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EXPERIMENTAL STUDYING OF THE ENREGY EFFICIENCY OF METHODS USED TO REGULATE THE FLOW RATE IN FAN SYSTEMS

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Abstract: This work presents a comparative experimental analysis of the energy efficiency of three different methods for regulating a fan flow rate: throttling, using inlet guide vanes and changing the voltage frequency - the motor speed of rotation is varied using a variable frequency drive (VFD). The impact of the initial operating mode on the effectiveness of regulation using inlet guide vane apparatus and frequency inverter (refered to as the VFD flow control method) is assessed. The tests are conducted on an experimental setup located in the Laboratory of Hydraulic Machinery at the University "Angel Kante" of Ruse. A centrifugal fan for high pressure specifically the type 19CS48 is utilized for this purpose.

Keywords: Fan Systems, EnergyEfficiency, Regulating Methods, Flow Rate

INTRODUCTION

The regulation of fan systems involves adjusting the aerodynamic parameters of a fan, using special devices such as guide vane apparatus, throttles or frequency inverters - VFDs. This allows for continuous adjustments to the fan operating parameters. The goal of regulation is to increase or decrease the flow rate of the fan unit, thereby varying the pressure and power required to drive the fan unit.

Different methods for regulating the flow rate of ventilators are used in practice. The most commonly used methods for flow control are: regulation by throttling, regulation by means of an inlett guide vane apparatus (IGV) and regulation by varying the rotation speed of the driving electric motor using a frequency inverter (VFD). In this study, a comparative analysis of the effectiveness of these three methods in regulating the flow rate of a centrifugal fan for high pressure, specifically the type 19CS48, is conducted.

The experiments are carried our using laboratory equipment located in the Laboratory of Hydraulic Machines at the Department of Heat, Hydraulics and Engineering Ecology of Ruse University.

EXPOSITION

Experimental setup

The experiments in the present study are conducted using the experimental setup shown in Figure 1.
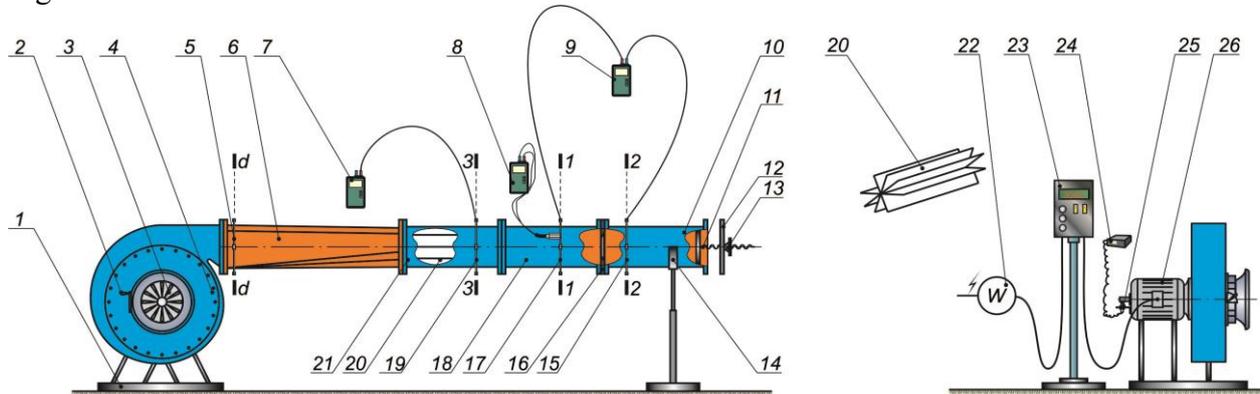

Fig. 1. Experimental setup includes: 1 – motionless stand; 2 – guide vane control mechanism; 3 – guide vane apparatus; 4 – centrifugal fan for high pressure (type 19CS48); 5 - collector for outputting the static pressure in a section d-d; 6 – transition section; 7 and 9 – micromanometer Testo 521; 8 – thermocouple with reading device; 10, 18, 21 – cylindrical tubes; 11 – screw; 12 – plane disc shutter; 13 – lock (safety) nut; 14 – motionless stand; 15 - collector for outputting static pressure in section 2-2; 16 – standardized aperture (blende); 17 - collector for outputting static pressure in section 1-1; 19 - collector for outputting the static pressure in a section 3-3; 20 – standardized guide vane apparatus; 22 – wattmeter; 23 – variable frequency drive (VFD); 24 – rotation speed reading device; 25 – inductive converter; 26 – drive motor.

The experimental setup is of the injection (discharge) type and designed according to ISO 5801:2007 requirements. The high-pressure centrifugal fan 4, fixedly mounted on stand 1, drawing air from the atmosphere through the smooth inlet of the guide vane apparatus 3 and directing it into a discharge pipe consisting of four sections - 6, 21, 18 and 10. The vanes of the inlet guide apparatus are set at a specific angle using control mechanism 2. The outlet flange of the fan, which has a rectangular shape, is connected to the transition section 6, which ends with a circular cross-section. Following the transition section straight pipe sections 21, 18 and 10 with internal diameters $D = 0.2 \text{ m}$ are installed in series. At the beginning of section 21 there is a fixed guide vane apparatus 20, design to straighten the flow streamlines after the transition section 6.

Between the straight pipe sections 18 and 10, a standard aperture (blende) 16 with an orifice diameter $d = 0.14 \text{ m}$ is installed. At the end of section 10, the screw 11 is installed, on which the shutter 12 is wound and locked with the counter nut 13. Stand 14 is used as a second mainstay for the fan. To reduce the static pressure in each of the sections d-d, 3-3, 1-1 and 2-2, four nozzles are installed, connected in common collectors 5, 19, 17 and 15.

The fan is driven by using an asynchronous electric motor (AC) 26 of the type 4AM100L2-0M2 with power $P_{el} = 5.5 \text{ kW}$, nominal speed $n = 2880 \text{ min}^{-1}$ and a coefficient of efficiency $\eta_{ed} = 0.875$. The electric motor is flange-mounted on the spiral body, and the torque from its shaft is transmitted to the impeller by means of a key joint. If necessary, to change the fan speed, it is possible to connect the variable frequency drive (VFD) 23 between the wattmeter 22 and the electric motor 26.

In order to determine the operating parameters of the fan during various types of tests, the following devices are connected to the experimental setup: micromanometer 7 for measuring the static manometric pressure in separate sections of the discharge air duct; thermocouple with reading device 8 for reading the air temperature in section 1-1; differential micromanometer 9 for determining the difference between the pressures in sections 1-1 and 2-2; an inductive converter 25 connected to a reading device 24 for measuring the fan speed; wattmeter 22 for determining the electrical power at the input of the VFD or the electric motor. In addition to the devices listed, in the room where the test is performed it is also necessary to have: a device for determining the atmospheric pressure (barometer), a thermometer for determining the temperature of the atmospheric air and a device for determining the relative humidity of the air (psychrometer).

Determining the fan operating parameters

- **Atmospheric air density**

Determining the density of atmospheric air is crucial when studying machines that operate with compressible fluids like fans. The density of moist air is impacted by relative humidity φ_{atm}, atmospheric pressure p_{atm} and temperature T_{atm}. After measuring these parameters, the moisture content of the air can be calculated using the following equation:

$$d_{atm} = 0,622 \frac{\varphi_{atm} \cdot p_v}{p_{atm} - \varphi_{atm} p_v}, \left[kg / kg \right], \tag{1}$$

where $p_v, \left[Pa \right]$ is the saturation pressure of water vapor, which is determined by taking its values from relevant reference books depending on the temperature.

The gas constant of atmospheric air is determined by the equation below:

$$R_{atm} = 15,935 \frac{28,96 \cdot d_{atm} + 18,016}{1 + d_{atm}}, \left[\frac{J}{kgK} \right]. \tag{2}$$

The density of air is determined by the Clapeyron equation:

$$\rho_{atm} = \frac{p_{atm}}{R_{atm} T_{atm}}, \left[kg / m^3 \right]. \tag{3}$$

- **Determining the volume flow rate of the studied fan**

The volumetric flow rate of the fan is defined as the volume of air (or gas) passing through the outlet of the fan per unit of time, based on the conditions at the inlet of the fan system (pressure, temperature and relative humidity of the atmospheric air). In this case, a method of measuring the flow rate using a throttling device - blende 16 and an associated differential micromanometer 9 is selected. The measurement principle involves creating a local pressure drop Δp in the air duct by throttling and measuring this drop using a differential manometer. An equation that expresses the relationship between this pressure drop Δp and the mass flow through the orifice is utilized:

$$q = \frac{C}{\sqrt{1 - \beta^4}} \varepsilon \frac{\pi}{4} d^2 \sqrt{2\rho_1 \Delta p}, \left[kg / s \right], \tag{4}$$

where:
- C is a coefficient, which is determined by an iterative procedure outlined in the standard ISO 5167-2:2003 "Measurement of fluid flow by means of pressure differential devices inserted in circular cross-section conduits running full - Part 2: Orifice plates";
- $\beta = \frac{d}{D}$ - the ratio of the aperture (blende) and tube diameters;

- ε - the coefficient of expansion, which is determined using the following equation:

$$\varepsilon = 1 - \left(0,351 + 0,256 \cdot \beta^4 + 0,93 \cdot \beta^8\right) \cdot \left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{1}{k}}\right], \tag{5}$$

where: p_1, $[Pa]$ is the absolute pressure in section 1-1, p_2, $[Pa]$ is the absolute pressure in section 2-2, and $k = 1,4$ - is the adiabatic exponent.

- ρ_1, $\left[kg / m^3\right]$ - the density of air in section 1-1.

The pressure p_1 is determined using the micromanometer 9, when disconnecting the connection with the collector in section 2-2. The air density ρ_1 is calculated after determining the temperature in section 1-1 using the thermocouple, installed in this section. The following equation is used for this purpose:

$$\rho_1 = \rho_{atm} \cdot \frac{p_1}{p_{atm}} \cdot \frac{T_{atm}}{T_1}, \left[kg / m^3\right], \tag{6}$$

After performing the necessary iteration procedures, the following approximate equation for determining the mass flow rate through the aperture (blende) is obtained:

$$q = 0,01679\sqrt{\Delta p}, [kg / s], \tag{7}$$

where Δp is the reading of the micromanometer 9.

The fan volume flow rate is determined by applying the Continuity equation:

$$Q = \frac{q}{\rho_{atm}}, \left[m^3 / s\right]. \tag{8}$$

- **Determining the fan pressure**

It is necessary to determine the pressure corresponding to the energy that the fan gives to one cubic meter of air passing through its outlet. In the proposed experimental setup scheme, the fan pressure is determined by applying the following equation:

$$p = p_{M3} + \left(1 + \zeta_{пр}\right) \cdot \rho_3 \frac{\upsilon_3^2}{2}, [Pa], \tag{9}$$

where:

- p_{M_3} is the static gauge pressure measured in section 3-3 (fig. 1);

- $\zeta_{пр} = 0,46$ - the experimental determination of the coefficient of resistance from the previous section, which relates to the velocity in section 3-3, used to indicate the energy loss between sections d-d and 3-3;

- ρ_3 - the air density in section 3-3, which is determined by applying the following equation:

$$\rho_3 = \rho_{atm} \frac{p_{atm} + p_{M3}}{p_{atm}}, \left[kg / m^3\right]; \tag{10}$$

- υ_3 - the average velocity of the flow in section 3-3, determined by using the equation:

$$\upsilon_3 = \frac{4Q}{\pi D^2}, [m / s]. \tag{11}$$

- **Determining the dynamic fan pressure**

The dynamic pressure of the fan is defined by the kinetic energy of the air (gas) in section

d-d. It is calculated according to the equation:

$$p_{dyn} = \rho_d \frac{\upsilon_d^2}{2}, \left[Pa\right]. \tag{12}$$

With sufficient accuracy it can be assumed that the density is $\rho_d = \rho_3$, and the average velocity in section d-d is determined by the equation:

$$\upsilon_d = \frac{Q}{b \cdot h}, \left[m / s\right], \tag{13}$$

where the sides of the rectangular section d-d are: $b = 0,1 \text{ m}$ and $h = 0,2 \text{ m}$, respectively.

- **Determining the static fan pressure**

The static fan pressure represents the difference between total and dynamic pressure and is determined by applying the following equation:

$$p_{st} = p - p_{dyn}, \left[Pa\right]. \tag{14}$$

- **Determining fan power**

Fan power refers to the mechanical energy supplied to the fan shaft per unit of time. The experimental device presented allows for the determination of the electric power P_{el} of the driving electric motor, by using the wattmeter 22. By knowning the coefficient of efficiency of the electric motor (obtained from the manufacturer catalog data $\eta_{ed} = 0.875$), the fan power can be determined using the following equation:

$$P = P_{el} \cdot \eta_{ed}, \left[kW\right]. \tag{15}$$

Equation (15) is only valid just in if ase the variable frequency drive (VFD) 23 is not connected between the wattmeter and the electric motor. When conducting experimental tests with the VFD on, as shown in Fig. 1, the equation used to calculate the output power is as follows:

$$P = P_{VFD} \cdot \eta_{VFD}, \left[kW\right]. \tag{16}$$

where η_{VFD} is parameter that accounts for the total impact of the VFD on the measured electrical power P_{VFD}. In the absence of data on the VFD efficiency, an experiment is conducted by setting six different fan operating modes, using shutter 12, without activating the VFD. The power P_{el} is then measured using wattmeter 22. The experiment is repeated with the VFD turned on at each shutter position and corresponding speed, and the fan power is again measured by the wattmeter. Thus, the fan power P_{VFD} is read by the watemmeter. Analysis of equations (15) and (16) reveals that the following statement holds true for the same fan operating mode:

$$\eta_{VFD} = \frac{P_{el} \cdot \eta_{ed}}{P_{VFD}}. \tag{17}$$

The experimentally obtained results are presented in Table 1. It can be seen that up to 4.32 kW the VFD has no impact on the power read by the wattmeter. However, for powers greater than 4.32 kW, the impact is significant. The relationship $\eta_{VFD} = f\left(P_{VFD}\right)$ between the range from 4,32 kW to 6,92 kW is approximated by a linear equation as follows:

$$\eta_{VFD} = -0,0182 \cdot P_{VFD} + 0,9514. \tag{18}$$

Table 1. Consumed electrical power in operation without/with (P_{el} / P_{VFD}) using VFD.

P_{el}, [kW]	P_{VFD}, [kW]	η_{VFD}
3.02	3.02	0.875
3.69	3.69	0.875

4.32	4.32	0.875
4.89	4.98	0.859186747
5.47	5.65	0.847123894
6.54	6.92	0.826950867

- **Determining the fan total coefficient of efficiency**

The total fan coefficient of efficiency represents the ratio between the hydraulic energy provided to the air (or gas) stream by the fan and the mechanical energy supplied to the fan shaft. If power is used instead of energy, the total coefficient of efficiency can be calculated as follows:

$$\eta = \frac{pQ}{1000P} . \tag{19}$$

- **Determining fan static coefficient of efficiency**

The fan static coefficient of efficiency can be determined by using the following equation:

$$\eta_{st} = \frac{p_{st}Q}{1000P} . \tag{20}$$

As well known, fan operating curves graphically display the relationships between the main operating parameters (pressure, power and coefficient of efficiency) and the flow rate: $p = f(Q)$, $p_{st} = f(Q)$, $P = f(Q)$, $\eta = f(Q)$ and $\eta_{st} = f(Q)$, obtained at a constant speed (n=const). During the experimental performance, the speed of the driving electric motor is varied within certain limits. This necessiates using the Affinity Laws to adjust the parameter values to a constant speed:

$$Q = Q_x \frac{n}{n_x} , \ p = p_x \left(\frac{n}{n_x} \right)^2 , P = P_x \left(\frac{n}{n_x} \right)^3 , \tag{21}$$

where the index "x" is used to indicate the parameters obtained at a specific speed n_x.

If needed, the fan operating parameters can be adjusted to standard atmospheric conditions. According to ISO 5801:2007 „Industrial fans - Performance testing using standardized airways", these conditions include: atmospheric pressure $p_0 = 101325 \ Pa$, air temperature $T_0 = 293.15 \ K$ and relative humidity $\phi_0 = 0.40$. Under these conditions, the gas constant of moist air is $R_0 = 288 \ \frac{J}{kgK}$, and its density is $\rho_0 = 1.2 \ kg/m^3$. Referring is done based on the following principles:

$$Q_0 = Q, \ p_0 = p \frac{\rho_0}{\rho_{atm}} , \ P_0 = P \frac{\rho_0}{\rho_{atm}} . \tag{22}$$

Universal operating curves of the fan studied

- **Universal operating curve, obtained at different speeds**

The universal operating curve for the fan at different speeds - $n = 2950, 2600, 2300$ and $2000 \ min^{-1}$, is shown in Fig. 2.

Figure 3 illustrates the relationship between the coefficient of efficiency and flow rate $\eta = f(Q)$ at the same speeds. It is evident from Fig. 3 that the fan being studied operates at its highest coefficient of efficiency $\eta_{max} = 0,6$, at a specific speed $n = 2950 \ min^{-1}$, flow rate $Q = 0,49 \ m^3/s$ and pressure $p = 4420 \ Pa$. As the speed decreases, the maximum coefficient of efficiency also decreases resulting in lower flow rates. At a speed of $n = 2000 \ min^{-1}$ the maximum value of the fan coefficient of efficiency is 0,57, achieved at a specific flow rate

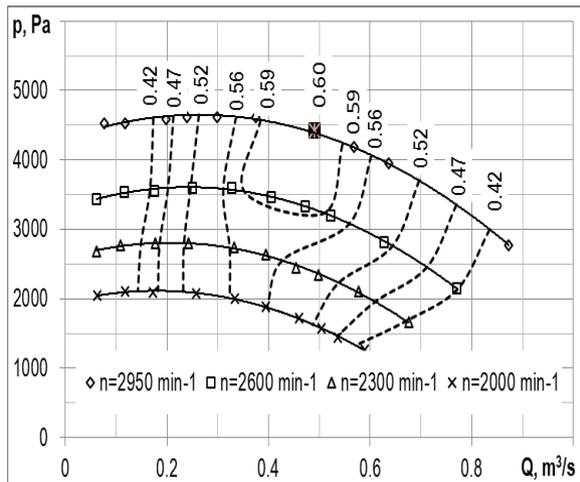$Q = 0,330 \text{ m}^3/\text{s}$ and pressure $p = 2000 \text{ Pa}$.

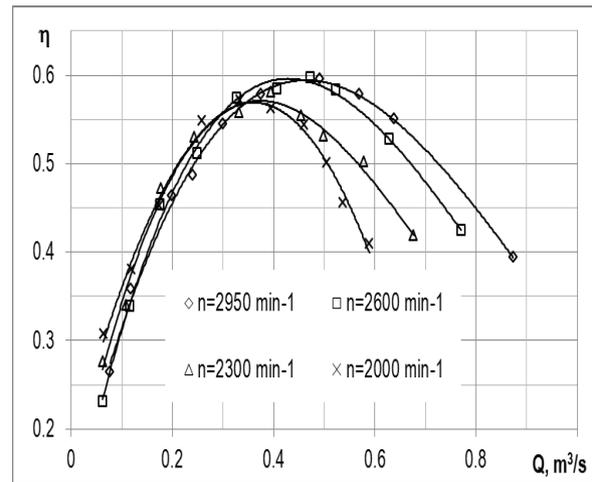Fig. 2. Universal fan operating curves obtained at different speeds.

Fig. 3. Fan coefficient of efficiency obtained at different speeds.

- **Universal fan operating curve plotted at different angles of the inlet guide vane apparatus.**

This curve is obtained at a constant speed $n = 2950 \text{ min}^{-1}$ for different angles of the inlet guide vane apparatus (see Fig. 4). Figure 5 illustrates the relationships of the type $\eta = f(Q)$, obtained at the following angles of the blades of the inlet guide vane apparatus (IGV): $\alpha = 0, 30, 40, 50$ and 60°.
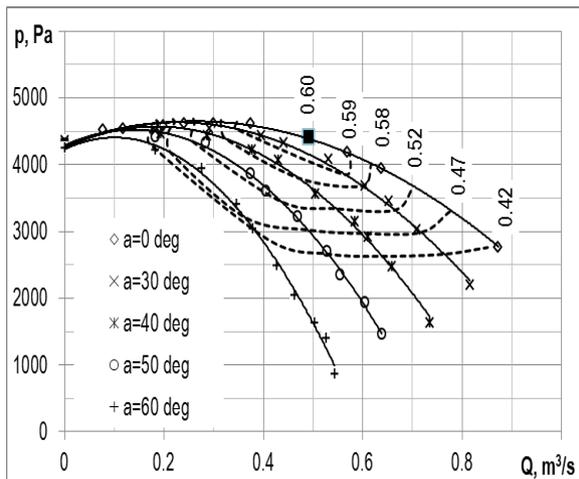

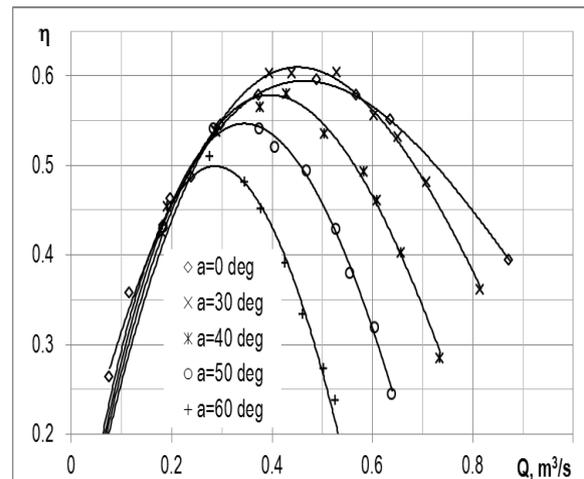Fig. 4. Universal fan operating curves obtained at different angles of the IGV.

Fig. 5. The fan coefficient of efficiency obtained at different angles of IGV.

In Fig. 5, the trend of maximum efficiency decreasing with and increase in IGV angle α is depicted. Additionally, the maximum efficiency value shifts towards lower flow rates as this angle increases. For angle α values up to the maximum 30° efficiency remains high at about 0,6. However, as the angle continues to increase efficiency decreases more rapidly compared to changes in rotation speed. At anggle $\alpha = 60^\circ$ the maximum efficiency coefficient is 0,51, achieved at a flow rate $Q = 0,28 \text{ m}^3/\text{s}$ and pressure $p = 3940 \text{ Pa}$.

Energy efficiency of flow control methods applied to the studied fan system

- **Flow rate regulation through throttling**

Throttling is one of the most commonly used methods for regulating flow rate in ventilation systems. This is due to its simplicity and reliability, as well as the inexpensive devices required for implementation. However, this method is the least efficient in terms of energy efficiency. In modern fan systems, reducing the flow rate by 40% through throttling leads to a reduction in

system efficiency by about 30% for fans with backward-facing blades and about 37% for fans with forward-facing blades. Throttling device can be installed both before and after the fan. If the throttle is located at a sufficient distance before or after the fan, it can be considered an element of the pipe system. In this case, the decrease in flow rate in the system is due to a variation in the pipe system resistance curve when throttling (it becomes steeper). If the throttle is located in close proximity to the fan, then it affects the flow at its inlet. In this case, it can be considered an element of the fan that impacts its operating curve $p - Q$, resulting in a variation of the system operating mode. In the experimental setup used for this study, the throttling device is located at the end of the discharge air duct.

- **Flow rate regulation through utilizing an input guide vane apparatus (IGV)**

This method involves the use of a guide vane apparatus installed before the fan inlet to control the flow rate of a fan system. There are various types of guide vane apparatuses available, with this study focusing on the use of an axial guide vane device due to its superior energy efficiency in regulation. The axial guide device is comprised of a system of blades that can rotate simultaneously around their axes, at the same angle. It is positioned just before the inlet of the impeller. The principle of regulation when using IGV involves creating a transfer component of the absolute velocity υ_{u1} of the flow in front of the impeller inlet, which alters the pressure provided by the fan and, consequently, its operating curve $p - Q$. This can be observed by examining the fundamental equation of working turbomachines:

$$p_T = \rho u_2 \upsilon_{u2} - \rho u_1 \upsilon_{u1}. \tag{23}$$

where: p_T is the theoretical fan pressure; ρ - air density; u_1, u_2 - the transition velocities before the inlet and after the outlet of the fan impeller; υ_{u1}, υ_{u2} - the transitional components of the absolute velocities before the inlet and after the outlet of the fan impeller.

If the direction of the velocity υ_{u1} coincides with the direction of u_1 it decreases, otherwise it increases. This leads to variations in the operating mode of the fan system.

- **Flow rate regulation through variable frequency drive (VFD) control**

This method is the most economical in terms of energy efficiency for regulating the flow rate of a fan system. With the availability of frequency inverters, this method is easily feasible. Its effectiveness depends not only on the efficiency of the fan at different rotation speeds, but also on the efficiency of the electric motor and the frequency inverter at different operating modes.

- **Results obtained after conducting experimental studies**

The main criteria for assessing the efficiency of fan systems is the specific fan power (SFP). This is defined in the European standard EN 13779 and represents the energy consumed to transport a unit volume of gas.

This relative energy parameter is calculated by using the following equation:

$$\text{SFP} = \frac{P}{Q} = \frac{p}{\eta}, \left[J / m^3 \right], \tag{24}$$

where P, Q, p and η are the power, flow rate, pressure and coefficient of efficiency of the fan, respectively. In this study, to assess the energy efficiency of three methods for regulating the flow rate in a fan system, using the parameter $e_{V,el}$, which is determined using the following equation:

$$e_{V,el} = \frac{P_{el}}{Q}, \left[J / m^3 \right], \tag{25}$$

where the experiment also consideres energy loss in the electric drive of the fan-electric motor aggregate and frequency inverter.

Experiments are conducted at four different initial flow rates: $Q_0 = 0,85 \text{ m}^3 / \text{s}$,

$Q_0 = 0,67 \text{ m}^3/\text{s}$, $Q_0 = 0,58 \text{ m}^3/\text{s}$ and $Q_0 = 0,48 \text{ m}^3/\text{s}$. The relationships of the type $e_{V,el} = f(Q)$, graphically presented, concerning the three flow control methods applied at different initial flow rates are shown in Fig. 6. The figures clearly indicate that regardless of the initial flow rate, the power is lowest when regulated using a frequency inverter (VFD), and highest when regulated using throttling. It is also seen that up to a certain point, regulation with a IGV apparatus is equally efficient as in the case of applying VFD flow control method.

In order to obtain a more accurate comparative assessment of the economy of the three methods studied, equations of the type $e_{V,el}^* = f\left(\dfrac{Q}{Q_0}\right)$ are established, where Q represents the current flow rate value obtained by applying one of the flow control methods studied, Q_0 - the initial flow rate, and $e_{V,el}^*$ represents determined by using one of the following equations:

$$e_{V,el}^* = \frac{e_{V,el,IGV}}{e_{V,el,TH}}, \tag{26}$$

$$e_{V,el}^* = \frac{e_{V,el,VFD}}{e_{V,el,TH}}, , \tag{27}$$

where $e_{V,el,IGV}$ is the parameter, determined by using eq. (25) when regulating the flow rate using an input guide device apparatus; $e_{V,el,TH}$ - the same parameter, apllies when using throttling; $e_{V,el,VFD}$ - the same parameter, apllies when using VFD flow control.

The relationships of the type $e_{V,el}^* = f\left(\dfrac{Q}{Q_0}\right)$ are presented graphically in Fig. 7. The analysis of these figures indicates that VFD and IGV flow control methods are equally efficient in reducing the flow rate by up to 5% of the initial flow rate Q_0. The electrical power consumed when regulating with IGV flow control method is about 25% less than in the case of throttling, when the flow rate is reduced by 40% of the initial rate. When regulating using a frequency inverter, the power consumption is up to 60% lower than in the case of throttling.

a). Q_0=0,85 m³/s

b). Q_0=0,67 m³/s

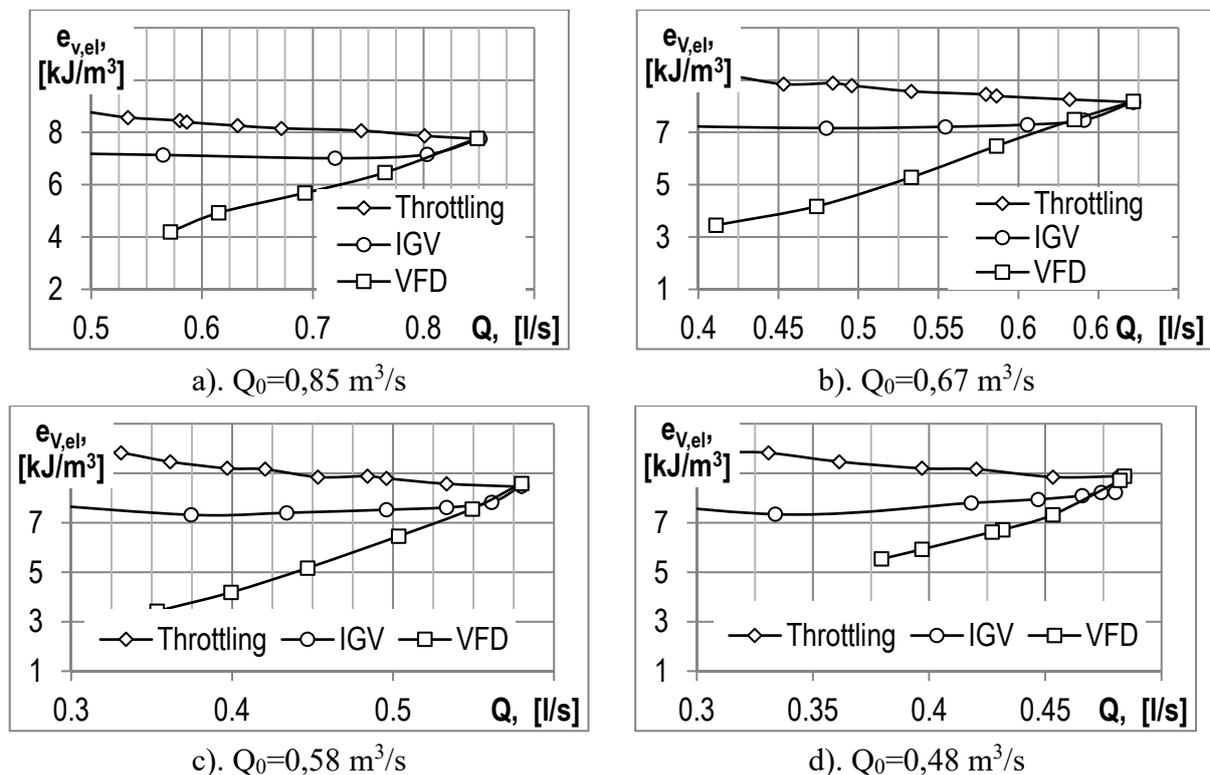c). Q_0=0,58 m³/s

d). Q_0=0,48 m³/s

Fig. 6. Relationships between $e_{V,el}$ and the fan system flow rate, obtained for the three methods of flow rate regulation applied at different initial flow rate Q_0.

a). $Q_0 = 0,85$ m^3/s

b). $Q_0 = 0,67$ m^3/s

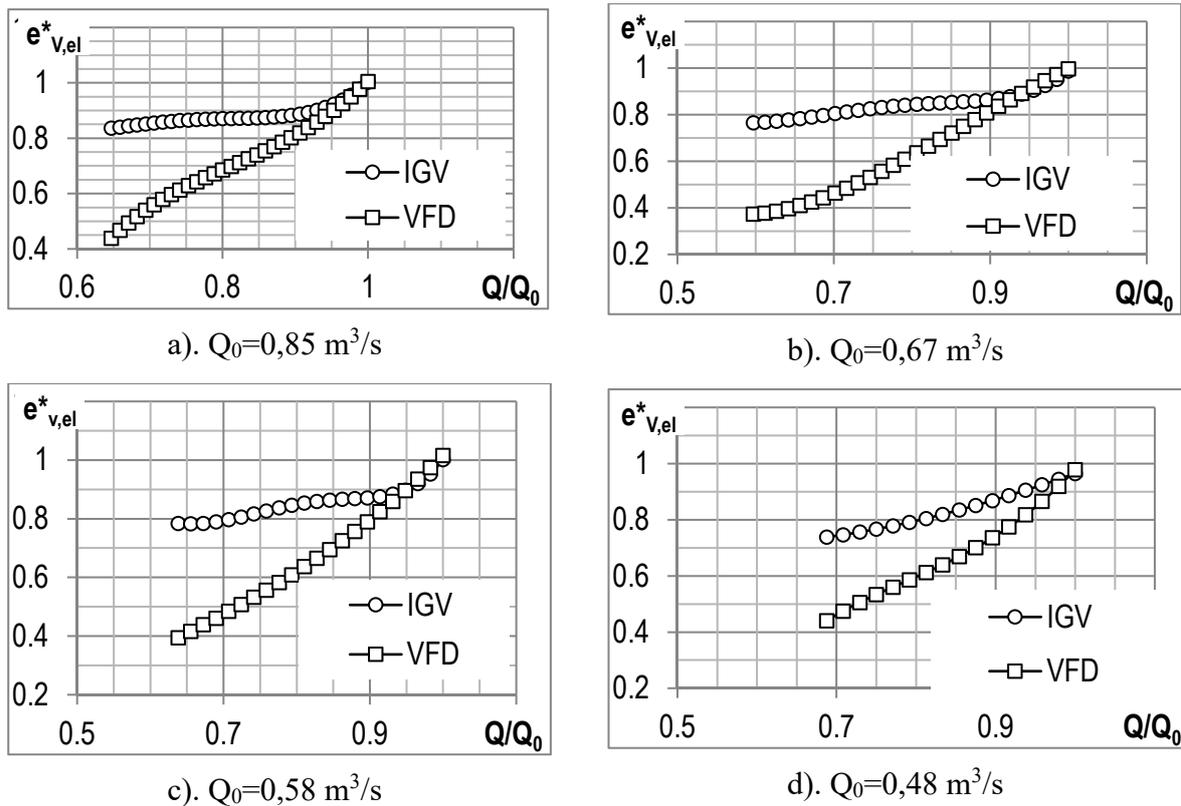c). $Q_0 = 0,58$ m^3/s

d). $Q_0 = 0,48$ m^3/s

Fig. 7. Relationships between $e_{V,el}^*$ and the relative fan system flow rate using IVG and VFD flow control methods applied at various initial flow rates.

To evaluate the impact of the initial operating mode on the effectiveness of the applied flow control method, the relationships of the type $e_{V,el}^* = f\left(\dfrac{Q}{Q_0}\right)$ are graphically plotted in separate coordinate systems and shown in Fig. 8-a when using an IGV apparatus and in Fig. 8-b when using a VFD. The impact of the initial fan operating mode on the IGV apparatus method of flow rate regulation is considered more significant. The analysis of the figure clearly indicates that at an initial flow rate $Q_0 = 0,48$ m^3/s, power decreases by up to 25% compared to throttle control, when the flow rate is reduced by 30% of the initial value. At an initial flow rate $Q_0 = 0,85$ m^3/s the power reduction compared to throttle flow control is about 15% at $Q = 0,7Q_0$.

a). IGV flow control
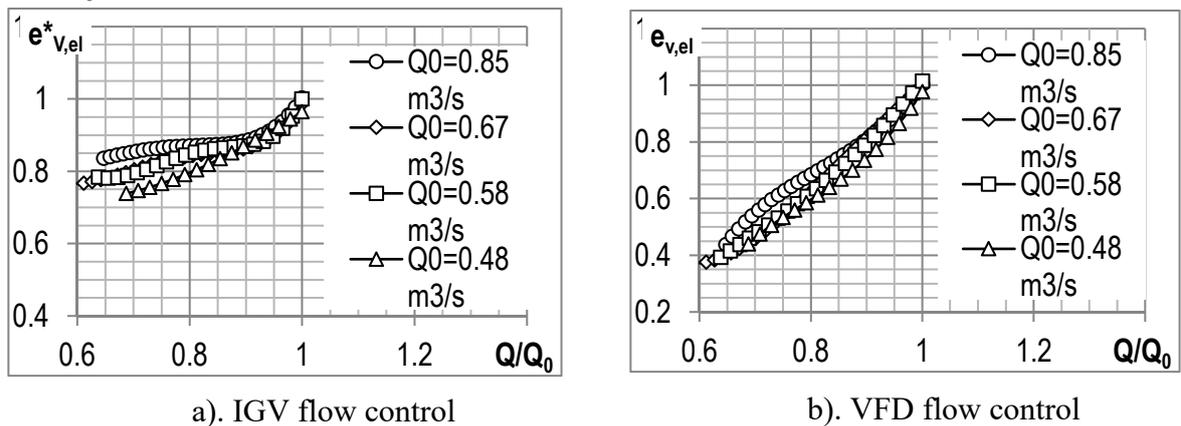
b). VFD flow control

Fig. 8. Relationships of the type $e_{V,el}^* = f\left(\dfrac{Q}{Q_0}\right)$ obtained in applying IGV and VFD flow control methods.

When regulating with a frequency inverter (VFD), the impact of the initial fan system operating mode on energy efficiency is insignificant.

CONCLUSION

Based on the experimental study conducted, the following important conclusions can be made:

- Both VFD and IGV flow control methods are equally effective when reducing the flow rate by up to 5% of the initial flow rate Q_0.

- The electrical power consumption is approximately 25% lower when using IVG flow control compared to throttling when the initial flow rate is reduced by 40%.

- When regulating with a frequency inverter (VFD), power consumption can be up to 60% lower than when using throttling.

- The impact of the initial fan system operating mode on the regulation of flow rate using IGV apparatus is more pronounced. However, when regulating using the VFD flow control method, the impact of the initial fan operating mode on the system energy efficiency is insignificant.

ACKNOWLEDGEMENTS

REFERENCES

Bleier, F. P. Fan handbook. Selction, applicationand design. The McGraw-Hill Companies 1998.

Bruk, A. D., Matikashvili, T. I., Nevelson, M. I., Raer, G. A., Solomahova, T. S. & Judin, E. J. (1975). Centrobezhnaye ventiljataray. Pod red. na T. S., Solomahova (avt.), Mashinostroenie, Moskva. (***Оригинално заглавие:** Брук, А. и др. Центробежные вентиляторы. Москва. Машиностроение*).

EN 13779: 2006 Ventilation for Non-Residential Buildings - Performance Requirements for Ventilation and Room–Conditioning System.

ISO 5167-2:2003 „Measurement of fluid flow by means of pressure differential devices inserted in circular cross-section conduits running full - Part 2: Orifice plates".

ISO 5801:2007 „Industrial fans - Performance testing using standardized airways"

Kostov, B., G. Popov, Kl. Klimentov. Energy Efficiency Of Pump Systems (2014). Agricultural machinery, Year LX, Vol.1 Sofia, 2014, ISSN 0861-9638.

Popov, G., K. Klimentov, B. Kostov, L. Kamenov (2023). INFLUNECE OF THE DIFFUSER LOCATION ON THE VENTILATION SYSTEM CURVE 62nd Annual Science Conference of Ruse University and Union of Scientists - Ruse, 115-121.

Popov G., K. Klimentov, B. Kostov, L. Kamenov (2023). Methods Used to Increase the Energy Efficiency of Ventilation Systems Механика на машините / Mechanics of Machines, 129, 99-104.

Popov, G., K. Klimentov, B. Kostov (2019). Investigation of the energy consumption in regulating the flow rate of fan systems. 2019 IOP Conference Series: Materials Science and Engineering 595 (2019) 012012.

Popov, G., K. Klimentov, B. Kostov. Investigation of the energy consumption in regulating the flow rate of pump systems (2011). DEMI'2011, Banja Luka (Bosnia), 2011, pp. 481-487.

Popov, G., K. Klimentov, B. Kostov. Methods to estimate the energy consumption in regulating the flow rate of pump systems (2011). DEMI'2011, Banja Luka (Bosnia).

Popov, G., K. Klimentov, B. Kostov. Analytical presentation of the Dimensionless Characteristics of centrifugal fans (2018). 57th Science Conference of Ruse University, Bulgaria, 2018. (***Оригинално заглавие:*** *Попов Г., Кл. Климентов, Б. Костов. Аналитично представяне на безразмерните характеристики на центробежни вентилатори (2018). 57 Научна конференция на Русенски университет. 2018*).

Solomahova, T. S. & Chebaysheva, K. V. (1980). Centrobezhnaye ventiljataray. Aerodinamicheskie shemay i harakteristiki. Mashinostroenie, Moskva. (***Оригинално заглавие:*** *Соломахова, Т., Чебышева, К. В. Центробежные вентиляторы. Аэродинамические схемы и характеристики. Москва. Машиностроение*).