Proc. of the 30th International Slovak–Polish Scientific Conference on Machine Modelling and Simulations 2025

New Method of Adjusting Flow Parameters of Axial Fans

P. Odyjas[a,*], I. Kuric[b], M. Macko[c], D. Pietrusiak[a], M. Saga[b], A. Szczepańczyk[c], Z. Szczepański[c], M. Śliwiński[c], K. Urbański[d], M. Vlasko[b], J. Więckowski[a] and P. Moczko[a]

[a] *Wroclaw University of Science and Technology, Faculty of Mechanical Engineering, Łukasiewicza 7/9, 50-371 Wrocław, Poland*
[b] *University of Žilina, Faculty of Mechanical Engineering, Univerzitná 8215/1, 010 26 Žilina, Slovakia*
[c] *Kazimierz Wielki University, Faculty of Mechatronics, Kopernika 1, 85-074 Bydgoszcz, Poland*
[d] *ELMINE sp. z o.o., Wierzbowa 7, 67-200 Głogów, Poland*

Doi: 10.12693/APhysPolA.149.S81 *e-mail: piotr.odyjas@pwr.edu.pl

The paper presents numerical investigations of a new method of regulation of an axial fan. This method is based on dividing the impeller blade and stator guide vane into two parts, i.e., fixed and movable. The proposed new method was compared with a reference fan, in which the system with whole movable blade regulation is applied. Flow simulations aimed at the evaluation of the regulation characteristic of both solutions. The results showed that the new regulation method has a wider regulation control parameter, with values 10.2% and 12.2% higher for flow and pressure, respectively, than those of the reference fan. The drawback of the new idea is the restricted regulation range in the case of higher flows and pressures, which results from growing fluid flow losses. The obtained results suggest that the optimal regulation method would be a combination of the two approaches considered in the project. It means that an impeller with whole regulation blade and a stator guide vane with movable front part should have better performance.

topics: axial fan, fan regulation method, numerical methods, computational fluid dynamics (CFD) simulation

1. Introduction

Nowaday, energy consumption and costs are very important issue in modern communities. Growing energy prices and demands, in conjuction with environmental requirements, result in the need of constant development of the most energy-consuming machinery. For instance, fans are the third largest group of industrial machinery responsible for energy consumption in the European Union (EU). The general electricity consumed by fans was 230 TWh annually in 2010, 300 TWh/a in 2020 — and the amount is still growing, with an expected value of about 345 TWh/a in 2030. Within this group, axial fans with a pressure above 300 Pa are responsible for electricity consumption of about 80 TWh/a. In EU, there are 4 billion fans, of which 214 million (about 5%) are industrial fans with power between 125 and 500 kW and are responsible for about 80% of the total fan electricity consumption [1]. This fact has led to the implementation of polices aimed at enhancing fan efficiency. Such minimal requirements are given, for example, in EU regulations like Commission Regulation (EU) No. 327/2011 and Commission Regulation (EU) 2024/1834 [2, 3].

The above facts, in combination with growing energy prices, are the main reasons for users of fans to want these devices to be more efficient. As a consequence, the development of new fan regulation methods is still relevant. Basically, fans are selected to operate around one working point. It means that at this point, they should work with the highest efficiency. Unfortunately, in the case of many industrial processes, the fan working point can vary, sometimes over a wide range. This is a result of varying industrial process requirements or changes in the installation (for example, the extension or reduction of an underground mine). Such situations lead to a decrease of the fan's average operation efficiency, which can be even much lower than the maximal one. To ensure operation with relatively high efficiency over a wider range, fans need

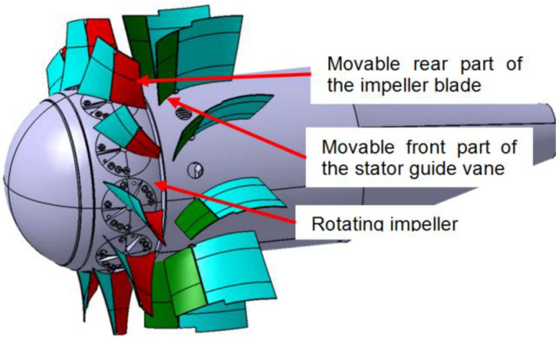

Fig. 1. A new idea of an axial fan with regulation based on the use of divided impeller blades and guide vanes into two parts, i.e., fixed and movable.

to be adjusted. There are many methods for regulating fans, from which, in the case of axial devices, regulation with use of frequency converters, regulation with the use of guide vanes, and regulation with the use of impeller movable blades are widely employed [4–13].

Literature survey indicates that there are fields where a lack of knowledge still exists. The need for further improvement is underlined in scientific works published in recent years [14–17]. Up to now, no works have been done to find whether an axial fan with both divided impeller blades and guide vanes are used, where one part is movable related to the other (fixed part). The movement of the movable blade or vane movable part leads to change of the fan's flow parameters. This new idea is shown in Fig. 1. Knowledge of the regulation parameters of a fan with such solution will give a chance to compare it with other methods, and because of this, the results of simulations of a reference fan with the regulation method using movable impeller blades will also be described. It seems that the fan with the new regulation method should have a wider operation range with relatively high efficiency, and the use of divided blades should result in lower value of force demanded to rotate movable blades. What is very important, such a method can be realized without need of implementation of the special steering devices. In such a case, blade adjustment is realized during fan standstill and is much cheaper than regulation mechanisms used during impeller rotation or frequency converters used to change rotational speed.

Within this paper the new concept of axial fan regulation is described. The scope is focused on the numerical investigation of the prototype of the fan with the new regulation method. The design and calculation stages, realized within the international EUREKA ETAF projec, are summarized within this work. They resulted in the final geometrical form of the prototype fan that is currently under construction and will be tested in the next months.

TABLE I

Geometrical data of the basic axial fan and its smaller model.

Fan	Unit	Basic axial fan	Smaller model of axial fan
Flow rate Q	$\mathrm{m^3/s}$	458.3	17.8
Total pressure rise Δp_t	Pa	4905	1386
External impeller diameter D_z	mm	3700	1000
Impeller rotational speed n	rpm	750	1475
Power P	kW	2800	31
Number of impeller blades	–	10	10
Number of guide vanes	–	9	9

For the purpose of the work described in the article, a fan with a common design has been chosen. In this design, the fan consists of a rotating impeller and stator guide vanes. Fans with such forms are widely used, for example, in ventilation systems, and because of this, the authors decided to design an axial fan as a potential replacement of the biggest centrifugal fan used in Polish underground mine ventilation systems, which is WPK-5,35. Nominal working point of this fan is within the flow rate $Q = 458.3\ \mathrm{m^3/s}$ with total pressure rise $\Delta p_t = 4905$ Pa. Basic data of the axial fan is presented in Table I. Due to the high power in the case of the WPK-5,35 fan (about 2.8 MW), to realize goals of the project, a smaller model of the axial fan was considered. Its data is given in Table I and was defined according to the affinity law [4–13].

The presented study will fill the gaps in knowledge about fan regulation methods, since it mainly covers a new, previously unknown, regulation method of the axial fan.

2. Numerical flow calculations

Numerical flow calculations are nowadays crucial during flow machinery design and preliminary evaluation [10, 14, 15, 18–23].

The method is based on the solution of the continuity and Navier–Stokes equations [10, 18–20, 24]

$$\frac{\partial c_i}{\partial x_i} = 0 \tag{1}$$

and

$$\frac{\partial(\rho\, c_i)}{\partial t} + \frac{\partial(\rho\, c_i\, c_j)}{\partial x_j} = \frac{\partial p}{\partial x_i} + \mu \frac{\partial^2 c_i}{\partial x_j^2} + \rho\, f_i, \tag{2}$$

where: c_i — velocity components, p — static pressure, f_i — external forces, μ — dynamic viscosity.

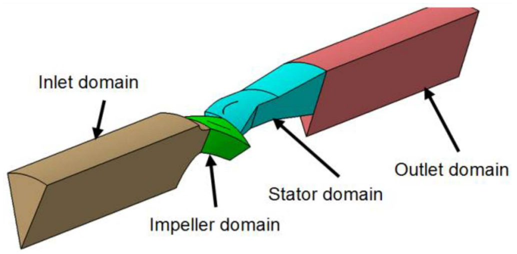

Fig. 2. Exemplary polyhedral mesh generated one the reference model of the axial fan.

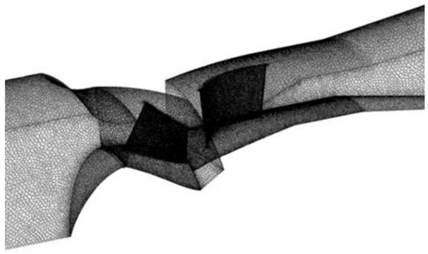

Fig. 3. Geometrical model od 1/10 of the impeller and 1/9 of the stator, with a description of the defined fluid flow zones.

In practice, partial diferential equations are replaced by a system of algebraic equations. Such system of equations is solved numerically until the results, mainly in the form of pressure, velocity, density, and temperature distributions as well as streamlines, are obtained within the calculation domain [24].

In the case of the current work, an axial fan with two different regulation methods has been considered. The first fan was equiped with an already known regulation method based on the rotation of the movable impeller blades. Such fan is a reference fan in the present project. The second fan was equiped with a proposed new regulation method (i.e., a change of geometry based on the divisions of the impeller blades and stator vanes into fixed and movable parts). Because of that, a preliminary comparison of the new method with the already known one can be performed.

Firstly, the verification of the reference model will be described. Then, computational fluid dynamics (CFD) simulations of the fan with the new regulation method will be described.

2.1. Verification of the model

In order to get reliable turbo machinery flow performance curves using CFD technique, the simulation model should be both verified and validated. In the case of the current work, only verification can be realized, as these are design stages of the project and the prototype of such fan will be tested in the near future. In practice, the verification of numerical model can be realized by evaluating the discretization error. This is performed through a grid refinement study, where at least two models with different numbers of elements should be calculated [25, 26].

Within this work, the verification has been carried out based on the reference model of a fan in the basic blade position. Models with three different mesh sizes have been prepared: (i) a model with the most refined mesh and the number of elements $N_1 = 1447217$, (ii) a model with a medium mesh size and the number of elements $N_2 = 1138548$, and (iii) a model with a coarse mesh and the number of elements $N_3 = 541847$. In each case, the generated mesh was a polyhedral mesh with refined grids along the walls to ensure that the wall distance parameter $y+$ remains below 250. The simulation models used in this work have been prepared with periodic boundary conditions, since the impeller is equipped with 10 blades and the stator with 9 vanes, and the other elements are axially symmetrical. A model with the defined different fluid flow domains is shown in Fig. 2, whereas an exemplary mesh is shown in Fig. 3.

Simulations were conducted for steady-state flow, assuming a fixed position between the impeller and stator. To account for the components responsible for the rotational movement of the fluid in the rotating zone, the so-called frozen rotor approach was applied, with a rotational speed of the impeller equal to 1475 rpm [24, 27, 28]. In order to obtain fan characteristics under standard conditions, air with a constant density of 1.2 kg/m^3 was defined as the fluid. The calculations were performed with the use of two equation turbulence model (k–ε, *Realizable*), which is useful in such simulations [10, 18–22].

To comply with standardization, fan flow performance curves should be evaluated based on at least five different working points (WP). For this reason, simulations have been performed at five different points (numbered P1 to P5, from the minimum to the maximum considered flow rate value). In each case, the inlet boundary condition was velocity inlet with velocity value defined according to the required flow rate. Inlet and outlet boundary conditions (BC) were defined respectively as velocity inlet BC and pressure outlet BC. Inlet velocity was defined according to required flow rate whereas outlet pressure value was equal to 0 Pa.

The results of the calculations verificating a reference model are summarized in Table II. From the fan performance curves (Fig. 4), it is clear that the models with the considered mesh sizes give comparable results for the total pressure rise (Fig. 4a) and efficiency (Fig. 4b). Larger differences between the curves are seen for the power characteristic (Fig. 4c). Despite of this, the model can be considered sufficiently accurate for further calculations.

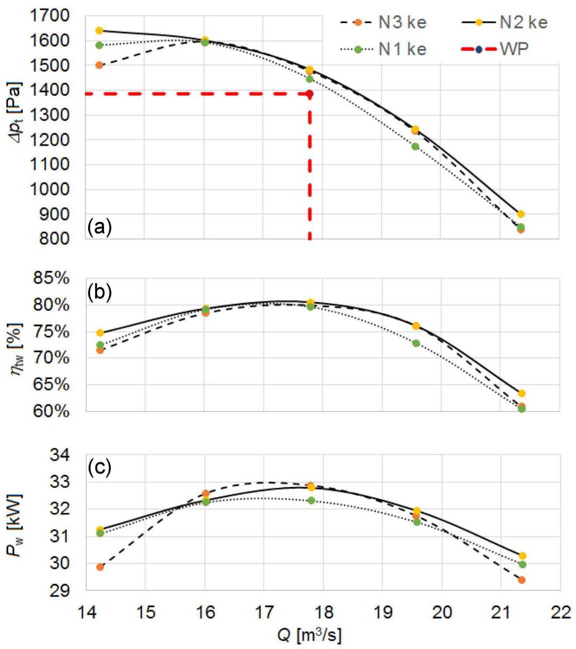

Fig. 4. The results of the model verification process presented as fan performance curves ((a) total pressure rise, (b) fan efficiency, and (c) power in function of flow) for models with different mesh sizes N_1, N_2, N_3; WP — fan design point.

TABLE II

Results of CFD calculation and the verification process of the reference fan; WP — working point, Q — flow rate, Δp_t — fan total pressure rise, P_w — shaft power, P_h — hydraulic power, η_{tw} — total efficiency related to shaft power.

Mesh	Number of elements	WP	Q [m^3/s]	Δp_t [Pa]	P_w [kW]	P_h [kW]	η_{tw} [%]
N_3	541847	P1	14.24	1501	29.9	21.4	71.5
		P2	16.02	1596	32.6	25.6	78.5
		P3	17.79	1477	32.9	26.3	79.9
		P4	19.57	1235	31.8	24.2	76.1
		P5	21.35	839	29.4	17.9	60.9
N_2	1138584	P1	14.24	1640	31.3	23.4	74.7
		P2	16.02	1601	32.3	25.7	79.3
		P3	17.79	1484	32.8	26.4	80.5
		P4	19.57	1243	31.9	24.3	76.1
		P5	21.35	901	30.3	19.2	63.5
N_1	1447217	P1	14.24	1583	31.1	22.5	72.5
		P2	16.02	1594	32.2	25.5	79.2
		P3	17.79	1447	32.3	25.7	79.7
		P4	19.57	1175	31.5	23.0	72.9
		P5	21.35	850	30.0	18.1	60.5

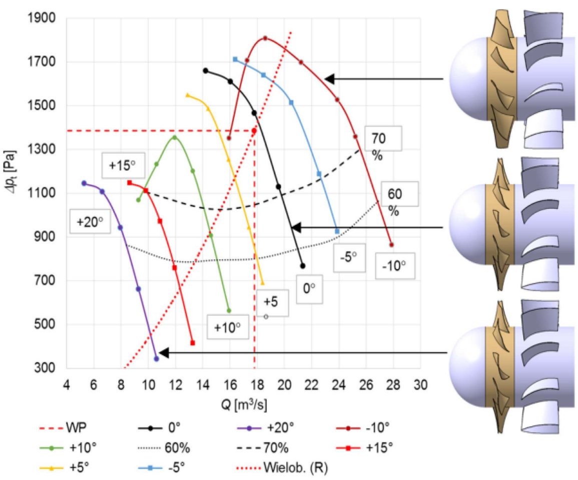

Fig. 5. Reference fan regulation characteristic for different blade positions. Curves of minimal efficiency equal to 60% and 70%, as well as an exemplary system pressure drop curve (R) passing through the fan design point (WP), are also shown.

2.2. Calculations of a reference fan with movable impeller blades

An improved version of the basic fan was used in further calculations. The aim was to determine the regulation characteristic for a reference fan with whole movable blades. The regulation was achieved by varying the blade angle, which is well known and widely used method. The impeller blades were assumed to rotate in the range from $-10°$ to $+20°$, relative to the basic blade position. Calculations of the fan regulation characteristic were performed for these and few other middle blade positions $(-5°, +5°, +10°, +15°)$. The resulting regulation characteristic is shown in Fig. 5, which also indicates the basic $(0°)$, minimum $(+20°)$ and maximum $(-10°)$ blade positions. In this figure, the minimum range of the fan operation corresponding to minimum efficiency equal to 60% and 70% is drawn. An exemplary system pressure drop curve (R) is also included.

2.3. Calculations of a fan with the new regulation method

Calculations of the axial fan using the invented, new regulation method were realized in two steps. Firstly, CFD simulations have been used to estimate which of the impeller blade division is the most appropriate for the considered fan. Three different divisions were assumed for evaluation (movable blade length equal to 1/2, 1/3, and 1/4 of the whole blade length). In the case of the stator vane, it was assumed that the length of the movable vane part is

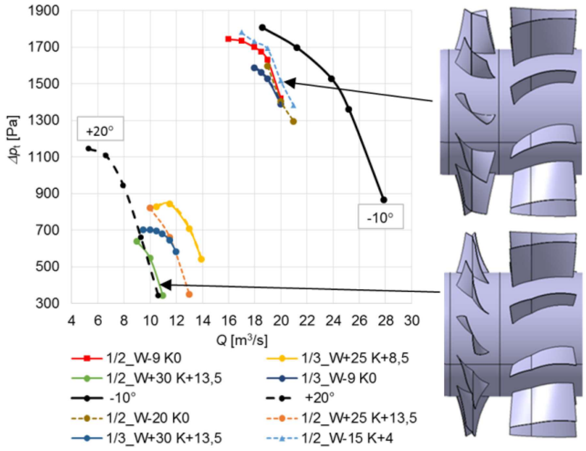

Fig. 6. Comparison of total pressure curves for limiting positions calculated using the new regulation method, with two different blade divisions (1/2 and 1/3) and different blade and vanes rotation angles. For comparison, the curves of a reference fan at the limiting the movable blade positions (+20° and −10°) are plotted. Additionally, two fans representing the final limiting positions of the movable blade and vane parts are shown.

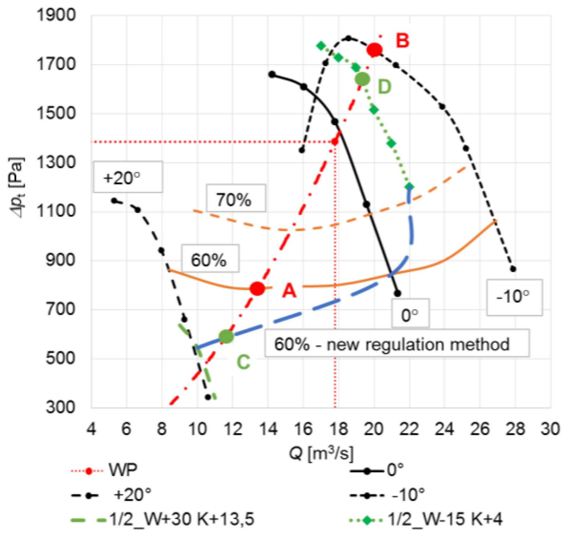

Fig. 7. Fan regulation characteristic (limiting fan pressure curves for a reference fan with whole movable blade, with drawn lines of minimal efficiency 60% and 70%, and limiting fan pressure curves for a fan with the new regulation method and with a line of minimal efficiency 60%). The dotted red line shows the fan selection point and the center red line represents the system pressure curve; points A–D are used to calculate the fan control range for both methods.

1/3 of the whole vane length, and this was not verified within the project. The second step was to find limiting positions of the movable blade and vane parts with the goal to cover a similar regulation range as for the reference fan (shown in Fig. 6 for limiting movable blade positions of −10° and +20°). In this step, the work was also focused on optimizing the movable blade and vane rotation angles to enhance the fan efficiency.

The selection of the most appropriate movable blade length was performed iteratively in tens of simulations. In this process, comparison were made not only for the fan with blade divided in the portions 1/2, 1/3 and 1/4 for the fan design point and basic blade position, but also for other blade positions and other working points. Only such approach gave a chance to select the most appropriate movable blade division, since the axial fan should work with the highest possible efficiency over a wide range. The results obtained for the calculation at the fan design point are summarized in Table III. The final comparison of the obtained results can be performed for few considered movable blade and vane positions, compiled in Table IV with the flow calculation results presented in Table V. For these configurations, the obtained flow calculation results are plotted in Fig. 7. From this data, it is clear that the fan designed with the new regulation method and a 1/2 blade division has the best performance compared to the other considered divisions. Unfortunately, the calculations shown that the new idea has limitation regarding regulation to higher flow rates and pressure rise values and does not allow achieving results for the maximal whole movable blade position (−10°). It is caused by growing flow losses when blade rotation angles higher than about −15° are considered (compare curves for angles −20°, −15° with −9°).

Ultimately, based on the performed analysis, the following conclusions and further recommendations for the manufacturing of the fan prototype can be given:

(i) The most appropriate movable blade division is 1/2. Such configuration covers a wider range of working points with higher efficiency than the other divisions, with relatively smaller rotation angles of the movable blade and vane elements in the same time.

(ii) The limiting positions of the movable blade are selected as +30° and −15° and of the vane as 0° and +13.5°. Both positions are shown in Fig. 6.

2.4. Comparison of results

The simulation results allowed for a comparison of a reference fan with fully movable blade and a fan using the new regulation method. It was realized with the use of a control range parameter, which is calculated in terms of the flow rate or pressure. This parameter is calculated as the ratio of the difference between the maximum and minimum values of the

TABLE III

Comparison of simulation results for three different blade divisions calculated at the fan design point (P3); WP — working point, Q — flow rate, Δp_t — fan total pressure rise, P_w — shaft power, P_h — hydraulic power, η_{tw} — total efficiency related to shaft power.

Blade division ratio	WP	Q [m^3/s]	Δp_t [Pa]	P_w [kW]	P_h [kW]	η_{tw} [%]
1/2	P3	17.79	1436	31.4	25.6	81.4
1/3		17.79	1419	30.9	25.3	81.6
1/4		17.79	1414	31.4	25.2	80.1

TABLE IV

Description of the models used for the final consideration of the new regulation method.

Model designation	Blade division	Blade rotation angle	Vane rotation angle
1/2_W+30 K+13.5	1/2	+30°	+13.5°
1/2_W+25 K+13.5	1/2	+25°	+13.5°
1/2_W-9 K0	1/2	−9°	0°
1/2_W-15 K+4	1/2	−15°	+4°
1/2_W-20 K0	1/2	−20°	0°
1/3_W+30 K+13.5	1/3	+30°	+13.5°
1/3_W+25 K+8.5	1/3	+25°	+8.5°
1/3_W-9 K0	1/3	−9°	0°

flow or pressure to the maximum value, within the efficiency range of at least 60%, according to following equations [4, 29]

$$l_Q = \frac{Q_{\max} - Q_{\min}}{Q_{\max}} \tag{3}$$

in relation to flow rate, and

$$l_p = \frac{\Delta p_{\max} - \Delta p_{\min}}{\Delta p_{\max}} \tag{4}$$

in relation to pressure, where: $Q_{\max}$ — maximum flow value, $Q_{\min}$ — minimum flow value, $\Delta p_{\max}$ — maximum pressure value, $\Delta p_{\min}$ – minimum pressure value.

In Fig. 7, the total pressure rise curves for the limiting positions of the two compared methods are shown. Based on the exemplary system pressure drop curve plotted through the fan design point (WP), the regulation range is marked by points A and B for the reference fan, and by points C and D in the case of the new regulation method. Using (3) and (4), the regulation range for both fans has been calculated. A summary of this analysis is presented in Table VI. From the numerical data, it is clear that for the considered fan, the new regulation method ensures an increase of the flow regulation parameter by 10.2% and by 12.2%, respectively, in terms of flow and pressure. This is mainly due to the fact that the new regulation method extends the range of economical fan operation (generally with efficiency above 60% [4, 29]) in the range of lower flows and pressures. On the other hand, numerical investigations indicate that the new regulation method cannot achieve as high flow rates and pressures as the regulation with whole movable blades. In the case of regulation at higher flow and pressure values, losses increase more rapidly and due to this the maximal efficiencies in this range are smaller compared to the regulation with whole movable blade.

TABLE V

Results of CFD calculation of different blade and vane configurations; WP — working point, Q — flow rate, Δp_t — fan total pressure rise, P_w — shaft power, P_h — hydraulic power, η_{tw} — total efficiency related to shaft power.

Model designation	WP	Q [m^3/s]	Δp_t [Pa]	P_w [kW]	P_h [kW]	η_{tw} [%]
1/2_W+30 K+13.5	P1	9.00	638	9.9	5.7	57.8
	P2	9.99	546	9.0	5.5	60.7
	P3	10.99	340	7.2	3.7	52.2
1/2_W+25 K+13.5	P1	9.99	820	12.8	8.2	63.9
	P2	11.51	660	11.1	7.6	68.7
	P3	13.01	348	8.1	4.5	56.2
1/2_W-9 K0	P1	15.99	1744	37.6	27.9	74.2
	P2	17.00	1734	38.1	29.5	77.4
	P3	18.00	1698	38.6	30.6	79.1
	P4	18.50	1675	39.1	31.0	79.3
	P5	18.99	1629	39.5	30.9	78.2
	P6	20.00	1418	38.9	28.4	73.0
1/2_W-15 K+4	P1	17.00	1778	42.5	30.2	71.1
	P2	18.00	1729	42.5	31.1	73.3
	P3	18.99	1688	43.6	32.1	73.5
	P4	20.00	1514	43.3	30.3	70.0
	P5	21.00	1379	43.3	28.9	66.8
	P6	22.03	1249	43.6	27.5	63.1
1/2_W-20 K0	P1	18.99	1595	45.7	30.3	66.3
	P2	20.00	1405	45.5	28.1	61.8
	P3	21.00	1293	46.1	27.2	58.9
1/3_W+30 K+13.5	P1	9.50	700	12.2	6.7	54.4
	P2	9.99	701	12.3	7.0	57.0
	P3	10.50	694	12.3	7.3	59.5
	P4	10.95	680	12.2	7.4	61.2
	P5	11.49	644	11.8	7.4	62.6
	P6	12.00	581	11.3	7.0	61.7
1/3_W+25 K+8.5	P1	10.49	827	14.4	8.7	60.2
	P2	11.51	844	14.6	9.7	66.4
	P3	13.01	706	13.4	9.2	68.7
	P4	13.94	540	11.6	7.5	64.8
1/3_W-9 K0	P1	18.00	1586	36.3	28.5	78.6
	P2	18.50	1561	36.6	28.9	78.9
	P3	18.99	1525	36.8	29.0	78.7
	P4	20.00	1388	36.7	27.8	75.6

TABLE VI

Comparison of a control range parameter for the fan with whole movable blade and the fan with the new regulation method.

Control range parameter	Reference fan with whole movable blade	Fan with new regulation method
l_Q	32.5%	42.7%
l_p	54.9%	67.1%

Numerical tests of the new regulation method and its comparison with the reference fan leads to the conclusion that a better regulation method would be a combination of an impeller with whole movable blade and divided stator guide vanes with movable front parts. Such a solution should extend range of economical operation of the fan with whole movable blade, especially in lower flow and pressure values.

3. Discussion

The results of the fan with the new regulation method show its possible usage area. In cases where there is a requirement for regulation at lower pressures and flow values, it can be realized with enhanced efficiency. Due to this, a significant reduction of electricity consumption in many industrial sectors can be achieved. This method can be useful for larger fans, especially those with power above 500 kW. In such cases, users do not want to invest a lot of money in frequency converters and the required infrastructure, but they still want to have possibility to adjust the fan to a new working point, even if this is realized during fan standstill.

Regulation of the axial fan through its geometry adjustment has advantages related to the use of frequency converters. Firstly, such method does not influence the rotational speed and thus has a much lower impact onto stresses in the structure. In the case of frequency converters, there is a restricted region of regulation for higher flows and pressures, which can be reached only by increasing the rotational speed. In such a situation, stresses in the structure can exceed allowable limits. The second problem is connected with rotating masses and the resultant inertia forces. When the rotational speed is changed for larger fans with a high impeller mass, dynamic effects during such process can significantly decrease the reliability of the fan components (rotating parts, bearings etc.). This may even be increased in cases when any of the fan's natural frequencies are met, resulting in higher vibration level due to resonance effect.

Fan users accept the fact that a regulation method based on geometry change is the most reliable and much cheaper when there is no steering mechanism that would allow users to regulate the fan during its operation. In installations where flow and pressure requirements do not change rapidly, such solution can be a compromise between fan price, reliability, and electricity savings. In many cases, it is much better to have any regulation method than operate the fan in low efficiency ranges for longer period without the possibility to adjust its parameters.

4. Conclusions

The proposed new axial fan regulation method showed that it can be a useful way to adjust industrial axial fans with larger powers. The results proved that such fan adjustment can extend the operation range of fan with higher efficiency compared to fans without any regulation method.

Comparison of the invented regulation method with the well-known method used in the reference fan (with whole movable blade) uncovered some advantages and disadvantages of the proposed approach, considering limitations of the current project. The most noticeable features of the new regulation method are as follows:

- The new method provides a higher regulation control range parameter related to flow and pressure, by 10.2% and 12.2%, respectively.
- The new method offers a much wider regulation range in the lower flow and pressure region.
- The new regulation method is unable to reach pressure and flow values as high as those required for the reference fan. This is mainly due to the fact that the blade angle of attack does not change (i.e., does not decrease) in the same way as the movable blade end. As a result, growing flow losses occur.
- The division of the blade into a front (fixed) and rear (movable) part results in a gap between both elements. This gap causes pressure losses due to flow through the suction side to the pressure blade side. For this reason, this gap should be kept as small as possible.

These conclusions fill the gaps in knowledge about different regulation methods of an axial fan, since they address a newly invented method that — up to now — has not been considered or investigated. And what is more, this has been realized in comparison to already known and widely used regulation method.

Here it needs to be emphasized that conclusions stated above will be verified on a real object during field tests that are being prepared and will be carried out in next months.

Ultimately, an additional important conclusion arising from the conducted work is that better regulation parameters would be ensured by a fan with combined regulation method proposed in the reference fan and in the new idea of regulation. This means that the fan with whole movable blades and divided stator guide vane (movable front part) should have significantly better performance than both solutions considered in the project.

Acknowledgments

This research was completed within the EUREKA Innovations Beyond Borders programme, under the 2022 call, and co-financed by the National Centre for Research and Development (grant agreement No. EUREKA/2022/83/ETAF/2023). The project title is EFFECTIVE TECHNOLOGY FOR FLOW PARAMETERS ADJUSTMENT OF AXIAL FANS (ETAF).

References

[1] European Commission, "Industrial Fans — Ecodesign requirements apply to this product" (accessed Jul. 2025).

[2] EUR-Lex, "Commission Regulation (EU) No 327/2011 of 30 March 2011 implementing Directive 2009/125/EC of the European Parliament and of the Council with regard to ecodesign requirements for fans driven by motors with an electric input power between 125 W and 500 kW" (accessed Jul. 2025).

[3] EUR-Lex, "Commission Regulation (EU) 2024/1834 of 3 July 2024 implementing Directive 2009/125/EC of the European Parliament and of the Council with regard to ecodesign requirements for fans driven by motors with an electric input power between 125 W and 500 kW and repealing Commission Regulation (EU) No 327/2011" (accessed Jul. 2025).

[4] S. Kuczewski, *Wentylatory*, WNT, Warszawa 1978 (in Polish).

[5] E. Tuliszka, *Sprężarki, Dmuchawy i Wentylatory*, WNT, Warszawa 1969 (in Polish).

[6] S. Fortuna, *Wentylatory: Podstawy Teoretyczne, Zagadnienia Konstrukcyjno-Eksploatacyjne i Zastosowanie*, TECHWENT, Kraków 1999 (in Polish).

[7] L. Bommes, J. Fricke. R. Grundmann, *Ventilatoren* 2nd Ed., Vulkan-Verlag, Essen 2002 (in German).

[8] F. Mode, *Ventilator-Anlagen*, 4th ed., Walter de Gruyter, Berlin 1972.

[9] W. Bohl, *Ventilatoren*, Vogel-Buchverlag, Würzburg 1983.

[10] T. Carolus, *Ventilatoren. Aerodynamischer Entwurf, Sschallvorhersage, Konsruktion*, 3rd ed., Springer Vieweg, 2013 (in German).

[11] B. Eck, *Ventilatoren*, 6th ed., Springer-Verlag, Berlin 2003 (in German).

[12] F. Schledner, G. Klingenberg, *Ventilatoren im Einsatz: Anwendung in Geräten und Anlagen*, VDI Verlag, Düsseldorf 1996 (in German).

[13] W.T.W. Cory, *Fans & Ventilation: A Practical Guide*, Elsevier 2005.

[14] P. Odyjas, J. Więckowski, D. Pietrusiak, P. Moczko, *Acta Mech. Autom.* **17**, 16 (2023).

[15] P. Moczko, P. Odyjas, D. Pietrusiak, J. Więckowski, P. Scholz, M. Dix, T. Osiecki, T. Timmel, L. Kroll, *Energies* **15**, 893 (2022).

[16] Z. Chuangxin, L. Ziliang, H. Song., H. Ge, L. Xingen, Z. Shengfeng, Z. Junqiang, *Aerosp. Sci. Technol.* **109**, 106411 (2021).

[17] X. Wei, C. Genglin, S. Huijin, Z. Pengcheng, C. Xuemei, *Energy* **284**, 129363 (2023).

[18] J.F. Gülich, *Centrifugal Pumps. 2nd edition*, Springer-Verlag, Berlin 2008.

[19] S. Fortuna, *Badanie Pracy Użytecznej i Strat w Wentylatorze Promieniowym*, Wydawnictwo AGH, Kraków 2011 (in Polish).

[20] S. Fortuna, K. Sobczak, *Mechanics* **27**, 138 (2008).

[21] K. Jożwik, A. Papierski, K. Sobczak et al., *Trans. Inst. Fluid-Flow Mach.* **131** (2016).

[22] M. Zawiślak, *Transp. Przemysłowy i Masz. Rob.* **2**, 112 (2014).

[23] W.K. Ng, M. Damodaran, in: *Proc. 4th European Conf. on Computational Fluid Dynamics ECCOMAS CFD, Egmond aan Zee (The Netherlands)*, 2006.

[24] Ansys, ANSYS Fluent Theory Guide, 2024.

[25] "Uncertainty and Error in CFD Simulations" (accessed Mar. 2025).

[26] W.L. Oberkampf, T.G. Trucano, *Prog. Aerospa. Sci.* **38**, 209 (2002).

[27] Ansys, ANSYS FLUENT 12.0 Tutorial Guide, 2009.

[28] Ansys, ANSYS FLUENT 12.0 User's Guide, 2009.

[29] V.I. Kovalevskaja, G.A. Babak, V.V. Pak, *Shahtnye Centrobezhnye Ventiljatory*, Nedra, Moskva 1976 (in Russian).