow rate of fan with impeller of D2 outer diameter
Δp1—pressure rise of the fan with impeller of D1 outer diameter
Δp2—pressure rise of the fan with impeller of D2 outer diameter

There is third-order relationship between the fan flow rate and the impeller outer diameter (1) as well as the second-order relationship between the pressure rise and the impeller external diameter (2). This means that relatively small change in the impeller diameter significantly affects the flow rate and the pressure rise of the fan. The use of variable length impeller blades to control the flow parameters of centrifugal fans is the novel method of extending flow parameters of centrifugal fans. For this purpose, the impeller blade is split into a fixed and a movable/adjustable blade (Figure 2). More details about this solution is presented in the next chapter.

However, the developed solution requires a special approach to the assessment of dynamic phenomena arising during operation. In typical centrifugal fans with constant blade geometry, the basic factor influencing the minimization of operating vibrations is the correct static and dynamic balancing. In addition, at the stage of designing this type of rotary machines, dynamic analyzes are carried out in the field of identifying the shape and frequency of natural vibrations in order to limit the possibility of fan operation in the resonance area. These areas are most often the range of induced vibrations resulting from the rotation of the fan rotational unit. The basic excitation is the rotational frequency of the impeller, which can be determined using the following formula [5, 10, 21].

$$\mathrm{f}_{RF} = \frac{\omega}{2\pi} = \frac{n}{60}\,[Hz] \tag{3}$$

The second characteristic frequency is the blade frequency, which is determined by the product of the number of blades and the rotational frequency [22]:

$$\mathrm{f}_{BPF} = N\frac{\omega}{2\pi} = N\frac{n}{60}\,[Hz] \tag{4}$$

In the case of a fan with variable blade geometry, the assessment of its dynamic properties becomes more complicated. This is due to the change in mass, inertia, geometric and stiffness parameters with each change in the position of the moving parts of the impeller blades. Additionally, the blade motion control mechanism is also an area where resonant vibration may occur. For this reason, dynamic phenomena occurring in the operation of the presented new type of fan requires special attention. The research on these phenomena is presented below and a method of their assessment is proposed, which ensures safe operation.

2 Solution description

On the basics of current knowledge and practical experience the new method of centrifugal fan regulation has been proposed. This method was developed with cooperation of the industry and scientists from Wroclaw University of Science and Technology. It is a subject of Polish Patent no. 234339 [18]. The main idea of this solution is to change fan characteristic by means of variable blade length. Decrease or increase of the blade length causes decrease or increase of the impeller outlet diameter. The velocity triangle on trailing edge is decisive regarding centrifugal fan total pressure rise. In order to realize this regulation method impeller of the centrifugal fan consists of the impeller shroud and impeller backplate. Between them there is fixed blade shorter than basis because second part of blade is movable blade (Figure 2). When this movable blade is shifted downward, impeller outlet diameter decreases. When movable blade is shifted upward, this diameter increase. Such regulation can be realized during impeller motion as well as during standstill.

The method of centrifugal fan regulation during impeller motion was a subject of CORNET project titled “Efficient Fan Blade Technology for Industrial Centrifugal Ventilation Systems” realized in cooperation with Fraunhofer Institute for Machine Tools and Forming Technology, Technische Universität Chemnitz and industry form Poland and Germany. As a result of the project was the prototype of the fan according to new idea. The model and prototype with the mechanism that is responsible for blade translation during impeller rotation is shown on Figure 3. This prototype was a centrifugal fan with basic (middle) impeller outlet diameter equal to 810 mm and regulation of this diameter was within range ± 10 %.

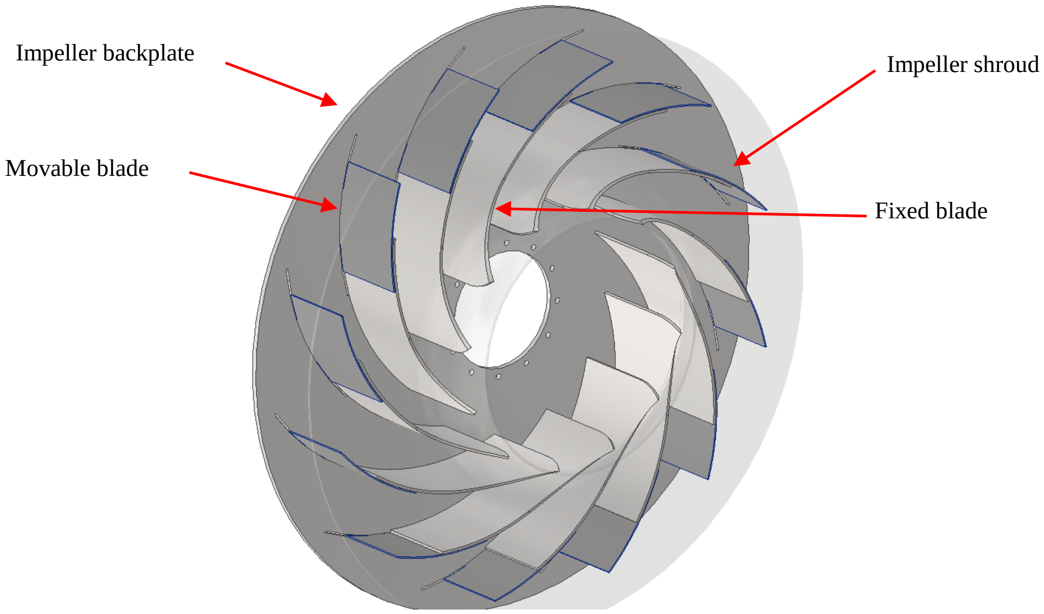
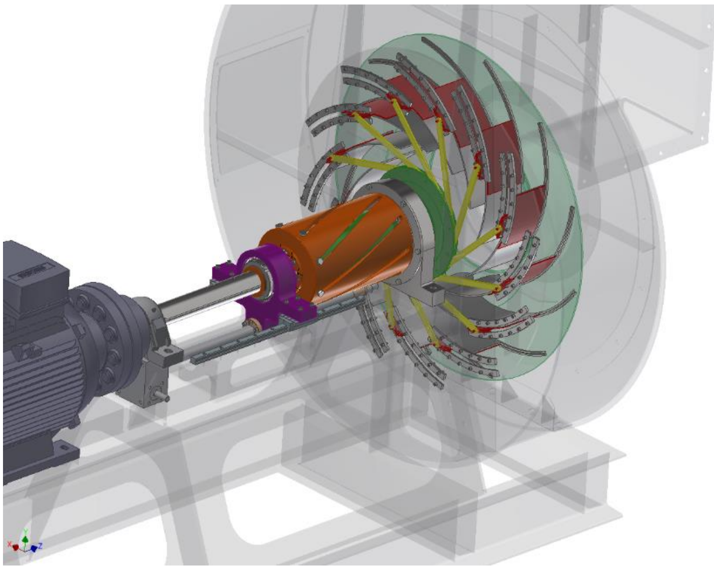


Figure 2. The idea of the impeller with movable blade



a)

b)

Figure 3. 3-D model of the fan with new regulation method (a) and prototype with such solution (b)

Evaluation of the fan performance curves were conducted on standardized airways test stand build in accordance with EN ISO 5801 [19]. This test stand is showed on Figure 4. Measurements were realized in few position of the blade, in range from minimal to maximal. Results are shown on Figure 5. There are total pressure rise curves with minimal (0%), middle (50%) and maximal (100%) blade length in function of flow rate. Additionally, areas of minimal efficiency equal to 60 % and 70 % are shown. All results are shown in case of nominal rotational speed of the impeller that was assumed in the realized project. That was speed equal to 1500 rpms.

Figure 4. Test stand to test the prototype of the centrifugal fan in accordance with standard EN ISO 5801

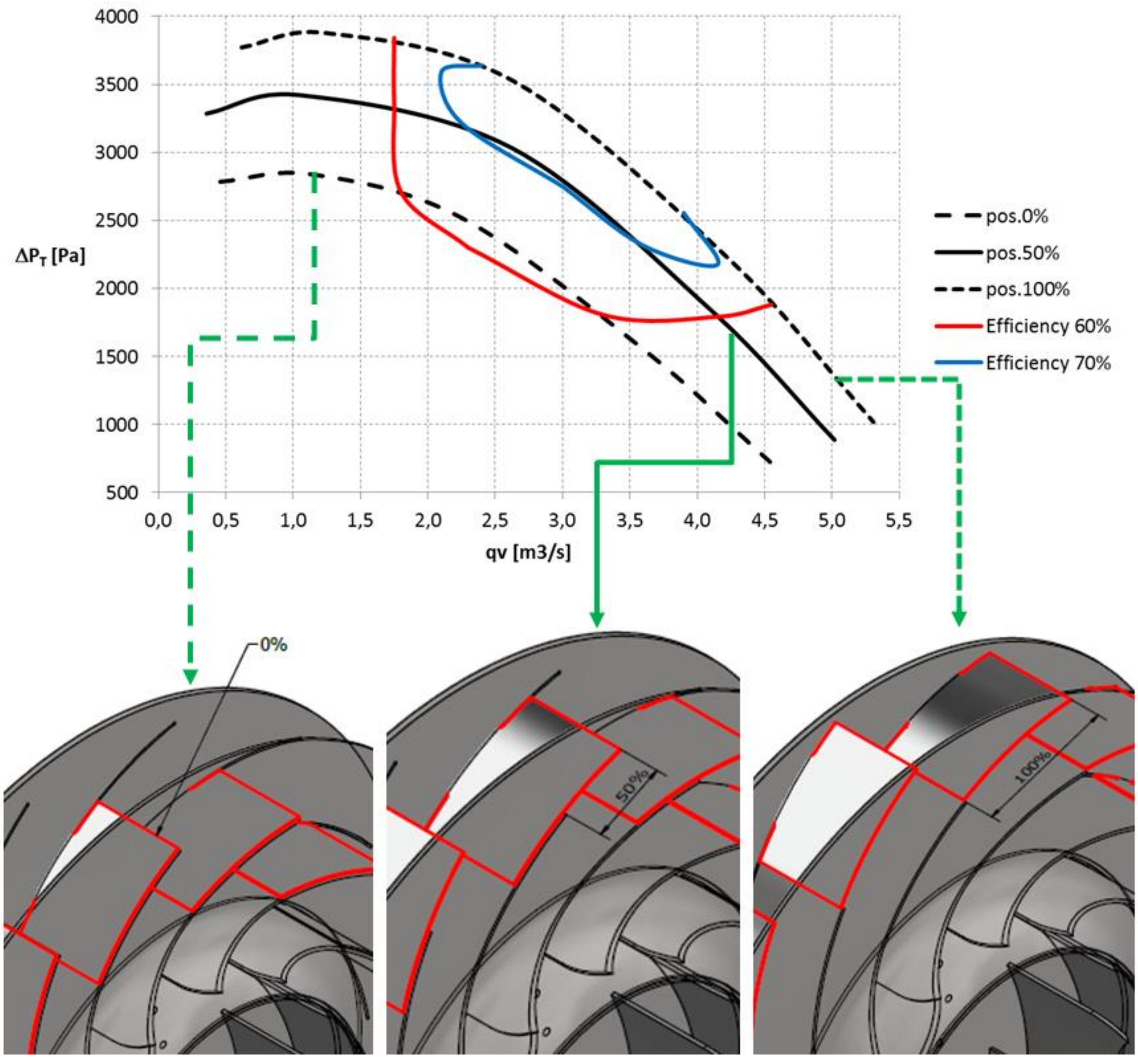


Figure 5. Prototype fan performance curves (Total pressure rise in function of volumetric flow rate) in case of extreme and middle positions of the movable blade

Received results shows that proposed solution could be a great alternative for fan for which operation with frequency inverter is costly or problematic due to high internal forces. These are especially larger centrifugal fans with power above 500 kW. The range of the fan operation with minimal efficiency above 60 % [6, 7], that is value recognized as minimal cost-effective value is widen. The pressure can be changed within this range by about 50 % and flow rate by about 60 %. In this solution there is no need to use frequency inverter and whole regulation is realized mechanically. Even if steering mechanism is complicated and raises the price of the fan, the costs are still below implementation of frequency inverter in case of larger fans. What is more, in systems where changes of working point occur in longer period (e.g. ventilation of underground mine), such regulation method can be implemented without steering mechanism. Then regulation of the blades is carried out during fan standstill. Such solution is cheap and easy to implement.

3 Experiment description

For the purpose of verifying the developed solution, described in the previous chapter, field tests were carried out. Its aim was to assess the dynamic properties during the operation of the new centrifugal fan. The prototype, as described in the previous chapter, is characterized by having a variable impeller geometry, for this reason the assessment of dynamics requires testing in different positions of the mechanism regulating the geometry of the blade. In contrast to solutions without the possibility of changing the impeller geometry, measurements at several operating points are not enough. In the described case, they also need to be duplicated for different positions of the mechanism regulating the position of the impeller blades. For the purposes of the article, the fan was tested for 3 blade positions (maximum and minimum blade extension and in the middle position). The photo below shows the described fan on the test stand. Its most important elements are marked with numbers on it. Number 1 - electric motor, 2 – shaft with mechanism to change the impeller geometry (it is located under the yellow safety cover) connecting the motor and the centrifugal fan, 3 – centrifugal fan, 4 - test bench frame and 5 inlet nozzle.

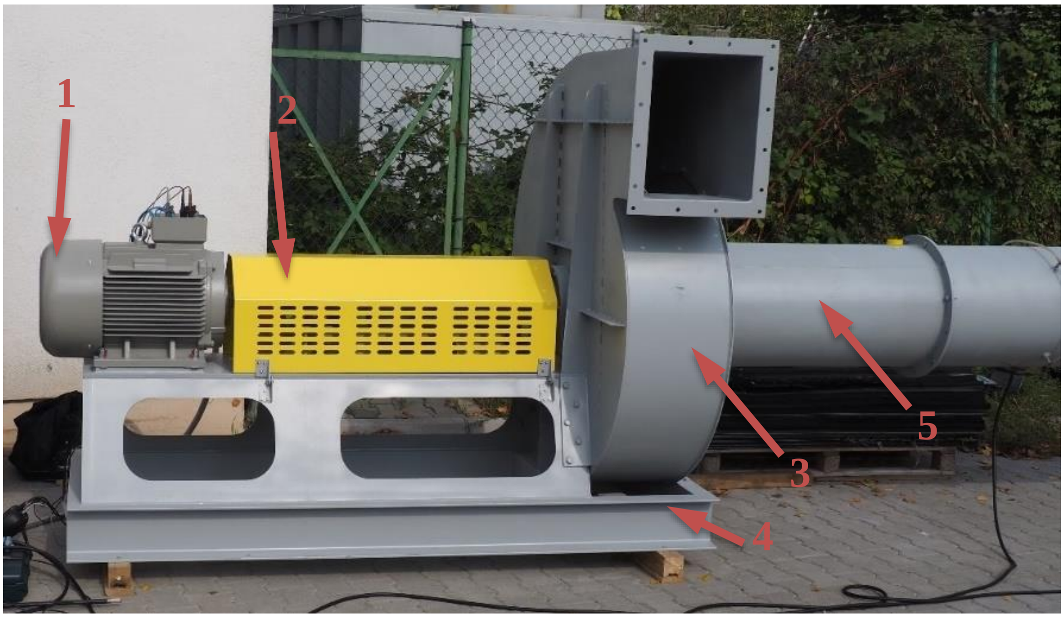


Figure 6. Field test stand for centrifugal fan

During the fan tests, it must also be possible to reproduce the situation when the conditions prevailing in the gas transport installation change. For this purpose, at the beginning of the inlet nozzle, a place is provided for mounting the special plates to change the free flow area. Their purpose is to increase the resistance of gas flow in the network, as a result of throttling effect. Thanks to this, it is also possible to determine the operating parameters of the fan in different operating conditions. The figure below shows the beginning of the inlet nozzle to the fan along with the special plates.

Figure 7. Start of the inlet nozzle for the fan together with the special test

The next stage of the research was the selection of measuring points. Their arrangement is shown in the next figure. Three different types of measuring equipment were used to identify the dynamic properties of the innovative centrifugal fan with variable blade geometry. The first was the optical tachometer marked in the figure as C1. It measured the actual rotational speed of the fan. Another apparatus used were piezoelectric single-axis accelerometers, marked with numbers from C2 to C6. The arrows in the drawing indicate the direction of measurement acceleration.

In addition to the previously rearranged points of the measuring sensor installation, there are also points on the fan characteristics at which the measurement will be made. They are shown in the charts below. Vibration measurements of the fan has been realized in three different position of the movable blade. That were extreme positions (minimal and maximal) and middle position. During whole test working point of the fan in case of each position has not been changed. It means that the flow reached maximal values when total pressure rise is minimal. Basically rotational speed in each case was nominal value equal to 1500 rpms. Working point of the fan during these measurements are shown on Figure 9.

In conclusion, the plan of the experiment can be divided into the following steps:

- Installation of the tested fan on the test stand.
- Selection of special plates to change the free flow area to simulate the resistance of gas flow in the installation.
- Choice of techniques and mounting points for sensors.
- Determination of the points on the fan characteristics where the measurement will be made. In the case of variable geometry fans, this step should be repeated for each analyzed position of the control mechanism.
- Measurement for different mechanism positions and conditions of the ventilation network.

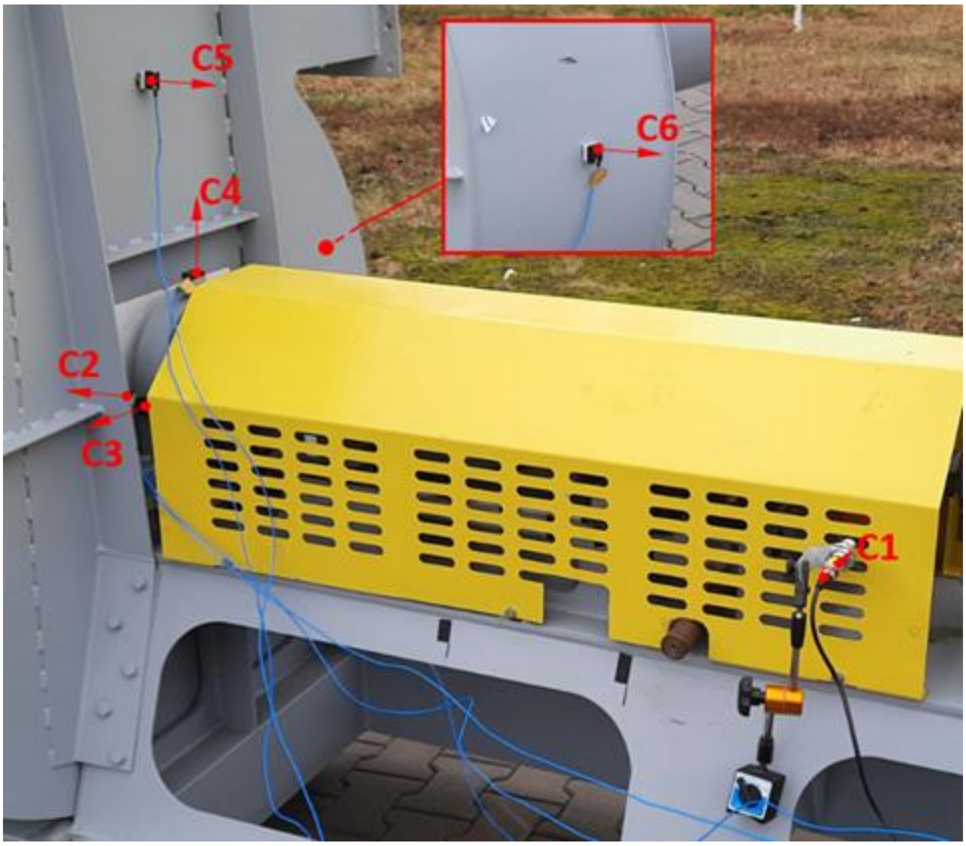


Figure 8. Location of measuring points (without acoustic camera)

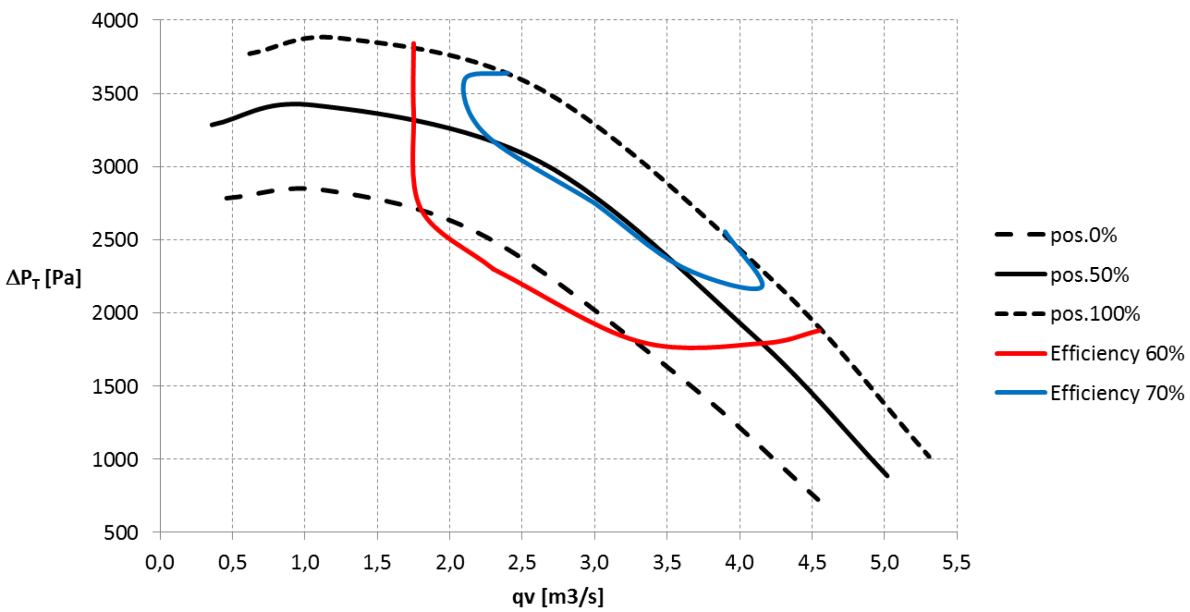


Figure 9. Working point (1, 2 and 3) for which vibration measurements were conducted shown on the fan performance curves in case of minimal (point 1), middle (point 2) and maximal (point 3) positions of the movable blade

4 Results and discussion

The measurements of the vibrations was taken, as described above, for the three positions of the blades. In Figure 10 the resultant spectra of the accelerations on the bearings are presented. The ranges of rotational frequency (RF) as well as blade passing frequency (BPF) are marked on the diagram. One can observe that the amplitude of the characteristic frequencies (RF, BPF) rises while the blade is moved to its maximum extension (100%).

Similar analysis was done for the fan housing (Figure 11). For the first look a great difference in the amplitude value, with respect to any other channels, is visible. The channel 5 was a measuring point on the fan housing side wall, which has relatively low stiffness in the direction of measurements but great sensitivity for pressure pulsations. The observed high values at channel 5 are local vibrations derived by the pulsation of the pressure inside the housing and some of normal modes. In the case of housing the biggest variation related to the blade position is 100Hz which is one of the higher harmonics of the RF.

Additional analysis was done to the vibration energy in the vide frequency band. (Figure 12, Figure 13). The RMS value of the vibration do not specify precisely any trend of the vibrations level correlated with the blade position. However, one can observe increase in the RMS on the fan housing correlated with the blade extension.

The increase of vibration level in correlated with the RF and BPS corresponds to the results of numerical modal analysis (Table 1). The simulations indicated that with the extension of the blade more modes “falls” into the range of 0,85 to 1,15 of the characteristic frequency.

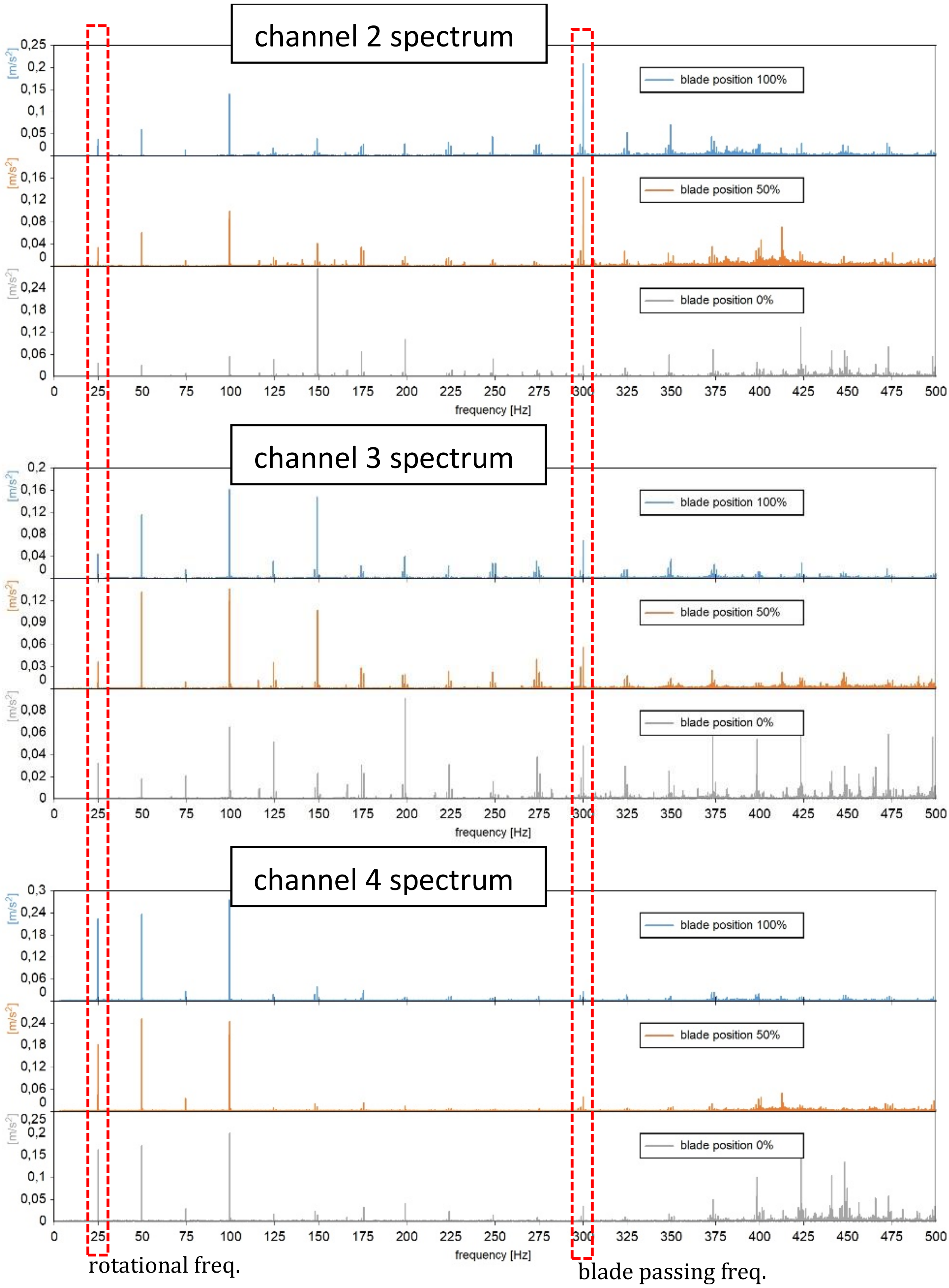


Figure 10. Vibrations spectra of the fan bearings

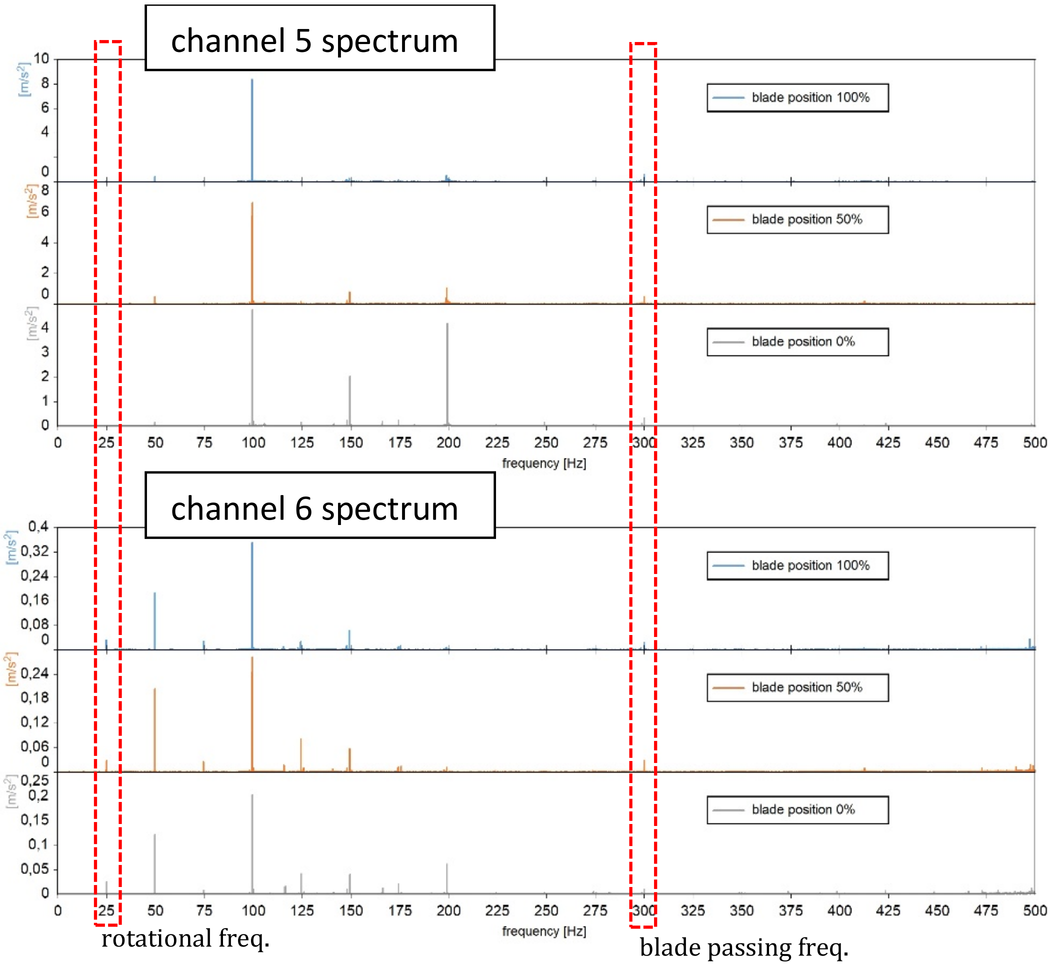


Figure 11. Vibrations spectra of the fan housing

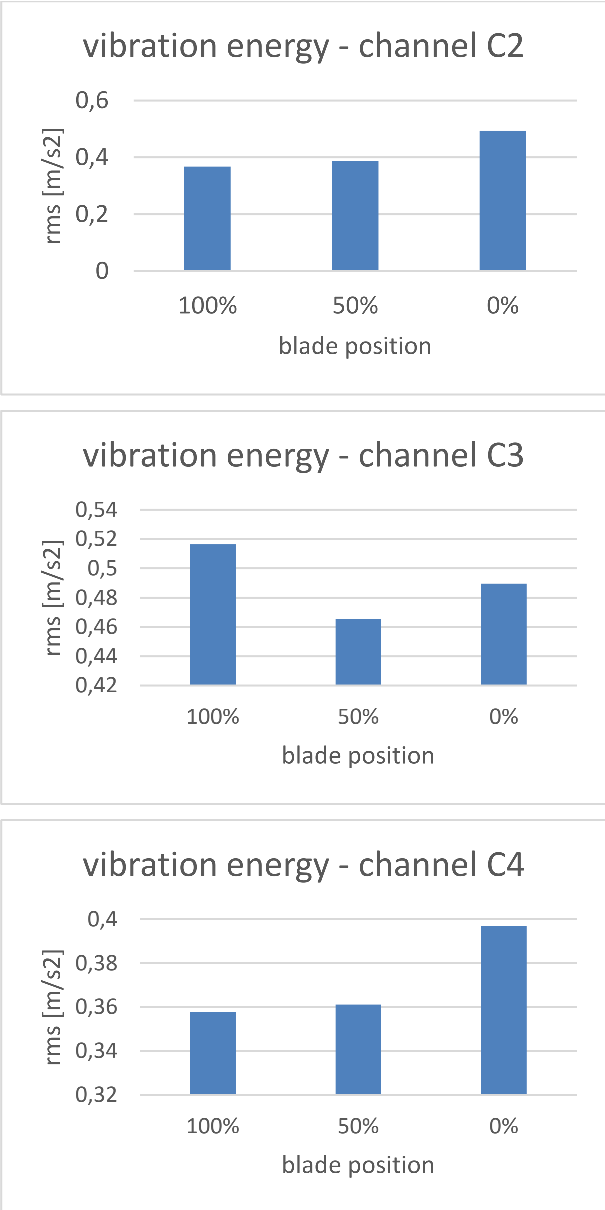


Figure 12. Vibration energy change - fan bearings

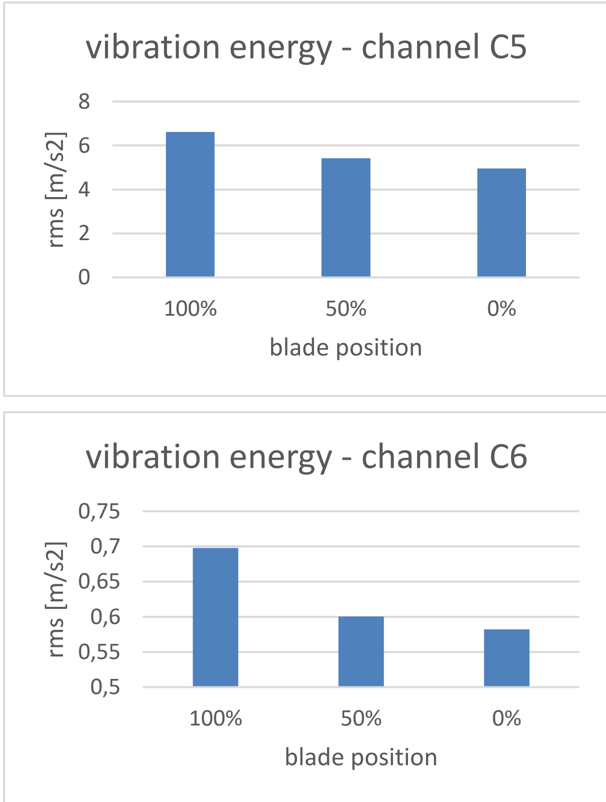


Figure 13. Vibration energy change - fan housing

Table 1: Numerical modal frequencies of the impeller

	Frequency [Hz]			
Mode	blade 0%	blade 100%	characteristic frequency	
1	12	-		
2	12	-		
3	14	-	21,25	0,85 RF
4	41	30	25	RF
5	43	41	28,75	1,15 RF
6	136	41		
7	214	80		
8	215	262		
9	238	262		
10	250	268		
11	255	268		
12	257	289	255	0,85 BPF
13	267	301	300	BPF
14	268	302	345	1,15 BPF
15	279	306		
16	289	309		
17	294	309		
18	294	310		
19	297	310		
20	305	312		

5 Conclusions

In the paper an unique centrifugal fan with dynamic adjustment of the outer diameter of the impeller is presented. The scientific basis of this idea is that a relatively small change in the diameter of the fan impeller significantly affects the flow rate and pressure increase of the fan. Therefore, the idea uses the variable length of the rotor blades, dividing them into a fixed part and a moving part. The moving part of the blade is controlled by a special mechanical system that allows it to slide over the fixed part of the blade. Significant in this solution is the ability to change the position of the movable blade during the fan operation. This technical solution is aimed to widen the range of high-efficiency fan operation, which will translate into a reduction in energy consumption, increase of production efficiency and as a result reduction of the carbon footprint. The use of impeller geometry changes for this purpose eliminates significant investment costs that currently have to be incurred in order to obtain regulatory options. The described solutions allow to replace the expensive frequency inverter with a cheaper mechanism, which additionally increases the effective range of the machine's operation. As a result, it is possible to increase the range of high-efficiency fan operation by about 40% compared to classic solutions without such feature. This result was confirmed by numerical CFD simulations and tests on the prototype. On the other hand such new solution rises a challenge from the point of the operation, maintenance and dynamics control point of view. It is also a major design challenge due to the relatively complex kinematics of the blade movement and the requirement to maintain the symmetry condition of the displacement of all blades and the associated actuators under high dynamic loads caused by centrifugal forces and medium flow. The presented measurements results of the fan with adjustable fan blade indicates that the dynamics of the object require additional attention. Due to the changes in the impeller geometry the mass distribution, inertia and stiffness are changed. As a result the natural frequencies of the system shifts and changes in the operational performance may be observable. In the presented example, the operation with full extension of the blade caused increase of the vibration levels of the RF and BPF bands which may be caused by shifting some of the normal modes frequencies closed to the characteristic frequencies (RF, BPF). Moreover, the extension of the blade correlate with the housing vibrations which is caused by flow/pressure rise and thus stronger pressure pulsations. Based on the conducted investigations it can be concluded that changes in dynamic characteristics, caused by a change in the position of the rotating elements of the new fan flow parameters adjustment system, must be taken into account at the design stage of this new type of fan.

Acknowledgements

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