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THE IMPLEMENTATION OF CAD AND CFD TECHNIQUES IN THE DESIGN OF A CENTRIFUGAL VENTILATOR WITH RESTRAINTS OF FORM AND SIZE

Teodor MILOŞ, Associate Professor *
Department of Hydraulic Machinery
"Politehnica" University of Timisoara

Romeo F. SUSAN-RESIGA, Professor
Department of Hydraulic Machinery
"Politehnica" University of Timisoara

Liviu Eugen ANTON, Professor
Department of Hydraulic Machinery
"Politehnica" University of Timisoara

Alexandru BAYA, Professor
Department of Hydraulic Machinery
"Politehnica" University of Timisoara

Sebastian MUNTEAN, Senior Researcher
Center of Advanced Research in Engineering Sciences
Romanian Academy – Timişoara Branch

Adrian STUPARU, PhD Student
Department of Hydraulic Machinery
"Politehnica" University of Timisoara

**Corresponding author*: Bv Mihai Viteazu 24, 300223, Timisoara, Romania
Tel.: (+40) 256 403683, Fax: (+40) 256 403682, Email: milost@home.ro or mh@mec.utt.ro

ABSTRACT

The cooling of industrial equipments through ventilation in closed circuit leads to special conditions of the size and rotational speed regarding the construction of ventilators to which it adds the constructive and technological simplifications. The designing of this kind of a machine and the analysis of its functioning in the context of an imposed exterior circuit derives in a difficult problem. Using CAD software developed by us and the FLUENT code for analysis it obtains results which lead to optimization of the functioning of the ventilation.

KEYWORDS

Centrifugal ventilator, operating characteristic determination, operating point of a cooling system

NOMENCLATURE

Q	[m^3/s]	flow rate
T	[K]	absolute temperature
R	[J/kg·K]	gas constant
p_{vap}	[Pa]	pressure of the saturated vapours of the water at 60°C
η_o	[Pa·s]	dynamic viscosity of the gas at 0°C
C	[-]	gas constant
n	[rot/min]	rotational speed
ω	[rad/s]	angular velocity
n_q	[-]	characteristic speed
n_s	[-]	specific speed
ω_{0ad}	[-]	characteristic angular velocity
η	[%]	probable efficiency
H	[m]	pumping head
h_p	[m]	losses
Δp	[Pa]	pumping pressure

1. INTRODUCTION

Tacking into account the available computing resources, professional software applications and our expert software applications, the design of a hydraulic or pneumatic machine can easily incorporate modern numerical techniques. Most of the fundamental relations and of the statistical relations remain valid and are inserted in our software applications. The grapho-analytical methods became pure analytical, by using modern method of computation like the Finite Element Method (EFM) or making interpolations on data bases for the dependence between some of the given values.

The results of the designing calculus have to materialize in the obtaining of the operating characteristic for the determination of the domains and operating points for the proposed purpose especially for the conditions when installation where the ventilator operates has also many geometrical and functioning restraints. This paper realizes the designing of a centrifugal ventilator of special construction and analysis its functioning in an exterior circuit using CAD and CFD techniques. This application it is referring to the cooling of an electrical motor with variable rotational speed.

The restrains of form are imposed especially by the technological execution process at a low cost through minimum mechanical processing and welding assembly. Reduced rotational speed, restrains of size and the constructive simplifications limit severely the degree of freedom of the designer so that the ventilator to realize the hydrodynamic parameters needed for the cooling of the electrical motor. The only elements on which can be made changes are the number of blades, their form and the width of the runner (through which the flow is controlled).

2. COMPUTATIONAL DESIGN OF THE ROTOR BLADES RESPECTING ONLY THE RESTRAINTS OF SIZE

From the available data regarding the thermal loss power in the electrical motor and from the analysis made, resulted that the worse case operating condition of the electrical motor take place at the minimum rotational speed of 215 rot/min. The estimated operating parameters of the ventilator are:

- maximum flow rate: $Q = 3.15\ m^3/s$
- static pressure: $\Delta p = 10$ mm H_2O

The air heated in the zones with thermal flux being the working fluid, its density ρ_a and its viscosity ν_a depends on the temperature, humidity and atmospheric pressure. From the analyses made results a medium temperature, t = 60°C, atmospheric pressure for local zone, p_{at} = 760 mm Hg, and the atmospheric humidity χ = 30 % (it is considered that the interior air is less influenced by the exterior humidity after a few minutes of functioning). The equation for the density is:

$$\rho_a = \frac{p_{at}}{R \cdot T} \cdot \left(1 - \chi \cdot \frac{p_{vap}}{p_{at}}\right) \cong 1 \frac{kg}{m^3} \qquad (1)$$

The dynamic viscosity of the air, η_a, it computes with Sutherland's relation, [], from which then results the cinematic viscosity, ν_a, function of the density:

$$\eta_a = \eta_0 \frac{T_0 + C}{T + C} \left(\frac{T}{T_0}\right)^{\frac{3}{2}} = 1.995 \cdot 10^{-5} \text{ Pa} \cdot \text{s} \qquad (2)$$

The cinematic viscosity results from the equation:

$$\nu_a = \frac{\eta_a}{\rho_a} = 1.999 \cdot 10^{-5} \cong 2 \cdot 10^{-5} \ \frac{\text{m}^2}{\text{s}} \qquad (3)$$

Using equations of the characteristic functions and a statistical equation for the probable efficiency η it obtains the results, together with the initial data like in table 1 and 2:

Table 1

Q [m^3/s]	Δp [mm H_2O]	n [rot/min]	ω [rad/sec]
1.92	10	215	22.51

Table 2

n_q	n_s	ω_{0ad}	η [%]
52.98	193.37	1.0008	67

The configuration of the runner domain in a meridian plane is presented in fig.1.

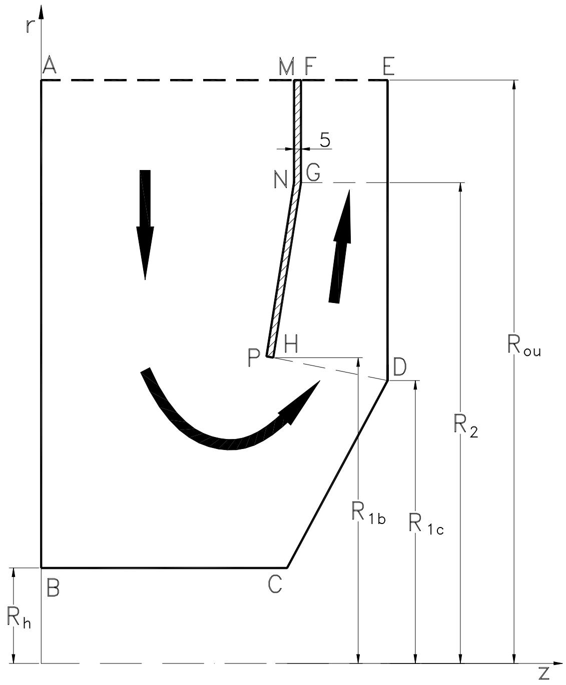

Fig. 1. The upstream domain and the runner domain in a meridian plane

3. THE AERODYNAMIC DESIGNING OF THE RUNNER

For the domain in fig. 1 it was numerical simulated the potential flow with the method of finite element. It resulted the structure of the flow and the variations of the velocities along the streamlines, like shown in fig. 3.

Initially the calculus was performed on 11 streamlines, and then observing that the differences between the shroud and hub are relatively small it was accepted the usual simplification for ventilators: the blade is straight and curved along a circular arc between the inlet and outlet. The results are presented in fig. 4.

The geometry of the runner blades is shown in fig. 5.

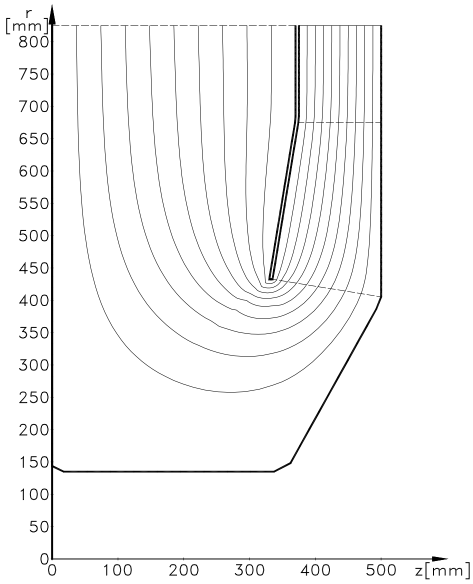

Fig. 2 Streamlines upstream of the runner and in the runner

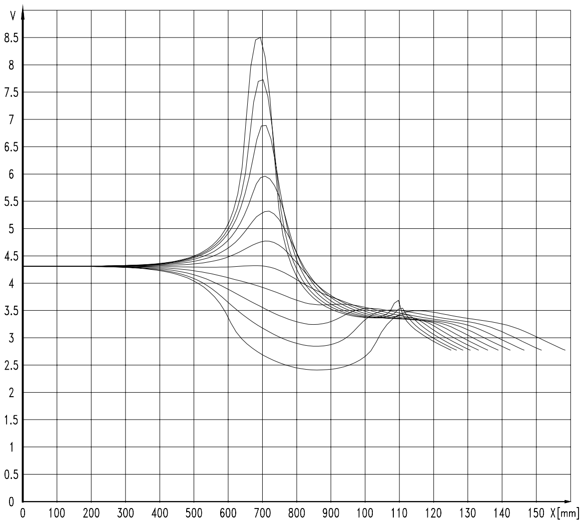

Fig. 3. The variation of the meridian velocities along the streamlines (only in the runner)

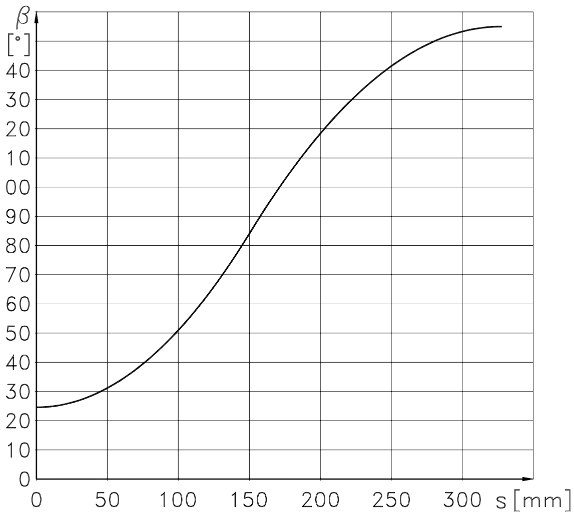

Fig. 4 The variation of β angle of the blade between inlet and outlet

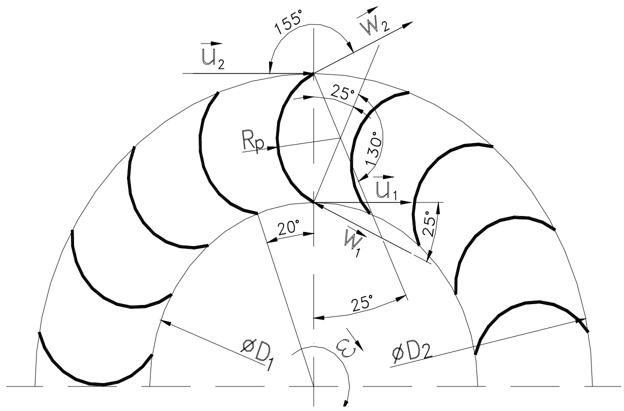

Fig. 5. The cinematic and angular elements of the runner blades

4. THE PREDICTION OF THE OPERATING CHARACTERISTIC OF THE RUNNER

4.1. Theoretical operating characteristics

After the determination of the cinematic and geometric elements of the runner with normal inlet ($\alpha = 90°$) it determines the theoretical pumping heads for finite and infinite number of blades with the equations:

$$H_{t\infty} = \frac{1}{g} \cdot u_2 \cdot \left(u_2 - \frac{1}{tg(gr \cdot \beta_2)} \cdot \frac{Q}{2 \cdot \pi \cdot r_2 \cdot b_2} \right) \qquad (4)$$

$$H_t = \frac{1}{g \cdot (1+p)} \cdot u_2 \cdot \left(u_2 - \frac{1}{tg(gr \cdot \beta_2)} \cdot \frac{Q}{2 \cdot \pi \cdot r_2 \cdot b_2} \right) \qquad (5)$$

With the results given by these two equations it results the graphic from fig. 6.

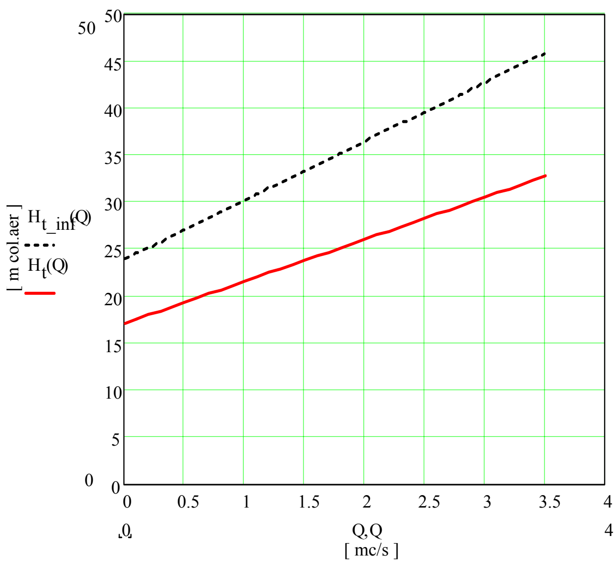

Fig. 6. The theoretical curves of the pumping head $H_{t\infty} = f(Q)$ and $H_t = f(Q)$ on the maximum flow rate domain

4.2. The estimation of the losses and the determination of the real operating characteristics

In the runner there are two category of losses: shock losses at the leading edge of runner's blades, friction losses and losses due to changing of direction in the curved channels between blades. For the first category from [5] the equation is:

$$h_{ps1} = \zeta_s \cdot \frac{u_1^2}{2 \cdot g} \cdot \left(1 - \frac{Q}{Q_0}\right)^2 \qquad (6)$$

For the second category of losses from [3], regarding the curved pipes with rectangular section, the equations are:

$$\zeta_{\cot} = k_\Delta \cdot k_{\mathrm{Re}} \cdot \zeta_1 + 0.0175 \cdot \frac{R_0}{D_H} \cdot \delta \cdot \lambda \qquad (7)$$

$$h_{p\cot} = \zeta_{\cot} \cdot \frac{w_1^2}{2 \cdot g} \qquad (8)$$

In fig. 7 are represented the losses curves:

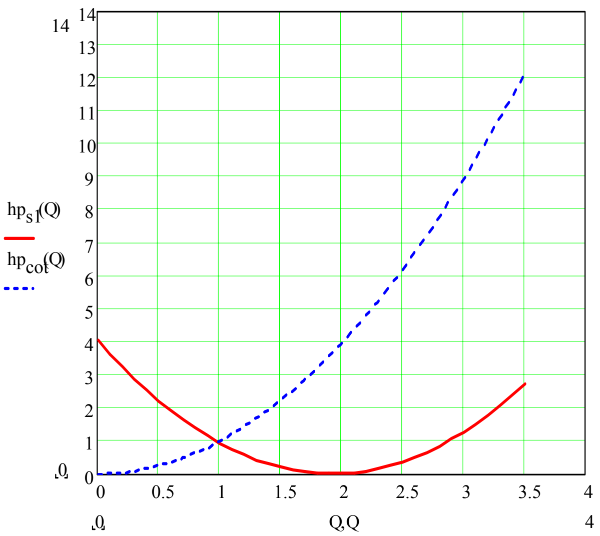

Fig. 7. The variation of the losses in the runner

Adding the losses and subtracting them from the theoretical curve results the probable real curve of the runner.

$$h_p = h_{p\cot} + h_{ps1} \qquad (9)$$

$$H = H_t - h_p \qquad (10)$$

Because the pumping pressure is usually expressed in mmH$_2$O, it performs the corresponding transformations and the graphic representation:

$$\Delta p_t = \frac{\rho_a}{\rho} \cdot H_t \cdot 10^3 \qquad (11)$$

$$\Delta p = \frac{\rho_a}{\rho} \cdot H \cdot 10^3 \qquad (12)$$

$$\Delta p_p = \frac{\rho_a}{\rho} \cdot h_p \cdot 10^3 \qquad (13)$$

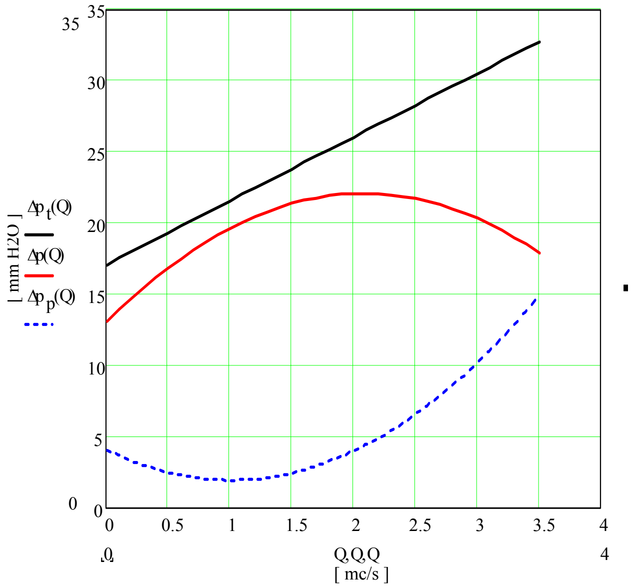

Fig. 8. The curves of pumping pressure: theoretical, probable real and added losses expressed in mmH$_2$O

5. THE COMPARISON AND CERTIFICATION OF THE CALCULATED OPERATING CHARACTERISTIC WITH ONE OF A SIMILAR EXPERIMENTAL TESTED VENTILATOR

In order to have a good reliability of the calculated characteristics, these will be compared with other similar curves from literature [4] which was experimental determined for a ventilator similar with this one. The data obtained was transposed through similitude at the dimensions and rotational speed of the studied ventilator.

When the transposition was made it was take into account the difference of the density of the air and the difference of the outlet width between the two ventilators.

Overlapping the two curves, the one calculated with losses and the one measured of the similar ventilator, one may observe a good overlapping especially in the exploiting domain which concerns us the most.

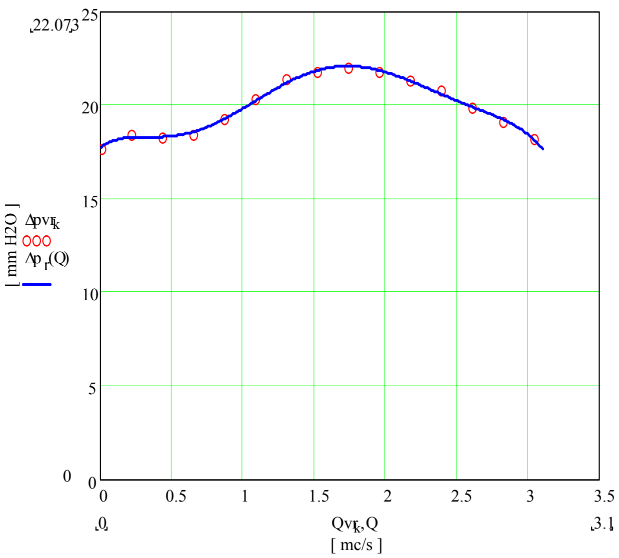

Fig. 9. The experimental curve of the TAGI TV-55 ventilator, transposed at the rotational speed and operating domain of the studied ventilator

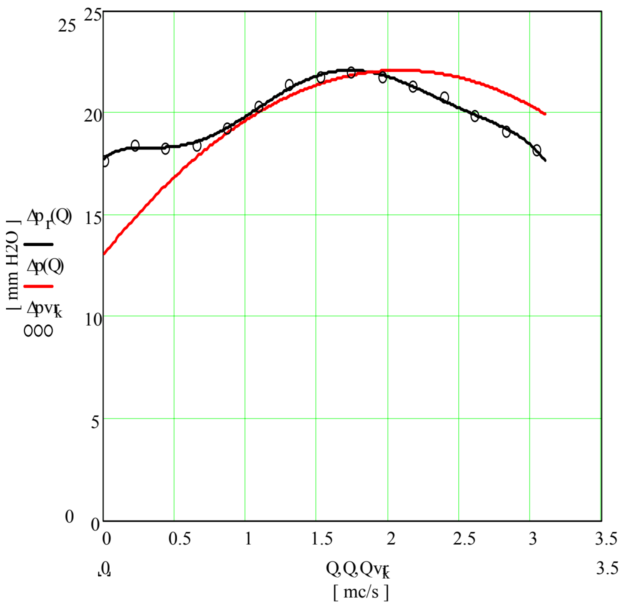

Fig. 11. Overlapped curves of the studied ventilator and of the reference ventilator (experimental determined)

6. THE DETERMINATION OF THE EXTERIOR CIRCUIT CHARACTERISTIC AND OF THE OPERATING POINTS USING 3D NUMERICAL SIMULATION

The pressure drops in the exterior circuit are obtained through 3D numerical simulation of the air flow. Through intersection of the operating characteristics of the ventilator and of the exterior circuit it obtains the operating points like in fig.12. It observes that the operating flow rate is at Q = 0.9 m^3/s. This value of the flow rate is much different of the estimated maximum value of 3.15 m^3/s.

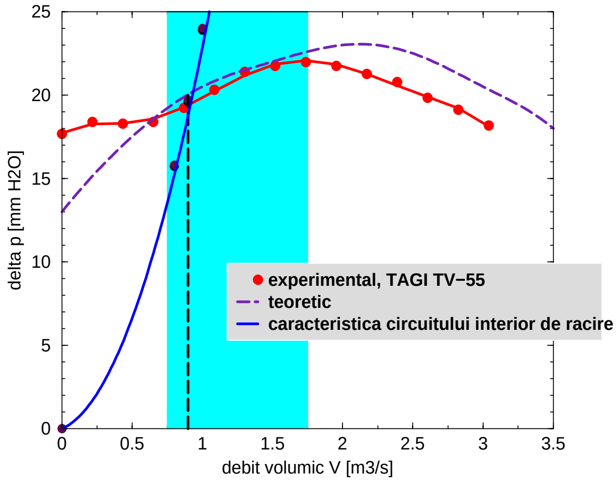

Fig. 12. The operating point of the ventilator in the exterior circuit

7. CONCLUSIONS ABOUT THE OPERATION OF THE VENTILATOR

From those shown above results that the ventilator has the operating characteristic closed to the one of a similar experimental tested ventilator. The equations used to determine the operating characteristic have a good covering on the exploitation domain.

The discharge of the air from the runner take place through an improper collector, which having relatively big section implies relatively small losses.

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