Performance confrontation of two blade impellers used in KGN refrigerator unit

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Abstract: The impeller of the semi-open type from a refrigerator unit is redesigned to improve efficiency and to reduce the pressure pulsations of the front case. Therefore, the closed-type of impeller with outer shroud is chosen. Dimensions are scaled according to Cordier diagram with rotational speed preserved. The resulting geometry is constrained by requirement of easy manufacturability by injection moulding process. These constrains don‘t permit to improve efficiency by redesigning impeller.

Key words: impeller, CFD simulation, ANSYS CFX 11.0 software

1 Introduction

The original impeller from refrigerator unit KGN is of the semi-open centrifugal type *(Figure 1)*. The gap between the impeller and a front case is large compared to the height of impeller blades. This feature reduces pressure pulsations at the front case, but it also worsens energy efficiency of the impeller. Furthermore, energy efficiency is worsened by simple geometric shape with flat inner shroud.

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Figure 1. *Impeller from refrigerator unit KGN*

2 The performance of the original impeller

The performance of the original impeller, which relates to specific values of mass flow, is showed by flow-head curve (indicated by "KGN fan head" label in *Figure 2*). Throttling characteristics was obtained experimentally in laboratory of Department of power engineering on the measuring stand with build-in Thomas cylinder. Particular points of characteristics were obtained so that at every change of mass flow rate of fan (in the range from 0.006 $kg.s^{-1}$ to 0.0225 $kg.s^{-1}$) was measured the pressure loss. The picture also contains system curve (indicated by label "system curve") that characterizes the pressure losses induced by flow in refrigerator unit, which were modelled in rectangle duct with dimensions 20x100x1400mm. The curve was interpolated from CFD simulation data. Efficiency curve (KGN fan eff.) was obtained from CFX simulation by means of equation which indicates ratio be-

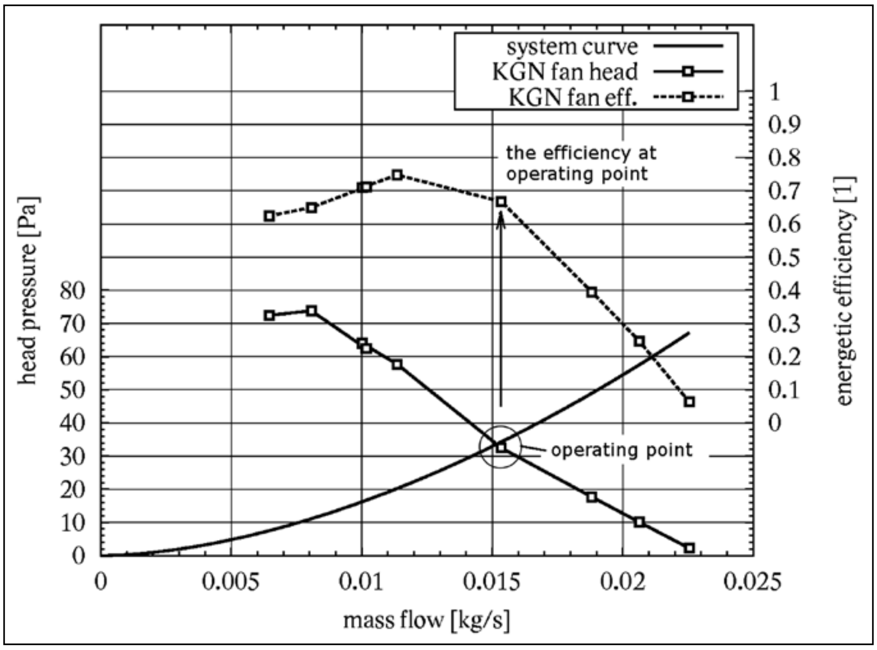

Figure 2. *Flow-head performance of the original impeller*

tween outlet energy of pressure forces and inlet work of torsion moment in the form of: effPowerliquidpump)=(-massFlowInt(ptotstn/Density)@outlet–massFlowInt(ptotstn/Density)@inlet / (Mk*abs(AngularVelocity))*180/3.14.

Impeller's operating point lies in the part of flow-head curve where the efficiency of impeller approaches its maximum. This implies that the former impeller design is well-matched to flow conditions but, actually, says nothing whether the type of impeller is the optimal construction solution. To compare with other possible impeller designs, the performance must be always expressed in non-dimensional form. Non-dimensional form must also suit the design process, where diameter D and rotational speed n must be independent. Thus, each impeller can be independently expressed as a function of the required mass flow Q_m and difference of total pressures Δp. These conditions are fulfilled by (non-dimensional) specific diameter D_s and specific speed N_s in terms of formula:

$$D_s = D\frac{\left(\frac{\Delta p}{\rho}\right)^{1/4}}{\left(\frac{Q_m}{\rho}\right)^{1/2}}, \quad N_s = n\frac{\left(\frac{Q_m}{\rho}\right)^{1/2}}{\left(\frac{\Delta p}{\rho}\right)^{3/4}} \qquad (1)$$

where Q_m is mass flow (kg.s^{-1}),
ρ is density of the working fluid (kg.m^{-3}),
D is outer diameter of impeller (m),
n is rotational speed (rad.s^{-1}),
Δp is difference of total head pressures (Pa).

Other definition of these numbers, based on volumetric flow and head height, can be found in engineering books (e. g. [1], [2]).

The performance of the impeller expressed in these numbers is plotted in (*Figure 3*). Apart from the flow-head curve, the figure contains another curve called Cordier diagram. Cordier curve is presented in various references, e. g. ([3, p. 27], [1, p. 61]).

All optimal flow machines operating at their best efficiency points lie very close to Cordier curve in this diagram (or nearby indispersion). The best efficiency point of the impeller from analysed body of KGN impeller doesn't lie on this curve therefore, from this point of view, it is possibly not optimal design structure.

3 Redesign of impeller

The process of impeller design is not straight and needs experience and iterations. Some basic rules can be found in [1] and [2]. In this article, a "quality" comparison of original and redesigned impeller (KGN4) is performed, while preserving mass flow, head pressure and rotational speed.

The first step is dimensioning based on Cordier diagram and the desired values of mass flow and head pressure that are dictated by system conditions. The D_s of the original impeller is slightly above Cordier diagram, therefore outer diameter of redesigned impeller should be reduced (from 0.12 m to 0.10 m).

The specific speed must be preserved also in redesigned one otherwise

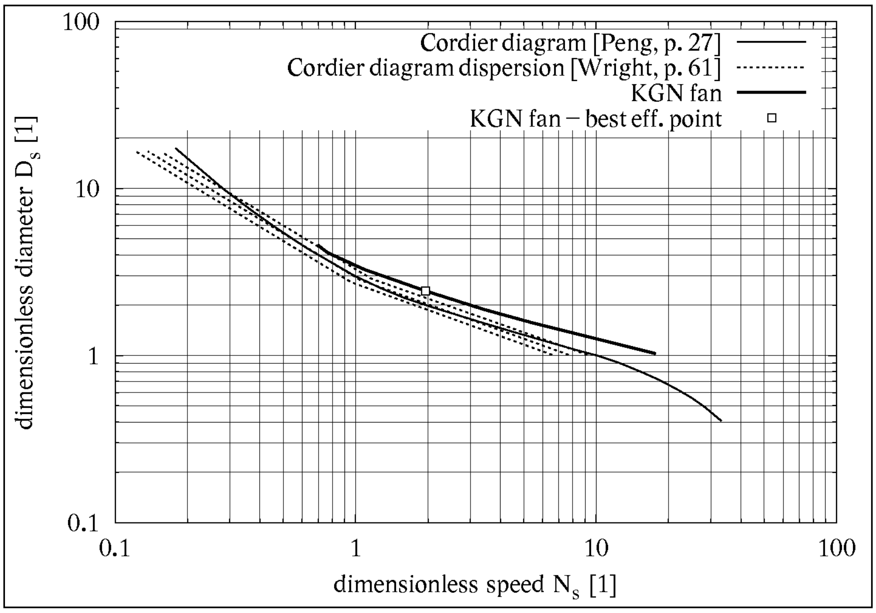

Figure 3. *Performance of the original impeller compared with Cordier diagram [3]*

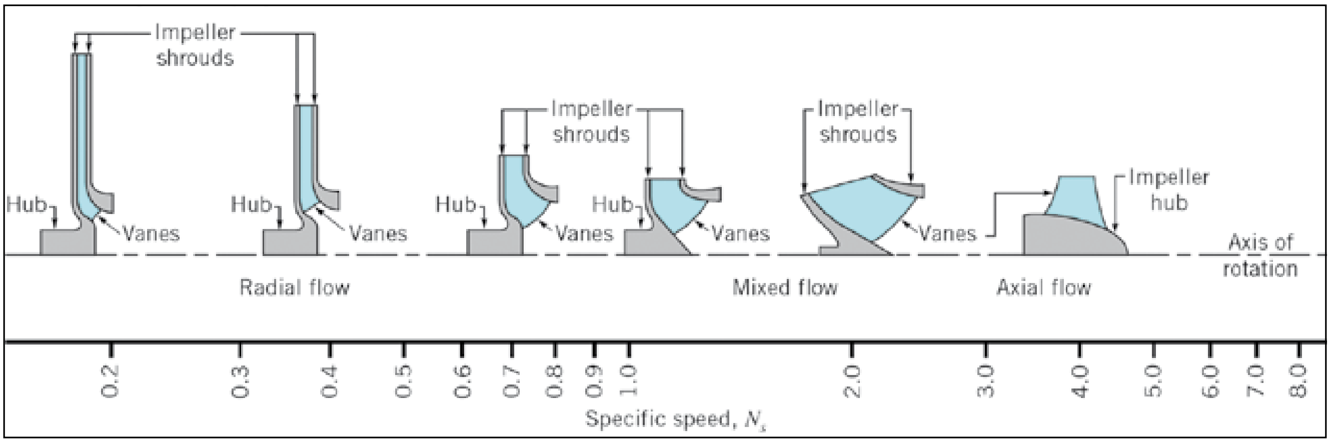

Figure 4. *Dependence of optimal design type on specific speed (from Munson 2002, p. 788)*

the rotational speed would change. However construction type of impeller will be changed. The type of optimal flow machine design, according to Cordier diagram, depends on specific speed N_s. This dependence is displayed in (*Figure 4*) (from [4, p. 788]).

It is evident, that the specific speed of the original impeller (N_s=2.0) lies in region of axial-radial impellers, therefore the original radial shape will be accordingly changed to reflect this fact. Moreover, the use of mixed flow impeller permits to experiment with the use of outer shroud of the impeller. The outer shroud prevents different pressures on sides of vanes to create pressure pulsations on front case. Easy manufacturability of impeller by injection moulding process imposes big constraints on possible shape. The inner diameter of the outer shroud must be larger than outer diameter of the inner shroud. The vanes of the impeller must be untwisted around radial axis. It means that their shape must be basically two-dimensional, with the only curvature in plane of rotation. This is severe limitation, because it means that the trailing edge on inner (bottom) side of vanes must have the same angle as the leading edge on outer (top) side of vanes. It also causes the sweep angle of vanes (relative to the direction of flow) and their short length (in direction of the flow). Therefore, to preserve the solidity (pitch/chord), the number of the blades is increased (from 11 to 13). The resulting impeller shape also with new shape of vanes (labelled as KGN4) is in (*Figure 5*).

4 CFD simulation of redesigned impeller

For simulation purposes of flow situation and gaining an information about "quality" of proposed impeller, the geometry of the CFD model is created as a 13th part of full circle (see *Figure 6*). The cut-out segment contains one vane in the middle, tubular inlet and radial outlet. The geometry of surfaces is simplified, smooth, without any real clearances and edges. The vane is infinitely thin. The mesh was created as a structured, using HEXA module of the ICEM CFD software. The resulting mesh contains 153 000 hexahedral elements and 164 000 nodes. The boundary layer details on vanes and outer shroud are meshed, so y^+ approaches the value of 1.

The computation was performed by software ANSYS CFX 11.0. The solver is based on Reynolds averaged Navier-Stokes equations. The fluid was assumed incompressible with constant density of 1.225 kg.m^{-3}. The turbulence was computed using k-omega SST model with gamma-theta model of boundary layer transition.

The boundary conditions were:
– the speed of rotation: 1950 min^{-1}

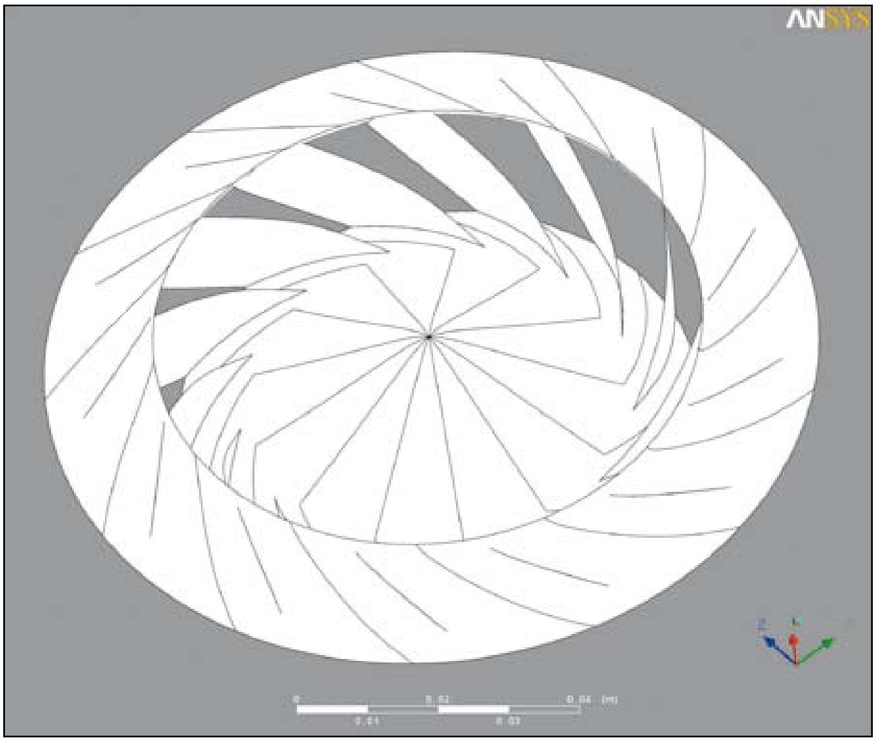

Figure 5. *The shape of redesigned impeller KGN4*

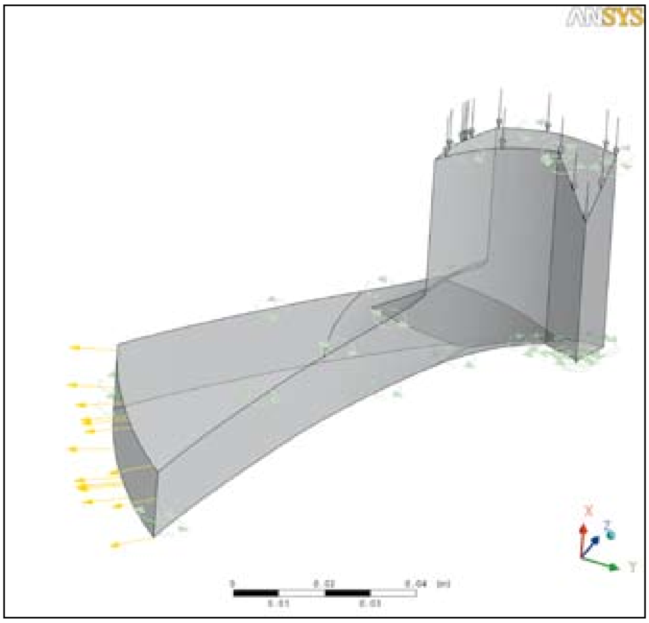

Figure 6. *The geometry of redesigned impeller*

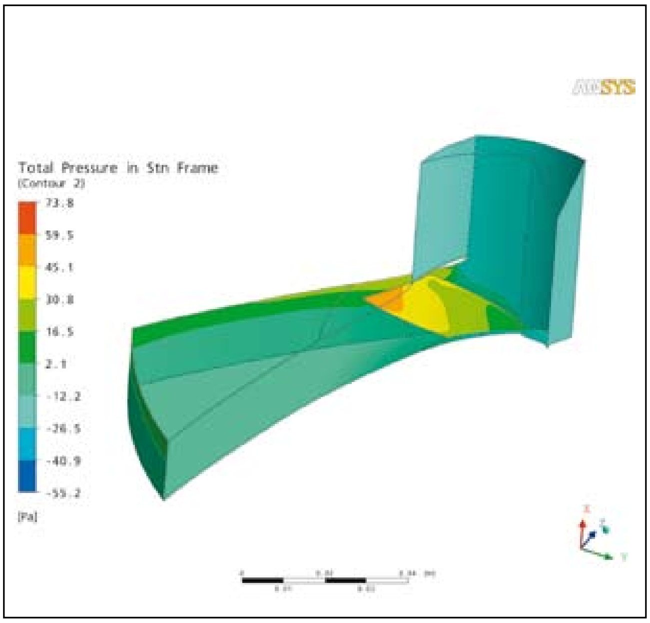

Figure 7. *Computed total pressure at low flow*

- inlet: constant normal velocity (2.26–4.6 m.s^{-1}), turbulence intensity 5%
- outlet: static pressure (0 Pa)

The solution in each case was stopped after 150–300 iteration, based on the monitor of head pressure during computation. Overall pressure distribution at minimum mass flow (Q_m = 0.005 kg.s^{-1}) of the air is found on (*Figure 7*).

5 Results and discussion

The resulting performance of the redesigned impeller is plotted in (*Figure 8*) together with performance of the original impeller. The optimum operating point of the new impeller is shifted to higher mass flow, and the head pressure is lower in total (progression shown in fig as "KGN fan head"). This can be partially corrected by the change of vane angles, but the lack of vane area on largest radius is probably equally important cause of lower head.

Solution process has shown that the efficiency of the redesigned impeller is not higher than of the original impeller. This is probably caused by severe limitations on vane geometry. The optimal shape of vanes needs to be twisted in three dimensions, to reflect the change of airflow on different radius and big sweep angle of leading and trailing edges. Especially at low mass flow it is evident, that sweep creates weak load of the inner part of the vane and this causes slow flow on the back case of the impeller (*Figure 9*). The second possible cause of low efficiency is rather low Reynolds number of vanes (Re = ~9000).

The display of best efficiency point of the new impeller in N_s–D_s graph (Figure 10) only confirms that it is not the optimal design and it is as far from Cordier diagram as the old impeller. Therefore easy manufacturability of the impeller as well the covering box by injection moulding process impose too severe limitations to improve efficiency of the original impeller.

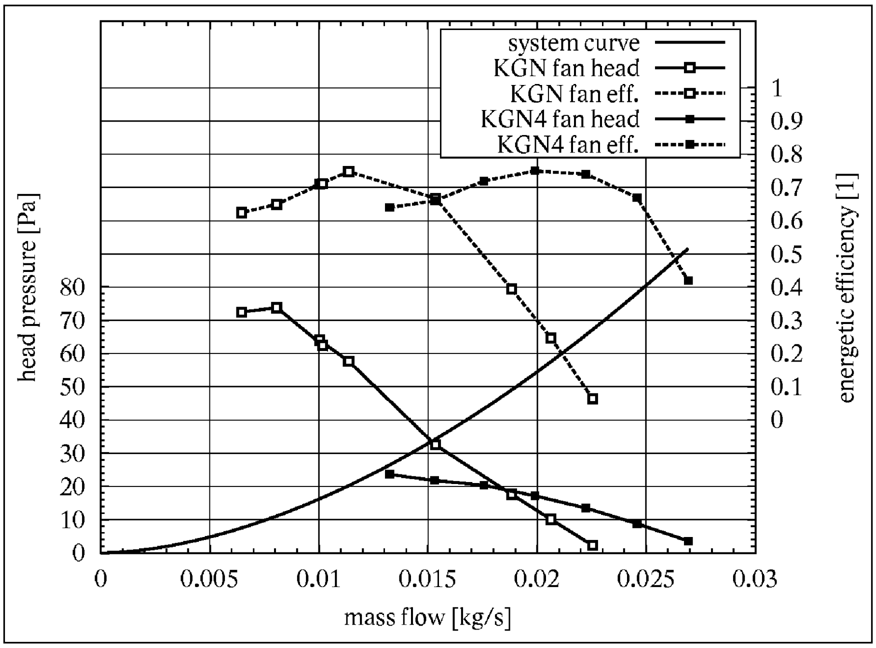

Figure 8. *Comparison of performance of original (KGN) and redesigned (KGN4) impeller*

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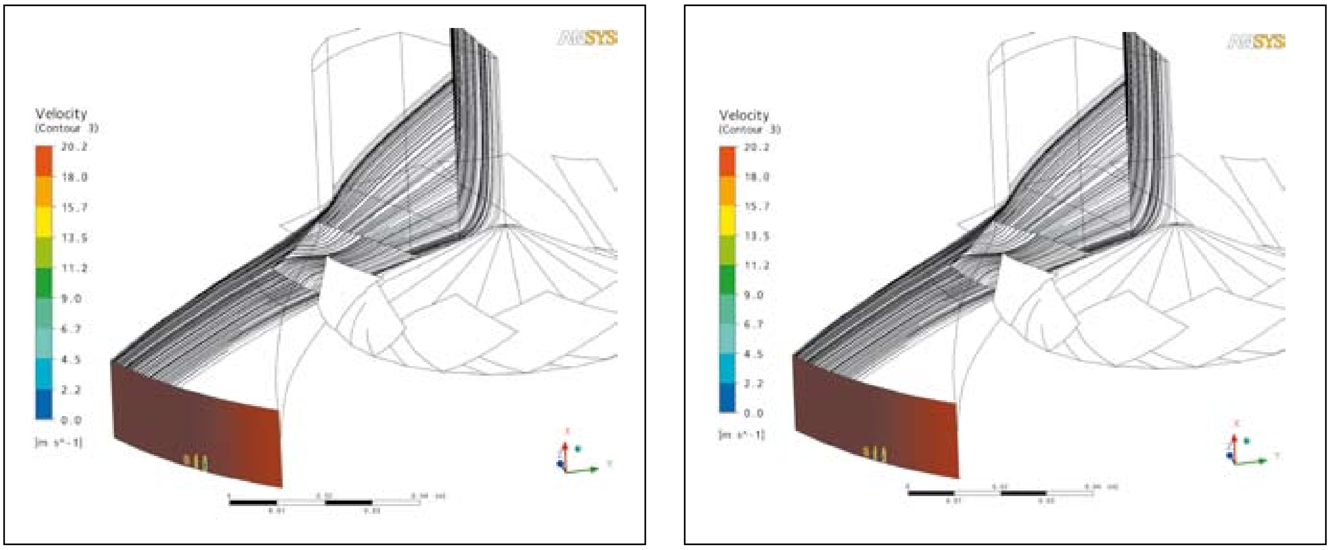

Figure 9. *Comparison of streamlines at low and high mass flow (KGN4 impeller)*

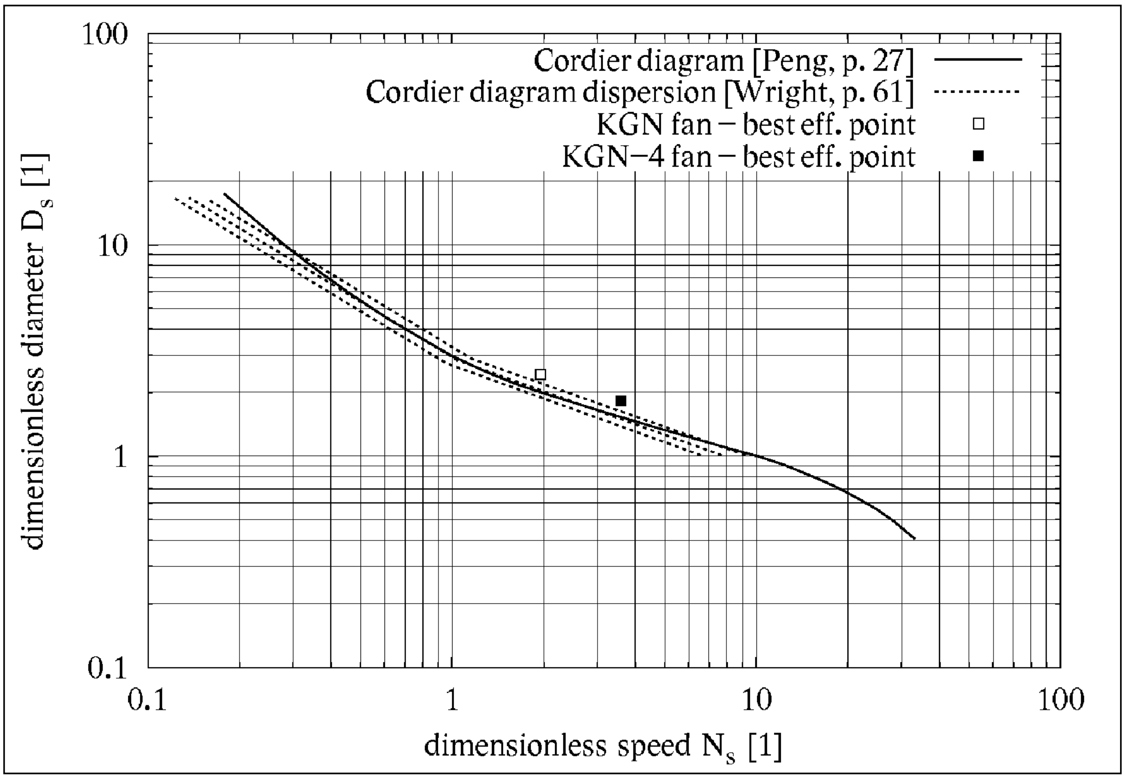

Figure 10. *Comparison of streamlines at low and high mass flow (KGN4 impeller) [3]*

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Primerjava karakteristik dveh rotorjev različnih geometrij v hladilnih enotah KGN

Razširjeni povzetek

V prispevku je predstavljena rekonstrukcija radialnega ventilatorja hladilne enote. Rotor odprtega tipa za hladilno enoto je bil preoblikovan z namenom, da se poveča učinkovitost in zmanjšajo tlačne pulzacije na sprednjem pokrovu ventilatorja. Zato je bil izbran zaprt rotor z zunanjim okrovom. Dimenzije ustrezajo Cordierjevemu diagramu in vrtilna hitrost je ohranjena. Dobljeno geometrijo omejuje zahteva po enostavni izdelavi z ulivanjem. Ta omejitev onemogoča preoblikovanje rotorja za povečanje učinkovitosti.

Pri reševanju problema se je pokazalo, da učinkovitost preoblikovanega rotorja ni večja od učinkovitosti originalnega rotorja. Vzrok je verjetno v tehnoloških omejitvah pri določevanju oblike lopatic. Optimalna oblika lopatic naj bi bila v danem primeru tridimenzionalna – zavita, tako da je zadoščeno vstopnemu toku zraka na rotorske lopatice. Zaradi geometrijskih omejitev so odstopanja tudi na izstopnem delu rotorja. Še posebej pri majhnem pretoku zraka je očitno, da vrtenje povzroča šibko obremenitev notranjega dela lopatice, zato se upočasni tok ob pestu rotorja (slika 9).

Prikaz maksimalnega izkoristka ventilatorja z modificiranim rotorjem na grafu Ns–Ds (slika 10) potrjuje, da ta oblika rotorja ni optimalna in da je glede na Cordierjev diagram ravno tako neustrezna kot oblika prejšnjega rotorja. Enostavna izdelava rotorja in pokrova z ulivanjem torej predstavlja resne omejitve za izboljšanje učinkovitosti originalnega ventilatorja.

Ključne besede: rotor, simulacija CFD, programska oprema ANSYS CFX 11.0