Determining the optimal working point and the loading characteristic of synchronous machine for fan and pump drive system

Lech Nowak, Krzysztof Kowalski, Dorota Stachowiak, Jacek Mikołajewicz,
Wojciech Pietrowski, Kazimierz Radziuk
Poznań University of Technology
60 - 965 Poznań, ul. Piotrowo 3A, e-mail: lech.nowak@put.poznan.pl

In the paper the computer software for determining the optimal working point of electric motors for fan (and pump) drive systems is presented. The main use of this software is the preliminary designing of permanent magnet synchronous motor for such type of drives. The second task of the software is determining the load characteristic of the fan system (pump system), i.e. relationship between loading torque and angular speed. The obtained characteristics can be used in the designing of control system for fan or pump drives.

1. Introduction

Electric drives for ventilation and drainage systems used in mining industry represent significant part of all electric drives. Thousands of electric motors with power ranging from just a few kilowatts to several megawatts are used in the ventilation systems. The second most power consuming systems are the drainage systems in which mainly the centrifugal pumps are employed pumps. Overall power usage of these systems in the mining industry in Poland can be counted in gigawatts. Proper design of the drives and the control systems is crucial to minimize energy consumption in this sector.

Asynchronous drives are usually used in these two systems, with the exception of the most powerful fan drive applications, where synchronous motors are employed. It can be observed, however, that the nowadays the tendency to use synchronous motors for the less power demanding applications is becoming more popular. Permanent magnet synchronous motors using high-energy magnets made with neodymium-iron-boron (ND-Fe-B) alloy, are particularly effective in these applications. The constant angular speed ensures constant air or liquid flow rate in any given period of time. Such drives have also higher power coefficient and have grater energy efficiency due to lack of power loss in the rotor.

Use of the variable-speed drives is the most efficient way to control the air or liquid flow. Control systems for synchronous motors are much simpler than those used to control asynchronous drives.

Prior to design of efficient synchronous motor and its control system it is necessary to determine load characteristics of fan or pump, i.e. the relation between its angular speed.

2. Determination of the working point of the ventilation system

The rated flow rate Q_N [m^3/s] at rated speed n_N is the basic parameter characterizing the fan or pump.

Value of the fan flow rate demand and characteristic of the ventilated system $Q = Q(\Delta p)$, where Δp is the differential pressure, must to be given prior to the fan system design process. The aerodynamic resistance of the ventilated objects strongly depends on the rate flow Q, therefore the characteristic $Q = Q(\Delta p)$ is strongly nonlinear. This characteristic is usually presented in an inverted coordinate system, i.e.: $\Delta p = f(Q)$ – Fig 1.

The characteristic of the turbo-machine (fan or pump) is also given in the same coordinate system. The pressure difference between the two sides of the fan decreases as a result of decreasing aerodynamic resistance of the ventilated system, i.e. with the flow Q increase. Fan characteristics $\Delta p = f(Q)$ is decreasing function – Fig. 1.

The point of intersection of these two characteristics (point K_N in Fig. 1) determines the working conditions of the system fan-ventilated object, i.e. system source-load.

Fan power output is equal to the product of differential pressure and flow rate values: $P_w = \Delta p \cdot Q$ [5]. If $Q = 0$ or $\Delta p = 0$ then the power $P_w = 0$. This means, that there is an optimal working point $K_{opt}(Q_{opt}, \Delta p_{opt})$ with the maximum fan output power – Fig. 2.

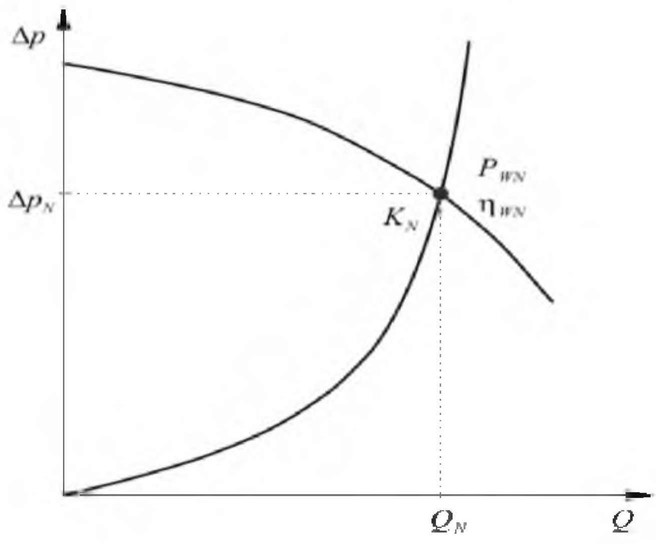

Fig. 1. Determination of the optimal working point of the system: fan-ventilated object

The motor rated power can be expressed as follows:

$$P_N = \frac{P_{wN}}{\eta_{wN}} = \frac{\Delta p_N \cdot Q_N}{\eta_{wN}} \tag{1}$$

and its rated torque:

$$T_N = \frac{P_{wN}}{\omega_N \cdot \eta_{wN}} = \frac{Q_N \cdot \Delta p_N}{\omega_N \cdot \eta_{wN}} \tag{2}$$

where ω_N, η_{wN} are the rated values of angular speed and efficiency, respectively. The rated working point of selected fan should correspond to the maximum output power.

Rated power and rated torque of the driving electric motor can be determined on the basis of the relations (1) and (2).

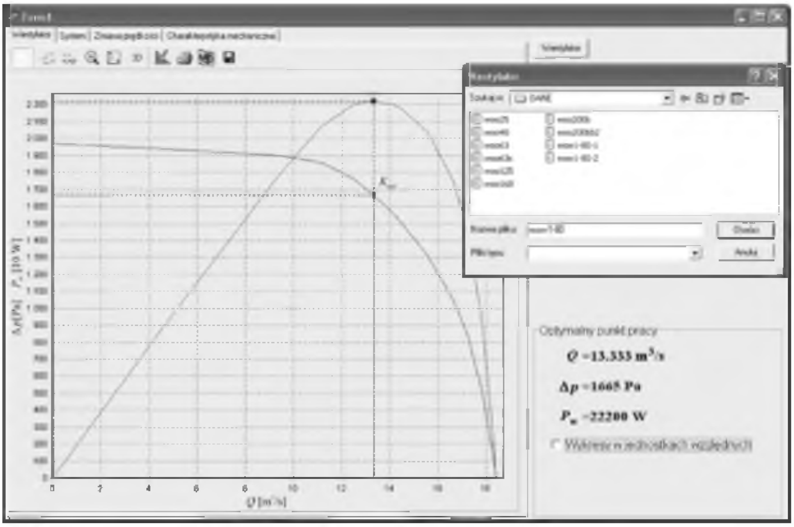

Fig. 2. Determination of the fan characteristics

3. Determination of load characteristics

These considerations apply to the fan motors with adjustable flow rate achieved by control of the motor angular speed. Flow rate of the fan at constant differential pressure is proportional to the speed and differential pressure at constant flow rate is proportional to the square of the speed [1, 5], i.e.:

$$Q = \frac{\omega}{\omega_N} Q_N \qquad \Delta p = \left(\frac{\omega}{\omega_N}\right)^2 \Delta p_N \tag{3}$$

Fan characteristic and optimal working point of the system for any angular speed that differs from rated value, can be calculated on the basis of relation (3).

Figure 3 shows the set of the fan characteristics for $\omega = \omega_N$ and ω = 0,9; 0,8; 0,7; 0,6 ω_N. Points of intersection of these curves and the characteristic of the ventilated object determine operating states (conditions) of the system at relevant speed values.

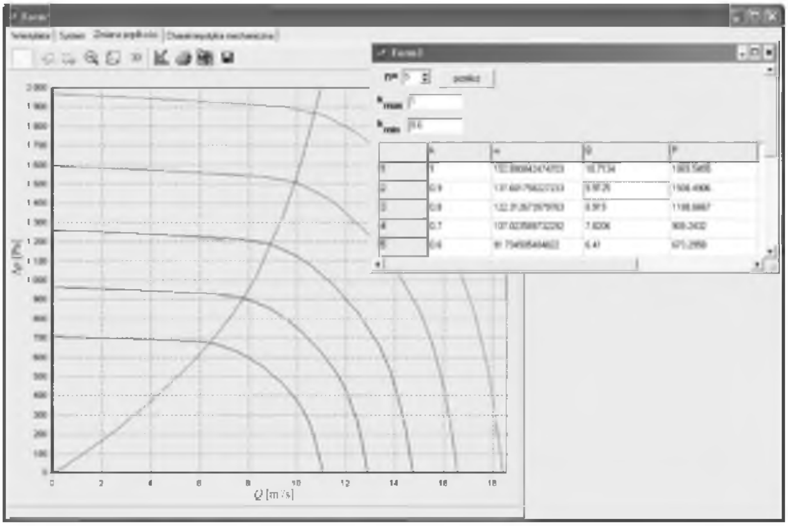

Fig. 3. Determination of characteristics for different angular speeds

Assuming that the aerodynamic resistance is proportional to the speed and efficiency of the fan is speed independent than equations (3) show that the torque is proportional to the square of speed. In the real system, however, flow rate and differential pressure values change with variations of the angular speed of the fan. Relationship of these two parameters is highly nonlinear in relation to both: fan and ventilated system [1, 5].

In order to determine the fan generated full-load characteristic $T_m(\omega)$ of the driving motor, using the Borland-Delphi programming environment, the computer software has been developed. The software allows to:

– enter data (e.g. from file) describing the characteristics of the fan and the system – Fig. 2 and 3;
– determine fan characteristics $\Delta p = f(Q)$, $P_w = f(Q)$ and calculate an optimal operating point – Fig. 2;
– determine set of fan characteristics for a selected range of angular speed and compute the set of operating points $\{Q_i, \Delta p_i\}$ – Fig. 3;
– determine the load characteristic $T_m(\omega)$ – Fig. 4.

Points (ω_i, T_{mi}) of load characteristics are calculated according to the formula:

$$T_{mi} = \frac{Q_i \cdot \Delta p_i}{\omega_i} + m_0 \frac{Q_N \cdot \Delta p_N}{\omega_N} + \beta_t \frac{Q_N \cdot \Delta p_N}{\omega_N} \left(\frac{\omega_i}{\omega_N} \right)^q \tag{4}$$

where $m_0 = \omega_N T_0 / Q_N \cdot \Delta p_N$ – relative static friction torque, $\beta_t = \eta_{wN}^{-1} - 1 - m_0$ – relative friction coefficient, q – exponent characterizing dependence of friction torque on speed.

An exemplary load characteristic constructed on the basis of equation (4) is shown in Fig. 4.

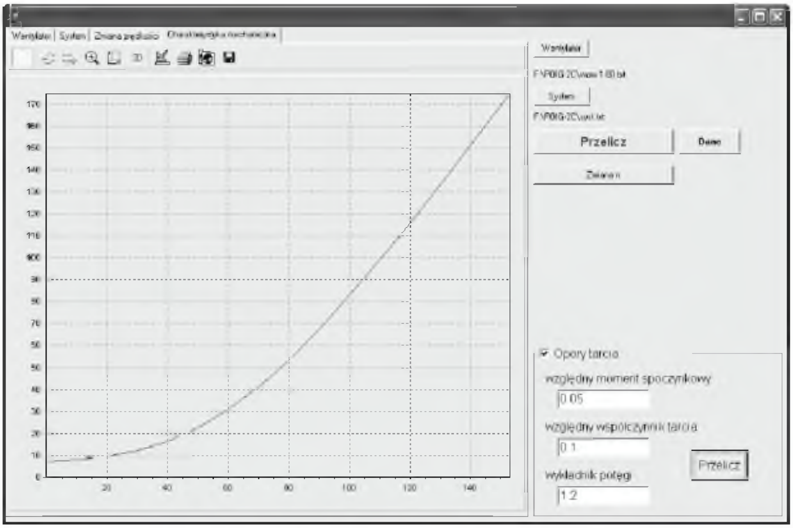

Fig. 4. Load characteristics $T_m(\omega)$

4. Characteristics of impeller pumps

In the mining industry mainly the impeller (rotodynamic) pumps are employed. In the underground mines the pumps are used for drainage of mine faces and drawing levels. Drainage pumps up to 6 kW are used on the rock walls, pumps up to 20 kW are used for dewatering of mine faces. Units up to 90 kW are used for pumping water between the intermediate levels. Pumps with power output of more than 1 MW are used to extract water directly onto the surface (even more than 600 m). In strip mines pumps are used to active drainage and boreholes dewatering [3].

Operation of the impeller pump is very similar to the axial flow fan operation. It crates differential pressure between the inlet (suction) and the outlet (discharge) of the pump. A characteristic aspect of the centrifugal pump is reverse flow of liquid after the pump stops, caused by e.g. water column pressure. In such case the pump start-up is often done at considerable high initial load and therefore, the driving motor must have high starting torque. On the other hand, the inertia of the rotating parts of the pump is noticeable smaller than inertia of the comparable capacity fan and in this respect the pump star-up process is less demanding.

The basic parameter characterizing the pump is its rated output capability (i.e. flow rate) Q_N [m^3/s] at rated speed ω_N. The other key parameters of the pump are: pressure expressed in height H of the water column (called the pump pressure head or lifting head or elevating head or delivery head), power P and efficiency η. They are presented as a function of the pump flow rate Q and are called pump characteristics. The most important characteristic is dependence of the pump lifting head H on the flow rate Q, at constant angular speed. It is so called the throttle curve. This characteristic can be stable (ACD curve, in Fig. 5) or non-stable (BCD curve in Fig. 5). Stable characteristic is monotonically descending curve and maximum of the pressure head H occurs at flow rate $Q = 0$. In the case of non-stable characteristics, the same value of pressure head H corresponds to two values of flow rate Q.

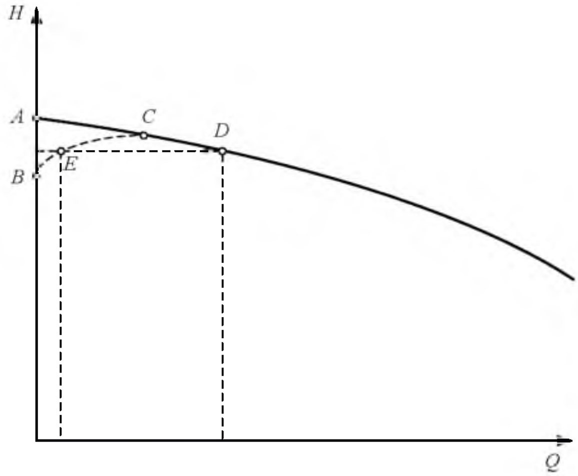

Fig. 5. Flow rate-pressure head characteristic

The pump systems (installation) consist of numerous components. Hydraulic losses are caused by viscous friction between the liquid and the installation. System is characterized by the dependence of hydraulic losses (expressed in meters of height of water column) on the flow rate. These losses depend on the diameter and length of pipes, type and dimension of valves and pipe fittings and, of course, on the velocity of flow. The operating point of the system is located at the intersection of pump and installation characteristics – Fig. 6. Hydraulic losses ΔH_r represent sum of losses ΔH_l on the sections of straight elements (pipes) and losses ΔH_m on other local obstacles (valves, elbows, etc). The pump pressure head H_c have to be equal to hydraulic losses an can be expressed by the following equation [4]:

$$H_c = \Delta H_r = \Delta H_l + \Delta H_m = \lambda \frac{l}{d} \frac{c_{sr}^2}{2g} + \sum \zeta_m \frac{c_{sr}^2}{2g} \tag{5}$$

were: λ is the dimensionless hydraulic resistance coefficient; l, d are is the length and diameter of linear-elements, respectively; c is the flow velocity, g is the gravitational acceleration; ζ_m is the dimensionless local hydraulic resistance coefficient dependent on Reynolds number, type of obstacles, inner surface roughness and valve opening degree.

After substituting $c_{sr} = 4Q/\pi d^2$, equation (5) can be written as:

$$H_c = \left(\lambda \frac{l}{d} \frac{1}{2g} + \frac{\sum \zeta_m}{2g} \right) c_{sr}^2 = \frac{8}{\pi^2 g d^4} \left(\lambda \frac{l}{d} + \sum \zeta_m \right) Q^2 = CQ^2 \tag{6}$$

where constant C is the characteristic coefficient of the installation, which is equal to:

$$C = \frac{8}{\pi^2 g d^4} \left(\lambda \frac{l}{d} + \sum \zeta_m \right) \tag{7}$$

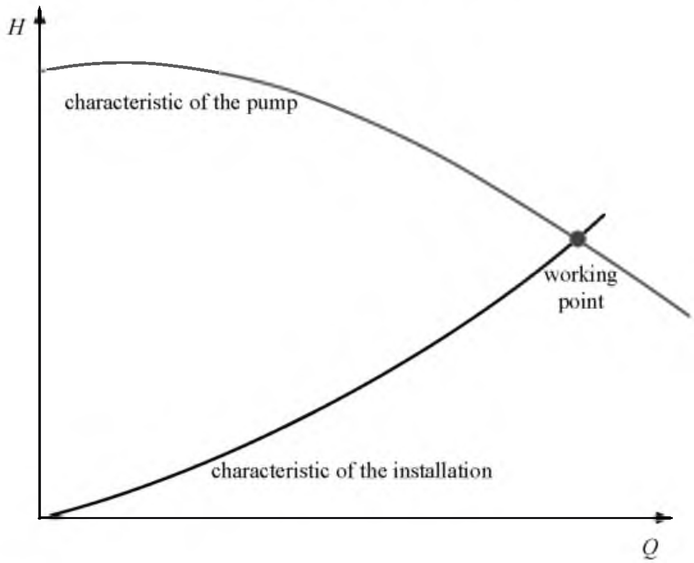

Fig. 6. Determination of operating point of the pump-installation

If there is difference in level H_g between the beginning and the ending of the installation, than:

$$H_c = H_g + \Delta H_r = H_g + CQ^2 \tag{8}$$

Characteristics of the installation and the pump are shown in Fig. 7.

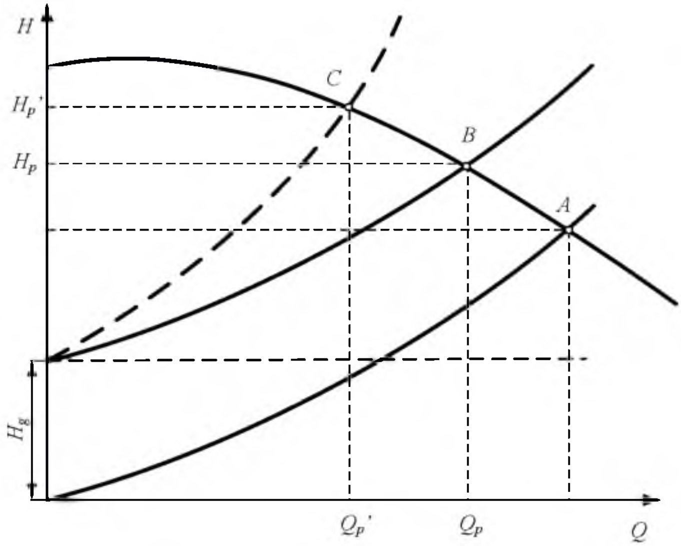

Fig. 7. Optimal working point of the system pump-installation, at constant speed

Point B of intersection of these curves is the optimal working point of coordinates Q_p, H_p. If the hydraulic resistance of installation will increase, e.g. due to the pipe bend or connecting smaller diameter pipe, the optimal working point moves to point C. The pump lifting head will increase to the value H'_p and its flow rate will decrease to the value Q'_p – Fig. 7.

5. Determination of the rated torque and pump generated load characteristics of the driving motor

Useful output power P_p of the pump is equal to [2,4]:

$$P_p = \gamma \cdot Q \cdot H \tag{9}$$

where γ is the specific gravity of liquid. If $Q = 0$ or $H = 0$, then $P_p = 0$. This means, that there is an optimal working point point (Q_{opt}, H_{opt}), in which the impeller pump has a maximum power. Rated torque of the driving motor can be expressed as:

$$T_N = \frac{P_{pN}}{\omega_N \cdot \eta_{pN}} = \frac{\gamma \cdot Q_N \cdot H_N}{\omega_N \cdot \eta_{pN}} \tag{10}$$

where ω_N, η_{pN} are the rated angular velocity and rated pump efficiency, respectively.

Pump, in respect to the entire system, should be selected in such a way, that the nominal working point should close to the optimal point, which in turn corresponds to the maximum power output and the maximum efficiency of the pump.

The pump flow rate (at constant pressure head) is proportional to the angular speed and pressure head (at constant flow) is proportional to square of the speed [2, 4]. On that basis, the pomp characteristics and the optimal working point for any speed ω different than the rated value ω_N can be determined. Fig. 8 shows the set of characteristics for $\omega = \omega_N$ and $\omega_1 < \omega_N$; $\omega_2 < \omega_1$. Points K_N, K_1, K_2, define the states of the system relevant to these speeds.

In the real systems the pump flow rate and pump pressure head vary freely with the change of the pump speed. Relationship between these parameters in respect to both: the pump and the installation is highly nonlinear.

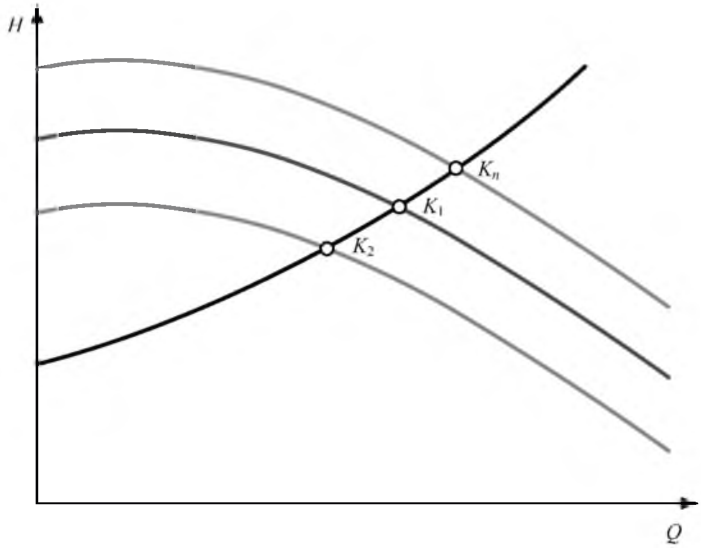

Fig. 8. Effect of the angular speed on the shape of the pump flow (rate) characteristics *H*=*f*(Q)

In order to determine the load torque and load characteristic $T_m(\omega)$ generated by the pump the computer software has been developed. The software allows to: enter data describing the pump and the installation, determine the optimal working point of system, determine the set of pump characteristics for selected range of speed (Fig. 9), determine the set $\{Q_i, H_i\}$ of optimal working points for different speeds ω_i and corresponding values of torque:

$$T_{mi} = \frac{\gamma \cdot Q_i \cdot H_i}{\omega_i \eta_{pi}} \tag{11}$$

The points calculated according to (11) allow to determine load characteristics $T_m(\omega)$ – Fig. 10. The expression (11) is valid only for angular speeds ω_i greater then critical value ω_g, which corresponds to the point of intersection of the pump and installation curves at $Q = 0$.

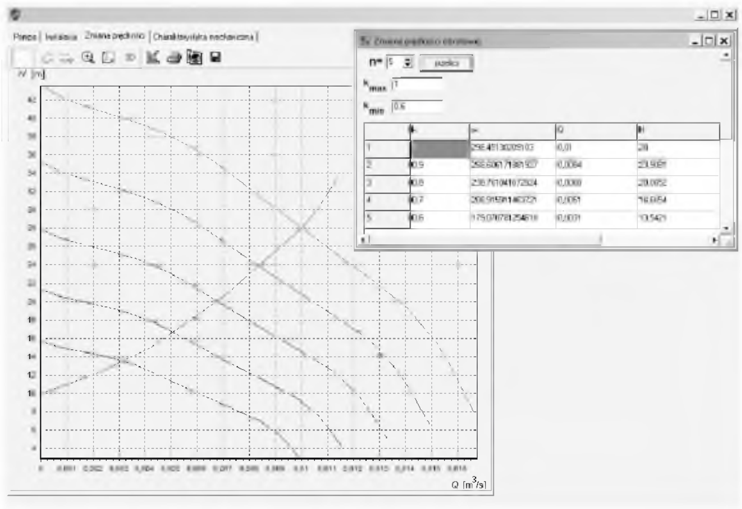

Fig. 9. Determination of the set of characteristics for various speeds

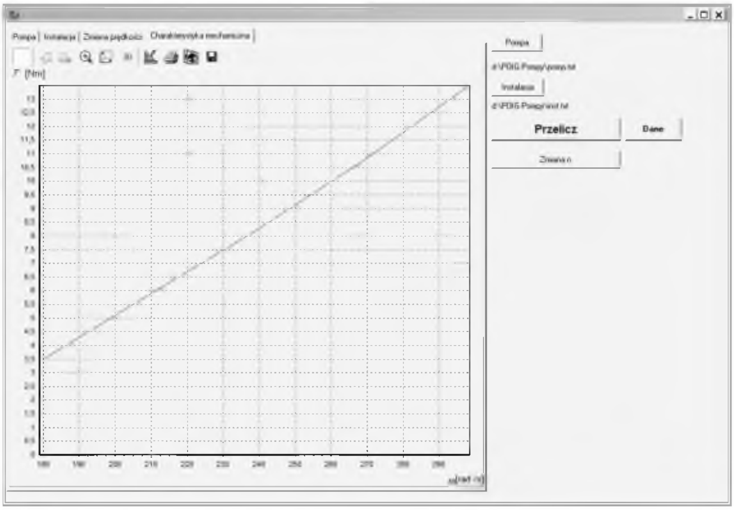

Fig. 10. Determination of load characteristics $T_m(\omega)$ for freely various flow rate and pressure head

After selecting the pump which ensures the desired flow at rated speed, the motor rated torque can be determine according to (10). Computed load characteristic (shown in Fig. 10) can be the basis for the variable speed drive design.

6. Conclusions

The developed software allowing to determine the optimal working point of fan or pump system can be used in the design of efficient driving motors. It also allows to determine the load characteristics, which can be used in the design of the engine control system.

References

[1] Gundlach W. R., Fundamentals of turbo machinery and energy systems (in Polish), WNT, Warszawa, 2008.
[2] Jędral W., The impeller pumps (in Polish), PWN, Warszawa, 2001.
[3] Katalog firmy Flygt: Drainage pumps. Pumps for the mining industry (in Polish), http://www.ittwww.pl/, 06.05.2010.
[4] Stępniewski M, Pumps (in Polish), WNT, Warszawa, 1985.
[5] Szklarski L., Skalny A., Strycharz J., Dynamics and control of stationary electric drives in mining (in Polish), PWN, Warszawa, 1984.