Szent István University

DEVELOPMENT OF AXIAL-FLOW FANS ASSEMBLED WITH SIMPLE GEOMETRY APPLIED IN THE AGRICULTURE

Thesis book of Ph.D. dissertation

Ildikó Molnár

Gödöllő
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Ph.D. school

name: **Ph.D. School of Technical Sciences**

discipline: **Science of Agricultural Technology**

head: **Dr. István Farkas**
Professor, doctor of Hungarian Academy of Sciences
Szent István University, Faculty of Mechanical Engineering
Gödöllő

supervisor: **Dr. Ferenc Szlivka**
Associate professor, Mechanical Engineering PhD
Szent István University, Faculty of Mechanical Engineering
Gödöllő

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Approval of the head of Ph.D. School

Approval of supervisor

Table of Contents

NOTATION

Note	Description	Measure
l	String length of the blade on the current radius	m
t	Division (2rπ/N)	m
c_f	Constant of the uptrust	N
N	Number of blades	-
$R=R_3=r/r_k$	Dimensionless radius	-
φ_{qy}	Quantity number within the ring	-
$\varphi_0=v_{0m}/u_k$ $\varphi_3=v_{3m}/u_k$	Quantity number varying along the radius	-
ψ'_{03}	Dimensionless pressure loss at the point „03”	-
$\psi(R)$	Pressure number varying along the radius	-
$\psi_{öi}$	Ideal total pressure number	-
$\psi_{öiv}$	Ideal total pressure number along the hub	-
$\psi_{öiv1}$	Ideal total pressure number along the circumference	-
$\psi_{öv}$	Real total pressure number	-
$\psi_{övt}$	Real planned total pressure number	-
ψ'	Pressure loss	-
p_{st}	Static pressure	N/m^2
$\Delta p'_{03}$	Pressure loss at the point „03”	N/m^2
$\Delta p_{ö}$	Total pressure increase	N/m^2
p_0, p_3	Static pressure at the points “0” and “3”, resp.	N/m^2
$\Delta p_{öid}$	Ideal total pressure change	N/m^2
η	Hydraulic efficiency	-
$\eta_ö$	Total efficiency	-
η_e	Resultant efficiency of losses	-
λ	Angular momentum shrinkage coefficient	-
$\nu=r_a/r_k$	Hub-ratio	-

1. PREMISES AND AIMS OF THE WORK

Application of fans is important on several fields of the agriculture. Modern aeration systems might have key effect on the competitiveness of an agricultural unit. Requirements against the fans are going to be stronger in the last couple of years: performance, low noise level, geometry, efficiency. Geometry for easy production and low cost level are crucial in point of the production, therefore efficiency legs behind in point of the construction. In this way it can happen, that the up-to-date fans have only efficiency about 40-50%.

Fans can be sorted into two main classes in point of the flow and its construction: radial or centrifugal fans, and the axial flow fans. In case of radial impellers the air comes into the fan parallel to its axle (or already with angular momentum), then it flows onto a plane perpendicular to the inlet- axle. This stream does not continuoue the flow along the radius after this direction-change, but it let develop different bended flow lines between the blades on perpendicular planes to the inlet-axle. In case of axial flow fans the inlet air flows outwards in the direction of the axle, too, maybe with angular monument. Streamlines flow between the blades as near helixes on a cylindrical surface being concentric on the axle.

Inland research on fans has long tradition. We have still broad professional literature, research background and production capacity on this field. Essential development on the operation of fans were carried out by *Keller* [Keller, C. 1937], *Howell* [Howell, A. R. 1942], who have enriched the theory of fans with the grid-theory. The method of singularities of Gruber [Dr. Gruber, J. 1978], which is international acknowledged, was further improved by *Vajna* [Vajna, Z. 1987], *Füzy* [Füzy, O. 1991], *Czibere* [Dr. Czibere, T. 1986]. The most important international authors are *Wallis* [Wallis, R. A. 1961], *Beiler–Carolus* as well as *Lakshminarayana* [Lakshminarayana, B. 1970], whos work is significant in point of measurement methods and calculations, too. The LDA-instrument is suited for detailed analysis of the fine structure of the streams developing in the environment of the blades of an impeller. Inland researches in this topic are e.g. as follows: *Vad János* and *Bencze Ferenc* [Vad, J., Bencze, F. 1998]. They managed to map the fine structure of the velocity field developing behind the blades of an impeller by the help of the LDA-measurements. The developed measurement method is suitable for measurement of all the three components of the velocity field; however it does not support that of the pressure ratios. International research topics in terms of fans and compressors are very widespread, which can be read in detail in the literature chapter of the complete dissertation.

Accordingly this research aimed first of all the followings:

- overview and classification of sizing of axial flow fans,
- experimental analysis of different blade types,

- utilization and further development of the available experimental methods,
- proposals for further development of fan engineering,
- completition of a numeric simulation for the analyzed impeller,
- comparison of the measurement results and that of the numerical simulation,
- re-engineering of the chosen existing fan type on the basis of mine development proposals.

The corresponding literature is discussed in the first part of my dissertation, then the description for the velocity and pressure field developint near to the impellers of the axial fan can be found, where the measurement method based on pressure measurement was improved. These results can be used very well during the measurement of the velocity field developing near to the impeller or a unique blade, furhtermore it allows us to find out the velocity- as well as the pressure field. Spatial distribution of the static pressure will be determined right before and behind the impeller, moreover the spatial distribution as well as the direction of the total pressure within the system attached to the blade channels. In this case there is the possibility to record even all the pressure map developing in the blade channels, and it allowed me the tracing of local loss sources of the flow processes within the impeller, and to deremine the pressure loss as well as the hydraulic efficiency in every point. It is followed than by the detailed discussion of the application of the nowadays widespread numerical methods, where the results of the Computational Fluid Dinamics (CFD) will be compared with that of the measurements. Among others the measurement results within the aeration channel serve as input, frame- and boundary conditions for the computational analysis of the stream, and on the other hand the resulted numerical calculations can be compared with measured values from other places. The flow modul of ANSYS-CFX was chosen from the CFD programs, which I have free access in the Szent István University - Faculty of Mechanical Engineering to. CFD supports the engineering in a very large scale; because it calculates in detail the stream flow, velocity space, pressure space etc., if the geometry of the impeller and the blade are known, respectively. There are two main tasks to solve during the design of axial flow fans in general: the first one is the direct design method, and the other one is the inverse task. These both are used recently together, as a united designing method and iterating process. In this way the direct design should apply first, where the features of the impeller and that of its blade will be determined, then the stream around the impeller assembled with the resulted blade geometry will be simulated by the CFD. The blade design will be modified according to these results. This iteration continuous until it results in the expected blade and impeller. But - of course - real operation tests are the final control against the engineering.

2. MATERIALS AND METHOD

2.1 Sizing for variable blade-circulation along the radius in case of a given impeller

Authority of some simplifying assumptions was confirmed by the measurements during sizing for variable circulation, but some cases those were declined. Therefore the refinement of the sizing method is necessary.

The design proceeds with a quantity and a pressure difference. The diameter (D), revolution (n) and hub-ratio (ν) are taken on the basis of our designing experience, which determine the main features of the fan. Then the quantity number (φ) and pressure number ($\psi_{öv}$) can be determined. The task of the variable circulation is to find such a ψ distribution function, which results in a fan with the easiest production and the best possible hydraulic efficiency. The φ distribution can be calculated along the radius by the help of the ψ distribution function and the determined differential equation detailed in the dissertation. Dynamic properties of the blades as well as the value of ($1/t \cdot c_f$) can be determined by the φ and ψ differential equation, respectively. During the further steps of the sizing the grid theory or the measurement data of a unique wing will be used to dermine the appropriate blade number and the correct blade shape, too.

Whereas *Somlyódy* has neglected the loss part on the right side of the differential equation (Eq. 2.1) required for the design of axial flow fans (therefore its derivated value remains zero) [Somlyódy, L. 1971], for the meantime - based on the real total pressure - *Bencze-Szlivka* have take the loss part into consideration, and they assumed the hydraulic efficiency constant along the radius (Eq. 2.2) [Bencze, F., Szlivka, F. 1985].

$$\psi'_{03} = (1-\eta)\cdot\psi_{öi}$$

Where: η means the hydraulic efficiency.

$$2\varphi_3 \cdot \frac{\partial\varphi_3}{\partial R_3} - 2\varphi_0 \cdot \frac{\partial\varphi_0}{\partial R_0} = \frac{\partial\psi_{öi}}{\partial R_3}\cdot\left(1-\frac{\psi_{öi}}{2R_3^2}\right) - \frac{\partial\psi'_{03}}{\partial R_3} \tag{2.1}$$

Somlyódy has neglected the $2\varphi_0 \cdot \frac{\partial\varphi_0}{\partial R_0}$ és a $\frac{\partial\psi'_{0-3}}{\partial R_3}$ coefficient during his experiments.

$$2\varphi_3 \cdot \frac{\partial\varphi_3}{\partial R_3} - 2\varphi_0 \cdot \frac{\partial\varphi_0}{\partial R_0} = \frac{\partial\psi_{öi}}{\partial R_3}\cdot\left(1-\frac{\psi_{öi}}{2R_3^2}\right) - \frac{\partial\left[(1-\eta)\cdot\psi_{öi}\right]}{\partial R_3} \tag{2.2}$$

2.2 Considering the variation of the hydraulic efficiency during the solution of the differential equation

The design point of both methods does not coincide with the ideal characteristic curve calculated from the measurement. This assumption will have been proven by the later introduced measurements.

The hydraulic efficiency part of the differential equation will be considered during this fan design case, and it is supposed to be variable along the radius. After further arrangement the equation 2.3 will be as follows:

$$2\varphi_3 \cdot \frac{\partial \varphi_3}{\partial R_3} - 2\varphi_0 \cdot \frac{\partial \varphi_0}{\partial R_0} = -\frac{\partial \psi_{öi}}{\partial R_3} \cdot \left(-\frac{\psi_{öi}}{2R_3^2} \right) + \frac{\partial (\eta \cdot \psi_{öi})}{\partial R_3} \tag{2.3}$$

where $\varphi_0 = \frac{v_{0m}}{u_k} = \text{constant}$.

$$2\varphi_3 \cdot \frac{\partial \varphi_3}{\partial R_3} = -\frac{\partial \psi_{öi}}{\partial R_3} \cdot \frac{\psi_{öi}}{2R_3^2} + \frac{\partial \eta}{\partial R_3} \cdot \psi_{öi} + \eta \cdot \frac{\partial \psi_{öi}}{\partial R_3} \tag{2.4}$$

Total efficiency consists of several part-efficiency, such as the hydraulic one (η) in the equation 2.4. The hydraulic efficiency is devided into 3 main parts in general: collision loss at the inlet, because the flow direction differs from that of the inlet edge, the second one is the frictional loss, and the third one is the angular momentum decreasment. The efficiency of the impeller was determined in pursuance of the measurements, which I call hydraulic efficiency. In my case this efficiency or the losses causing this has/have other components, as well. I do not want to undertake its accurate separation; I only enumerate the loss sources of the hydraulic efficiency, which was determined by measurements. Velocity and pressure were measured both right behind of the impeller, therefore the measured and the calculated hydraulic efficiency includes the losses establishing in the impeller.

Detailed description about loss sources can be found in the literature [Gruber, J. 1978]. The loss sources are as follows: gap loss between the impeller and the housing. Larger part of this loss loaded as well on the hydraulic efficiency during my fine structure measurements. Secunder losses exist because of the bended stream lines within the blade channels of the impeller. Additionaly a radial flow comes into existence, too, which must be in case of variable circulation. This loss is part of the hydraulic efficiency calculated from the measurements.

Other loss types arose already after the impeller, therefore these losses load only on the total pressure. The most important ones are as follows: loss of the diffuser, which exists because the flow has to fill up the whole cross section going out from the blade-wreate ring, henceforth diffuser-type stream occurs, which results in losses, by all means. The rotational loss, which occurs by fans without any deflector, in this case can be considerable.

Hydraulic efficiency and total efficiency of the fan are not equal. Total efficiency is always less then the hydraulic efficiency, because there are additional losses within the fan, too. The resultant efficiency of these efficiencies should call from now η_e.

Hence the total efficiency of the fan according to the annotations is the following: $\eta_ö = \eta \cdot \eta_e$.

2.3 Introduction of the applied measurement method

I thought important to do measurements before and behind an impeller to determine the flow features as well as to map the velocity distribution in case of different rpm and working points (velocity- and pressure distribution).

Pitot-tube as well as sidelong manometer was used to determine the characteristic curves of the fans.

A Testo521 type diaphragm differential pressure gauge with a measurement limit of 10 hPa was used beside the analogue pressure gauges, however quick pressure changes cannot be measured. To avoid the previous problem such a pressure gauge was necessary, which has much less inert time, and perhaps works with other theory.

The applied measurement method allows us to determine the expected behavior of the blades of a fan. The static and total pressure were measured with two different pressure gauges behind the impeller, therefore measurements were carried out not at the same time. Because of this time gap each properties of the fan have had to be set very accurately as well as its spatial locations, which made me much easier to compare the coherent static- and total pressure data later.

Pressure gauges were assembled with transmitters, as well. The scale of the pressure changes were some houndred Pascals, therefore microphone or other similar operating pressure gauge with piezocrystal was suitable for our aims. Every pressure transmitter with piezocrystal requires reference pressure on one side, which was – in my case – the athmospheric pressure. Pitot-tube or tubular sensor was used to measure the total pressure change, and on the other hand Ser-disc was used to measure the static pressure.

Ser-discs were used instead of the Pitot-tube to measure the total pressure, because of the positioning problem of the second one (it does not measure at the same points; it must be corrugated according to the rotation). For the sake of the determination of the accurate direction of the total pressure on the basis of the measurement data a new algoritm was developed, which will be detailed in the next chapter (determination of regression curves).

2.3.1 Method for the measurement of the total pressure

Pitot-tube assembled on the tubular sensor as well as the tubular sensor itself was used to measure the total pressure. The realization of this measurement was much harder then that of the static pressure. Total pressure values can only be determined, if the hole of the measuring gauge faces the actual velocity direction, however the scale and direction of the actual velocity behind the impeller varies all the time. Therefore the determination method of the total pressure values were as follows: velocity gauge was set at a given radius (the rod was pushed by 10mm steps inward) and in a given direction (α = 0° upto 90° with 10° steps).

The main aim of the measurement is to allocate the place, direction and value of the maximum pressure. Values of the maximum pressure can be determined from the regression curve-set. A unique single pressure characteristic curve belongs to every measurement point (at a given radius and rotation angle (α)).The angular transmitter has a trigger point once during the rotation, which represents the termination (start and end) points for the measurement. This method resulted in pressure characteristic curves with one single variable for every setting (gauge rotation and radius), though these are not equal the total pressure curves. Total pressure curves can be calculated, if the maximum values of each pressure characteristic curve as well as the related gauge rotation values, which give the direction of the outlet absolute velocity, are selected by the help of the corresponding appropriate points.

10 pieces of pressure characteristic curve belong to a given radius, and a single pressure characteristic curve consists of 1024 data points (these 1024 points means one single total rotation of the impeller). Determination of every regression curve belonging to a given radius was as follows (see Fig. 2.1): all pressure characteristic curves belonging to a given radius (10 pieces) were analyzed, where maximum points were chosen (location and direction (α_{max})). The regression method was chosen to reduce the inaccuracy.

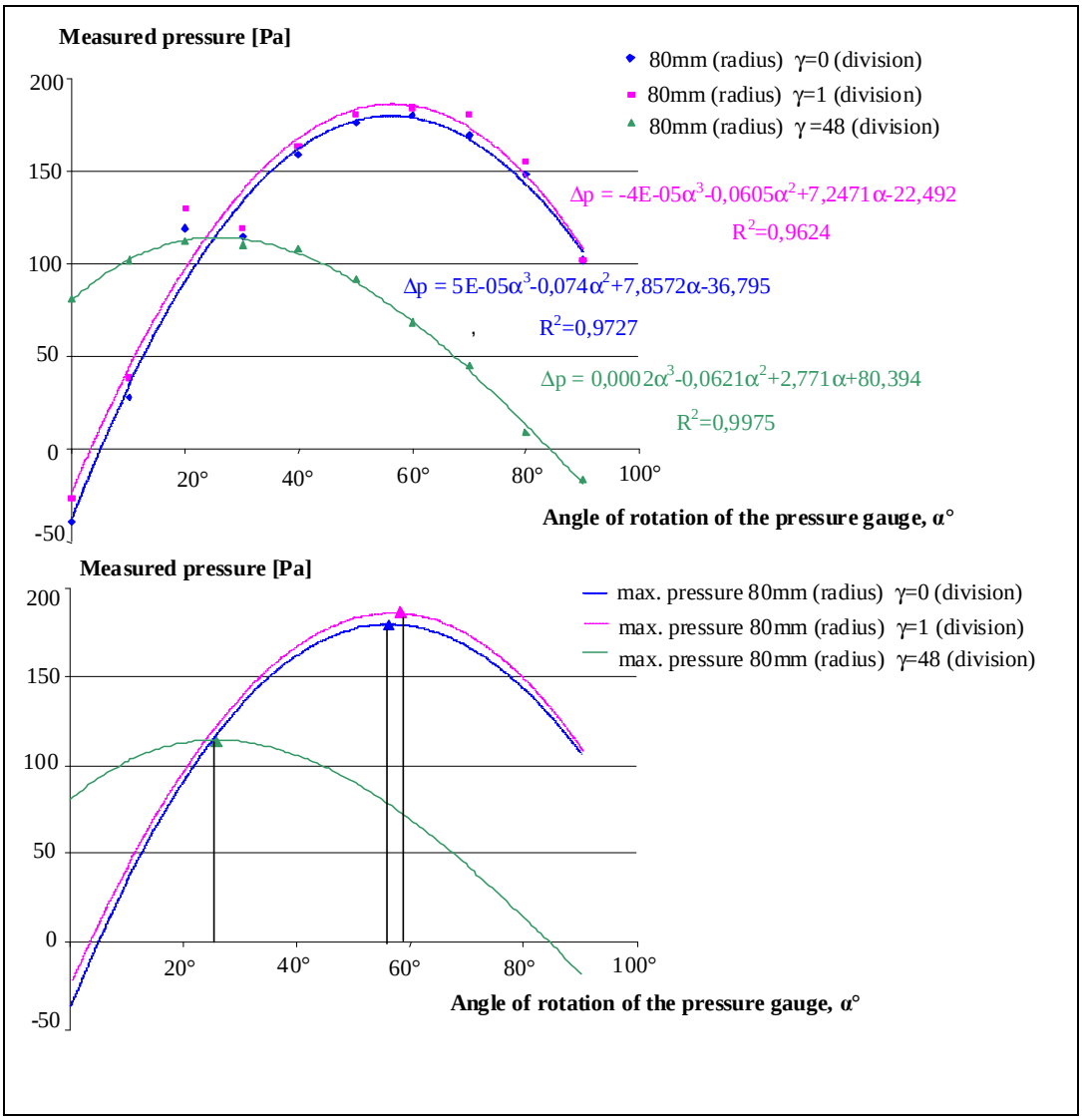


Figure 2.1 Allocation of maximal pressure values

First, second, third, fourth and fifth order regression curves were fitted for several measurement series to choose the most adequate one. In case of the first order fitting the maximum point will be at the frame of the range, therefore it is not suitable for our aims. By second order fitting there is the possibility, to set a maximum point within the analyzed range, therefore, and because it's simply form, its usage seemed to be obvious. Indeed the determination coefficient (R^2) value of the second order fitting was worst then that of the third, fourth or fifth order fittings (see Table 1.), moreover its deviation from the measurement points was much higher, too. Fittings above the third order resulted in just less improvement of the final result (approximation, correlation coefficient). Fig. 2.2 shows the

determination coefficients of the regression polinom fitted onto a measurement data set with $\gamma = 0{,}1$ and 48 division

Table 1: Values of the determination coefficient (R^2) calculated by the least square method

80mm radius	R^2 (linear)	R^2 (2nd order)	R^2 (3rd order)	R^2 (4th order)	R^2 (5th order)
γ=48 div.	0,9448	0,9774	0,997	0,9972	0,999
γ=0 div.	0,3996	0,9895	0,9926	0,993	0,9936
γ=1 div.	0,3954	0,9679	0,985	0,991	0,994

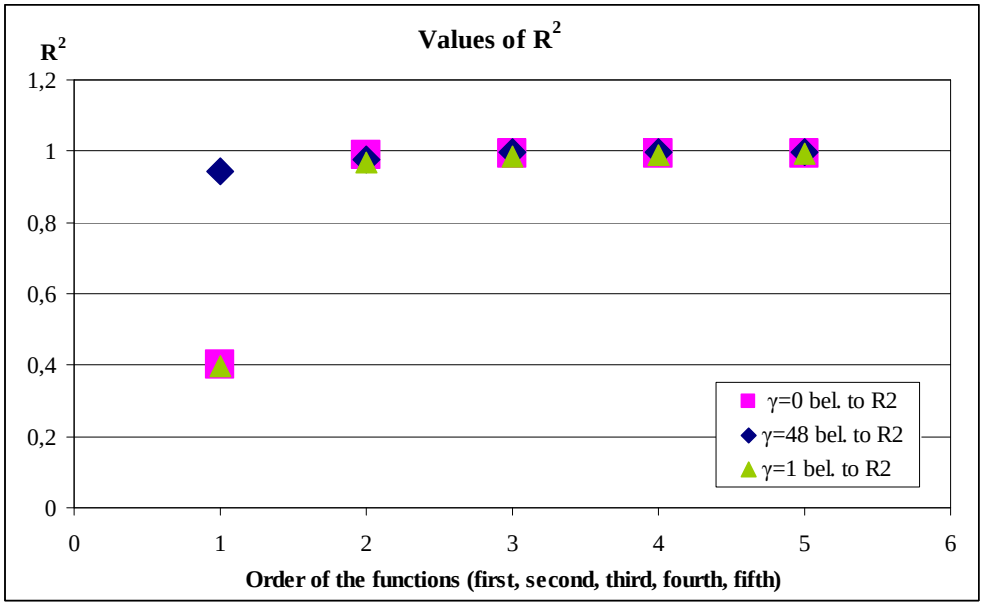


Figure 2.2 Approach with second, fourth and fifth order functions

2.3.2 Determination of the efficiency of the fan

To reach the goals of this dissertation it was crucial to take care of the most accurate determination of the efficiency of the axial fans, which would have been carried-out by measurements. For this sake the knowledge of the inlet performance of the fan is essential. The main problem during the determination of the efficiency value of a fan is the measurement of the torque; other properties – like pressure, volume flow or rpm – can be measured easier. Knowledge about the efficiency of the fans gives important informations about the features of the impeller.

Strain gauges were applied for the measurements, because in my point of view it seemed to be realizable. A built-in torque gauge could make the measurement more quick and simply, but this solution would require appreciable technical modifications at the joint between the impeller and the engine, therefore strain

gauges were chosen to measure the reaction torgues. Indeed a similar method to the aborted one was realized previously in case of measurement instruments of axial flow fans [Bencze, F., Füredi, G., Szlivka, F. 1989]. The analyzed impeller was built-in an aeration channel, where a multi-channel data aquisitor was used to receive the signs from the strain gauges with the values of the reaction stress.

Rotation of the impeller blades results in reaction forces, which causes reaction torque on the axle of the impeller in the plane of it but opposite direction to the rotation. Components of these reaction forces which are perpendicular to the axle of the impeller work as compressive forces and push the impeller from the exhaust to the suction side. Reaction torque and forces act on the (three-legged) bracket of the engine (through the drive engine) (Fig. 2.3), where its directions veer round, although its scales remain. The measurement method applied in this dissertation is based on the measurement of this compensation torque. The axis of this torque is parallel to that of the tube as well as the rotating axle of the fan.

Besides the compensation torque other torques and reaction forces act on the three legs of the bracket, as well. Weight of the engine itself results in draw- and compression forces within the supports, which makes the upper legs be drawn and the lower ones be compressed, respectively (Fig. 2.4). The axis of the reaction torque is parallel to the rotation axle of the fan, too, similar to the compensation torque acts on the impeller. This torque can be eliminated only that way, if it is considered as zero-deformation state, and the new torque will be measured according to this reference torque. This value is the same in the case of the stationary- and the rotating fans.

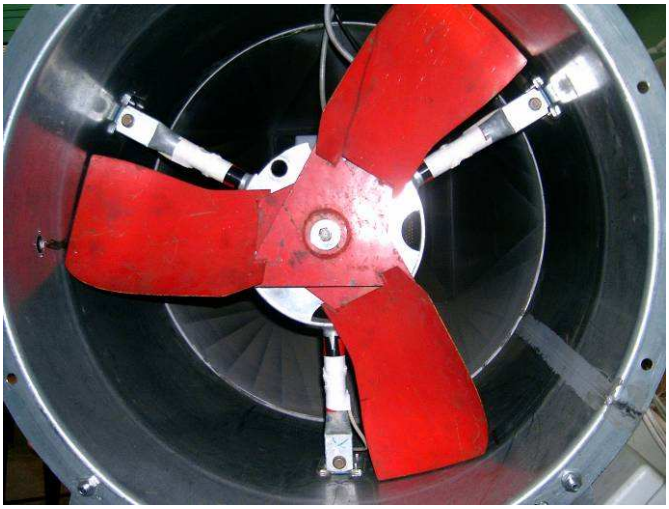


Figure 2.3 The bracket and the fan installed into the aeration channel

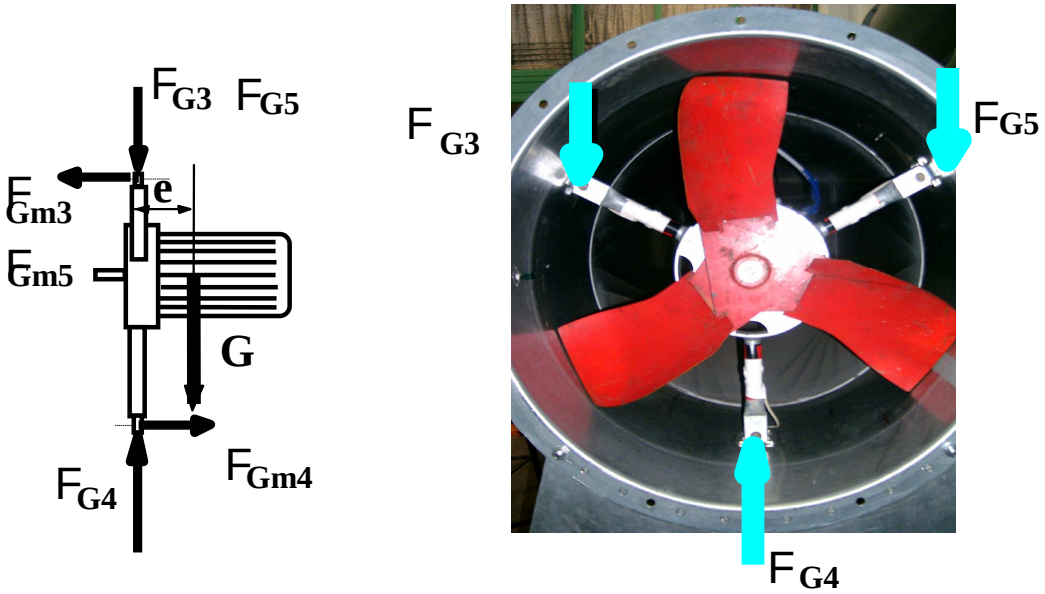


Figure 2.4 Schetch of the draw- and compression forces occurred by the engine weight

The center of gravity of the engine does not coincide with the plane of the bracket; hence a new torque will appear perpendicular to the plane of the previous ones, moreover the force which pushes the impeller backwards results in a bending in the same direction. Axes of the bendings coincide with the rotation plane of the fan; therefore it adds fortunately only small error to the measurements. This is an important fact in point of the accuracy of the measurement, because bending caused by the pressure is very different by the stationary- and the rotating fans, respectively. A schetch of reaction forces against the torque caused by the rotation of the fan can be seen in Fig. 2.5.

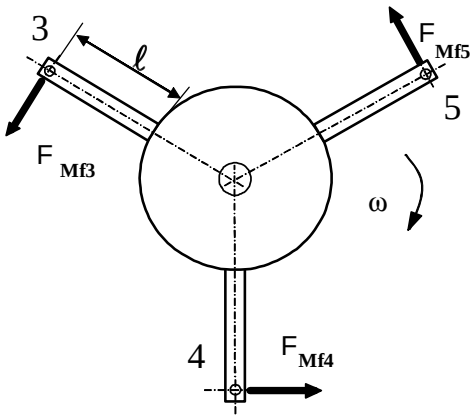


Figure 2.5 Reaction forces against the torque

3. RESULTS AND DISCUSSION

3.1 Characteristic curve determination of the analyzed impellers

First of all the characteristic curve of the impeller was determined by the measurement equipment. Knowing the characteristic curve is important during the measurements, because it helps us to choose the appropriate points of the characteristic curves for the sake of the deviation analysis. Measurement results of the characteristic curves can be seen on the corresponding figures (type “Rotating” – Fig. 3.1, type “Benji” – Fig. 3.2, type “Kamleithner” – Fig. 3.3).

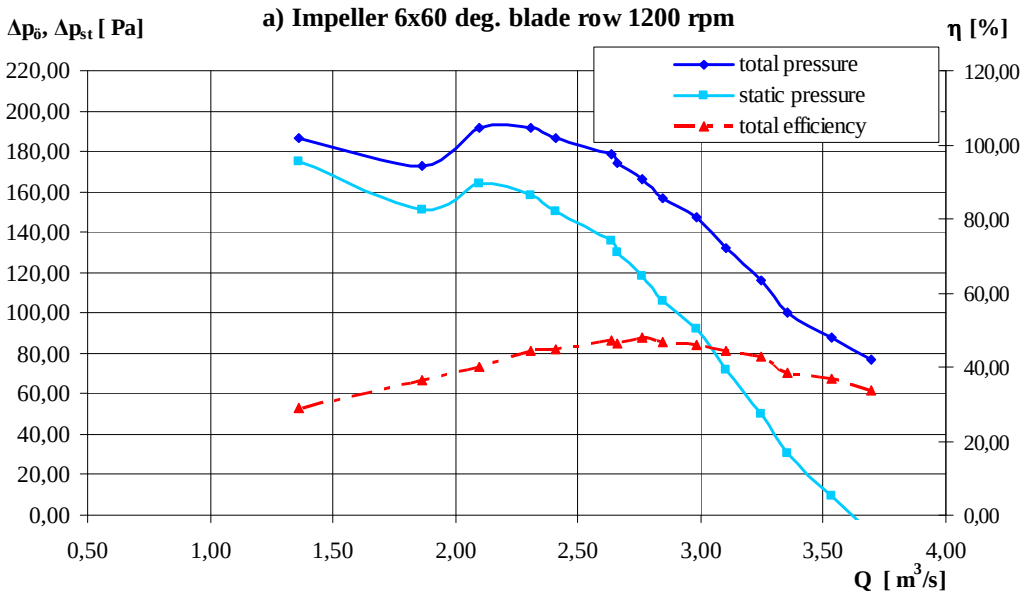


Figure 3.1 Characteristic curve of fan type „Rotating”

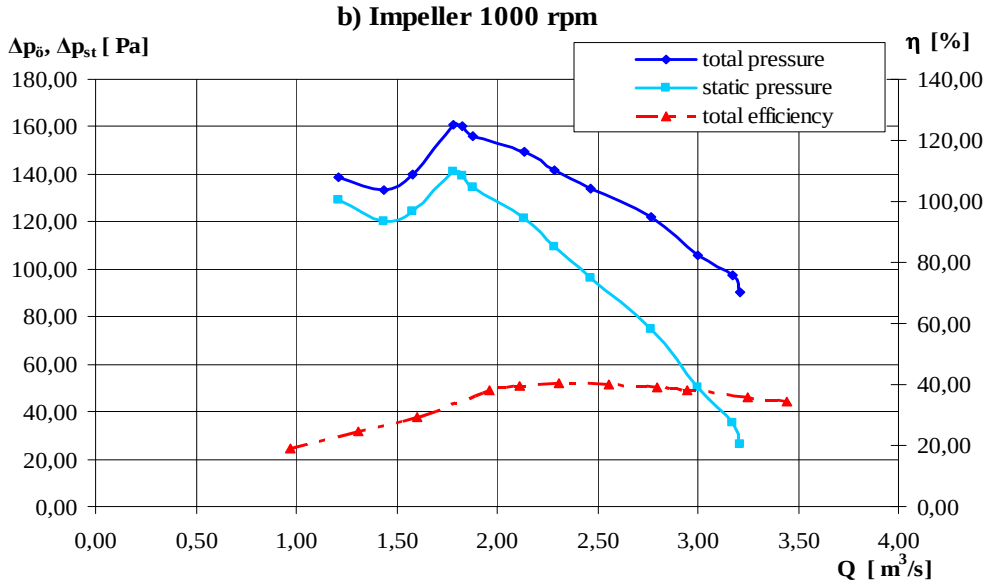


Figure 3.2 Characteristic curve of fan type „Benji”

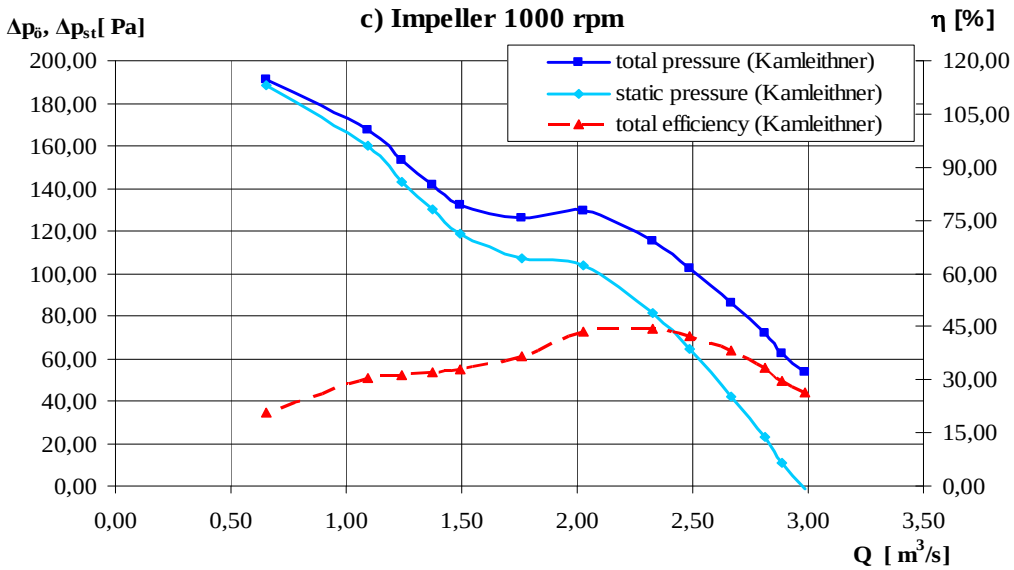


Figure 3.3 Characteristic curve of fan type „Kamleithner”

For the sake of better comparisonability the above figures were shown with dimensionless (relative) numbers, too (Fig. 3.4). The dimensionless pressure number (relative to the total pressure) as well as the quantity number was calculated by the equations 2.3 and 2.4, respectively. Design parameter of the “Rotating” fan will be detailed in the frame of the modified design method.

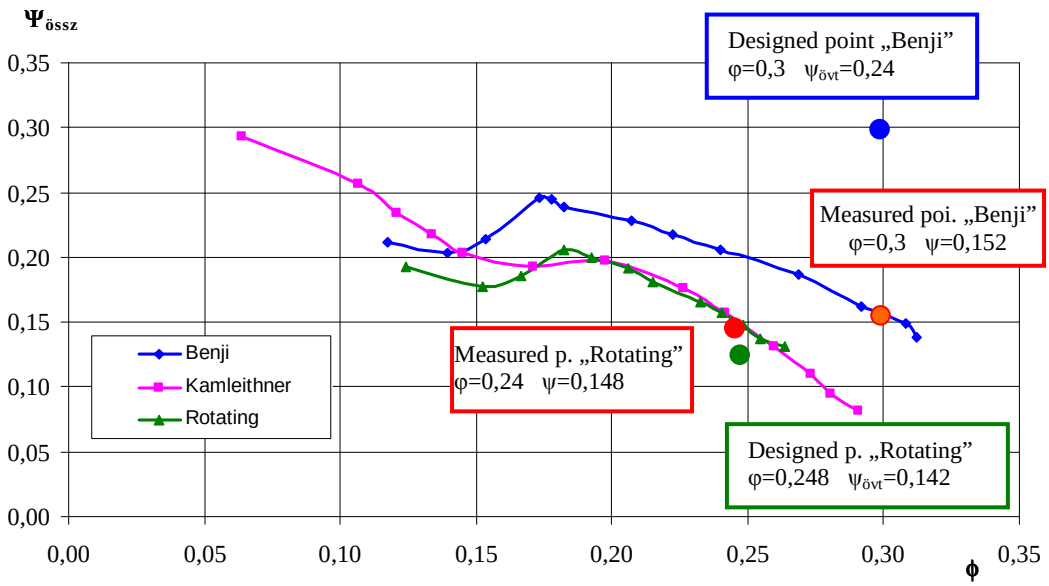


Figure 3.4 Dimensionless characteristic curves of fans type “Rotating”, “Benji” and “Kamleithner”

Original design parameters of the fan type “Kamleithner” were unknown, although after comparison of efficiency of the three fan types it can be seen, that the “Kamleithner” – against its shaped and bended cast-iron blades – showed similar flowing features. Its maximal efficiency exceeds just with 5% of that of “Benji”, and its acustic properties are only a little bit worst than that of “Benji”. (The above annotation about the acustics is based on subjective observation; no furher acustic measurements were carried out for this sake.)

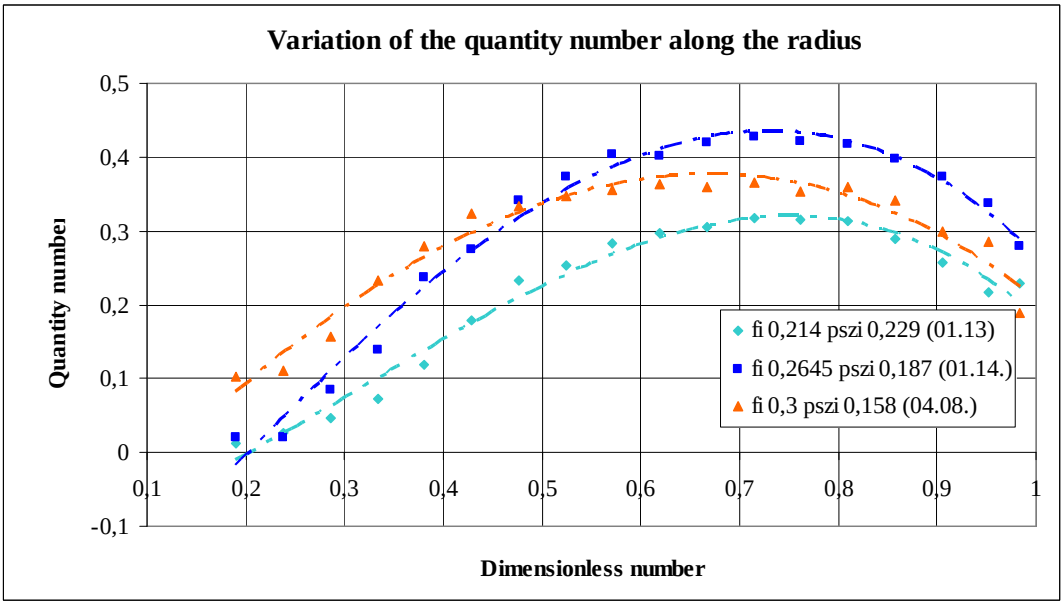
3.2 Fine structure distributions measured in the environment of the fan

Fine structure distributions developing in the environment of the impeller were analyzed in case of a cooling fan as well as fans in type of “Benji” and “Rotating”, respectively. These resulted measurement results were compared to that of the design for variable circulation. Three points on the characteristic curve of the “Benji” fan was chosen for the sake of the fine structure analysis: the first one from the high quantity range $\overline{\varphi} = 0{,}3$, $\overline{\psi} = 0{,}158$, and the second one from the largest realized total pressure range $\overline{\varphi} = 0{,}214$, $\overline{\psi} = 0{,}229$.

Application of the design method detailed in the chapter 3.1 resulted in the static and total pressure distributions, which depend on the circumstance and the radius. It allows calculating other important parameters of the fan. Variation of the properties of the velocity triangles along the radius in the above mentioned three points was analyzed during the measurements (every quantity was given in

dimensionless form). Quantity number and pressure number (φ, ψ), which varies along the radius, were shown in function of the relative radius ($R = r / r_k$).

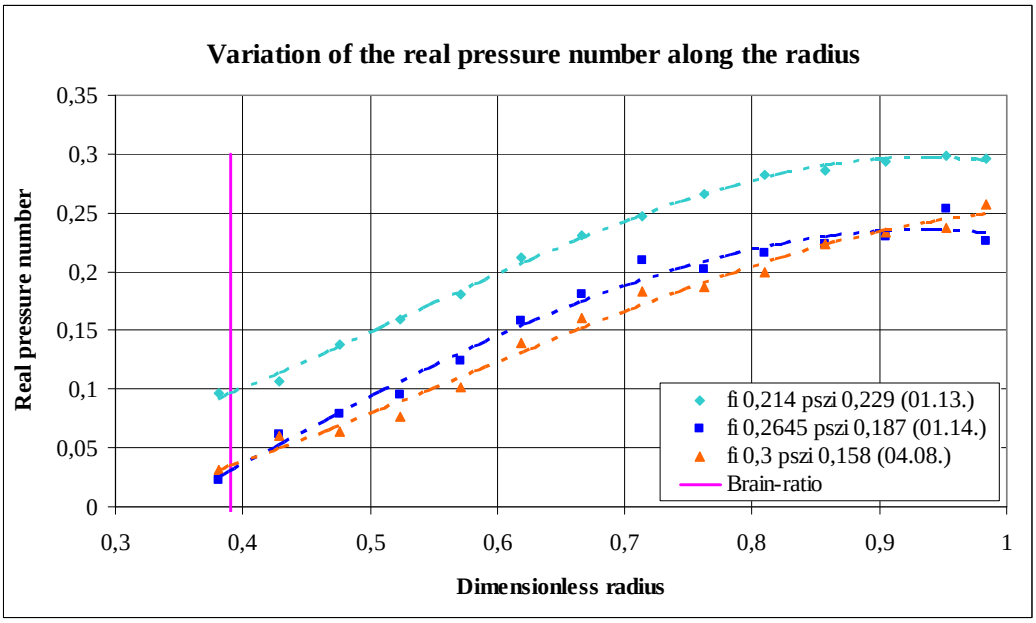
Variation of the quantity number along the radius can be seen in three different working points in Fig. 3.5. In case of high quantities ($\bar{\varphi} = 0{,}3$) the blade parts being on the inner radien transport relative high quantity, but along the circumstance there is some setback. In case of moderate quantities ($\bar{\varphi} = 0{,}2645$) the blade parts being on the inner radien transport the main part of the quantity, and the parts near the hub transport ever decreasing quantity. Backflow may even occur at radien below the hub-ratio, where the quantity number is negative.



„Benji”

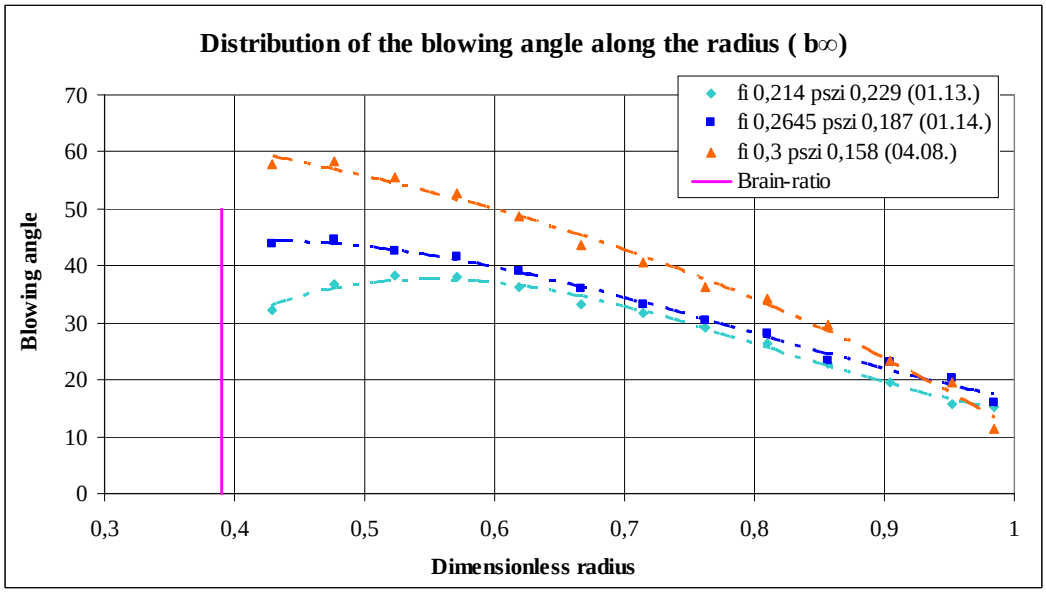
Fig 3.5 Variation of the measured quantity number along the radius

Pressure distribution along the radius can be seen in Fig. 3.6. As it can be seen, as lower is the quantity and higher is the pressure, as higher is the pressure distribution, progressively. The character of the distribution along the radius will not be modified. This value increases progressively outwards the hub, and it drops back only very near to the circumstance because of the wall and the gap. The pressure drops practically at zero near the hub. This distribution is tipical for the fans designed for variable circulation.



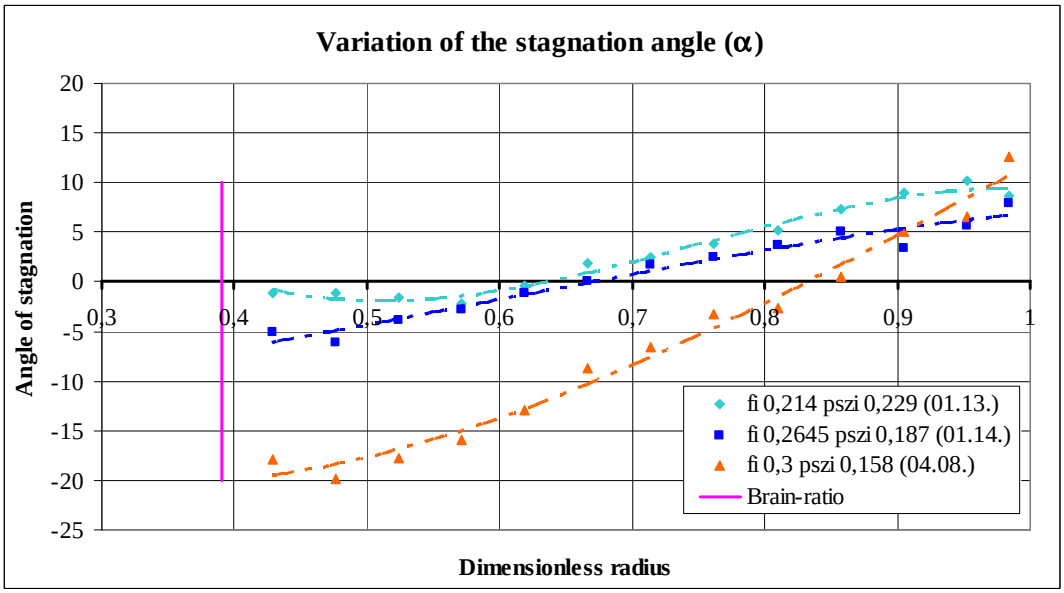
„Benji”

Figure 3.6 Variation of the measured pressure number along the radius



„Benji”

Figure 3.7 Variation of the measured relative blowing angle along the radius



„Benji”

Figure 3.8 Variation of the measured stagnation angle along the radius

Segments between 0.6 and 1 relative radius have realized high upthrust at the highest pressure range and with 10° stagnation angle.

Fine structure measurements were carried-out by the “Rotating” fan, as well. The results were similar to that of the “Benji” fan.

3.3 Flow simulation of the three bladed axial flow fans

Pressure- as well as velocity distributions can be seen in Fig. 3.9-3.12, which were recorded in the same plane right behind the fan, where the fine structure measurements were carried-out previously. Results were analyzed in all the three points chosen from the characteristic curve.

Q_1=2.2 m^3/s, p_1=150 Pa

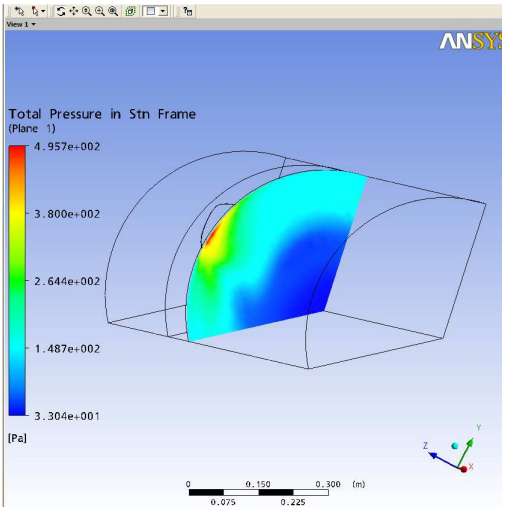
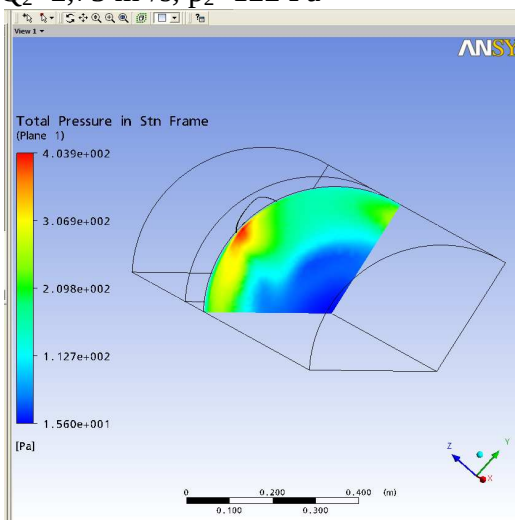


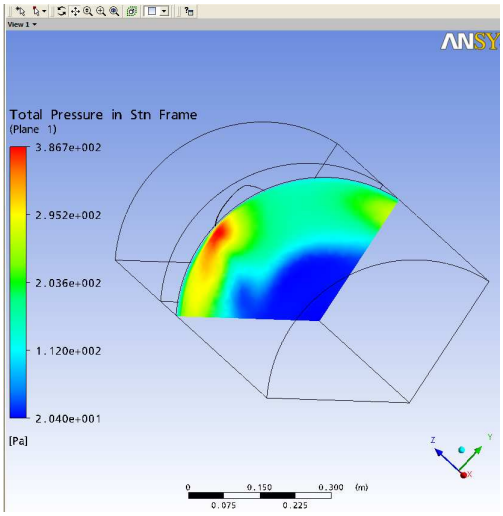
Figure 3.9 Pressure distribution right behind the impeller (φ=0.214, ψ=0.229)

Q_2=2,75 m^3/s, p_2=122 Pa



(φ=0,2645, ψ=0,187)

Q_3=3,05 m^3/s, p_3=103 Pa



(φ=0,3, ψ=0,158)

Figure 3.10 Pressure distribution right behind the impeller

Velocity distributions in the same planes like above can be seen in figures 3.11-3.12.-3.13.

Q_1=2.2 m^3/s, p_1=150 Pa

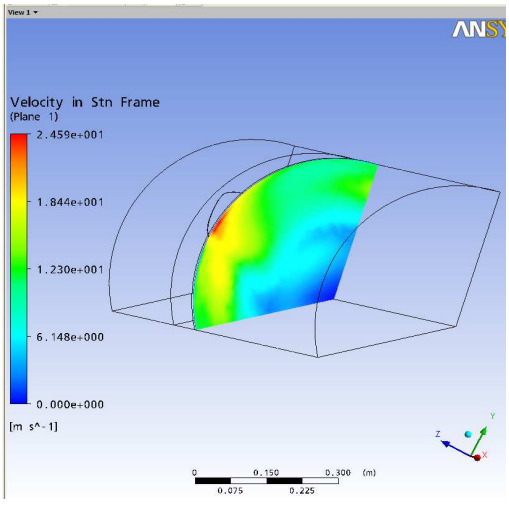
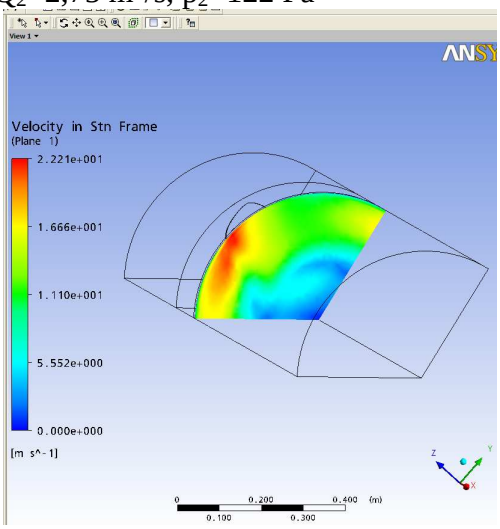


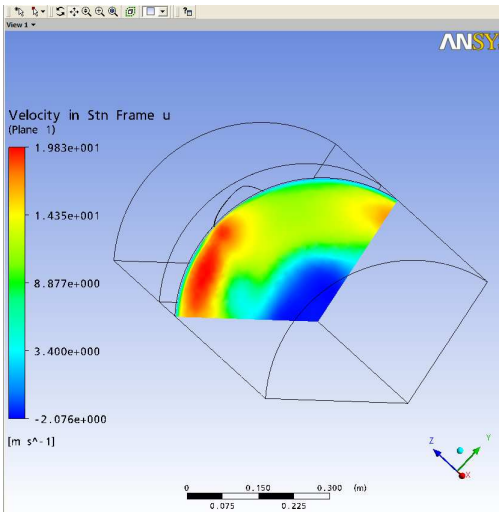
Figure 3.11 Velocity distribution right behind the impeller

Q_2=2,75 m^3/s, p_2=122 Pa



(φ=0,2645, ψ=0,187)

Q_3=3,05 m^3/s, p_3=103 Pa



(φ=0,3, ψ=0,158)

Figure 3.12 Velocity distribution right behind the impeller

3.4 Design process of the "Rotating" fan with the improved designing method

The main aim is to realize a blade geometry, which can be cut from a bended steel plate (simple production) with constant string length. Project of the desinging of the impeller of the "Rotating" fan is an industrial project, where the user should have the opportunity to vary freely the relative rotation of the impeller parts; hence the characteristic of the fan might vary between certain limits, as well.

During the design a pressure number variation function with two variables as well as variable efficiency was considered. The assumed pressure number- and efficiency distribution can be seen in Fig. 3.13. The solution of the differential equation resulted in the distribution of the quantity number. The program written according to the improved design method can be found in the M4 attachment of the dissertation. The program solves the differential equation described in chapter 2.2 and it calculates the geometrical properties of the blades. String length of the designed blade is 120 mm, which is constant along the radius. The adjustment angle is 25° right next to the hub, and it is 14° along the circumstance. For the sake of the simplicity the blade has constant bending angle (200 mm), therefore its sweep (7.55%) remains constant, too. The parametrized geometrical model of the blade was made in SolidEdge, where there is the possibility to draw the geometry on the basis of the resulted data from the M4 attachment of the dissertation.

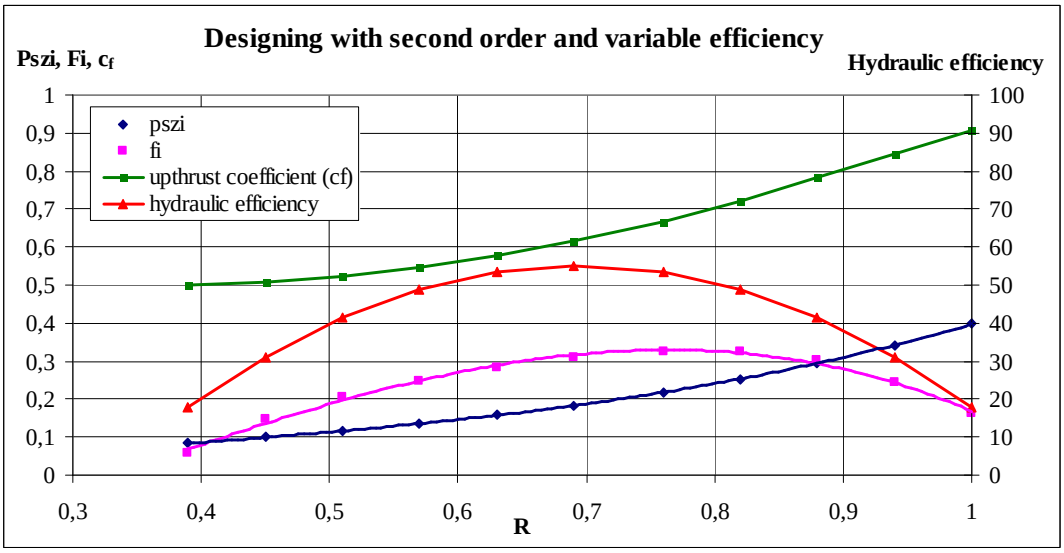


Figure 3.13 Variation of $\varphi(R)$, $\psi_{öi}(R)$, $\eta_h(R)$ in case of the "Rotating" fan

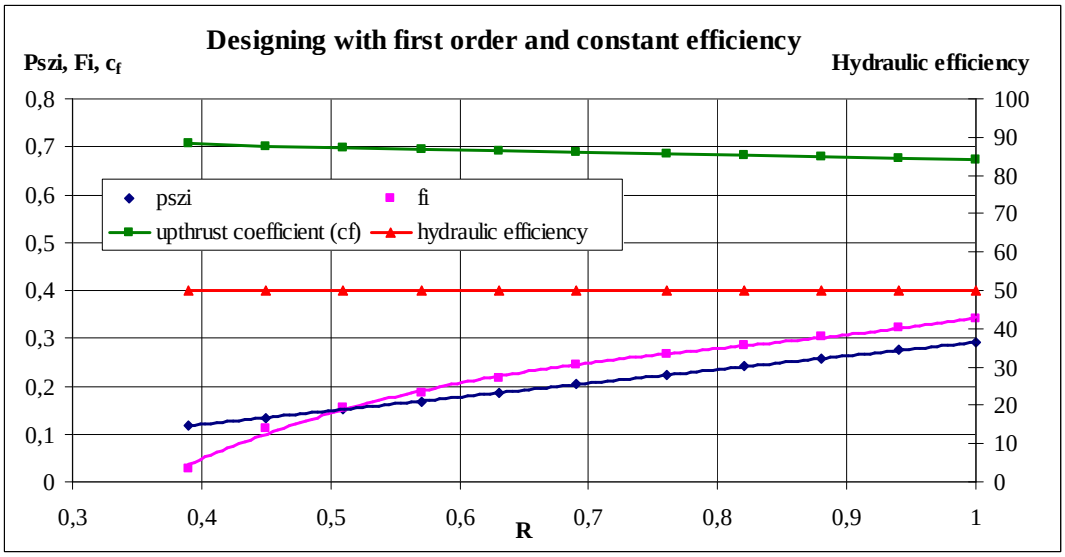


Figure 3.14 Variation of $\varphi(R)$, $\psi_{öi}(R)$, $\eta_h(R)$ of the "Rotating" fan

Result of the design method described by *Bencze-Szlivka* can be seen in Fig. 3.14. This method determines the curves of the pressure number, quantity number, upthrust coefficient and hydraulic efficiency in case of the same "Rotating" impeller. In this case the string length of the blade was 120 mm; moreover the adjustment angle was at the hub 25° furthermore 17° near the circumstance. The bending of the blade is constant (200 mm), and its sweep remains constant (7.55%) along the radius.

Quantity number near the hub is considerably higher by using of the second order method with variable efficiency than that of the first order method with constant efficiency. The biggest problem, which occurs during the design of variable circulation fans, is the high scale drop of the quantity number near the hub. This phenomenon promotes the stall-aptitude (see Fig. 3.14). To keep the string length constant, the c_f upthrust continuously grows from the circumstance to the hub, indeed this coefficient drops in case of the second order design. The difference between the two blades comes from the teta setting angle along the circumstance. The second order design method results in a blade with stronger bending along the radius, which gives a stronger bended string on the cylinder surface.

Figure 3.15 Blade geometry of the “Rotating” fan

3.5 Comparison of the measurement results with that of the simulation

In this section I am going to compare the results of the measurements and the simulation, respectively. In both cases was the MATLAB program applied for further data processing and chart drawing, because it handles huge amount of data very quickly and safe. Several m-files were written to process the data from the ANSYS as well as from the measurements, which can be seen in the M3 attachment of the dissertation. For the sake of simplicity only the 1/3 part of the 3-bladed impeller was analyzed in the ANSYS, and its results were multiplied for the whole 360°.

Pressure distribution field was made by the help of the measured pressure values, and its approached pressure field was shown in descartes- as well as in polar coordinate systems, respectively. Because of the better understanding basicly the charts with polar coordinate system are discussed, and all the other ones can be found in the M2 attachment of the dissertation.

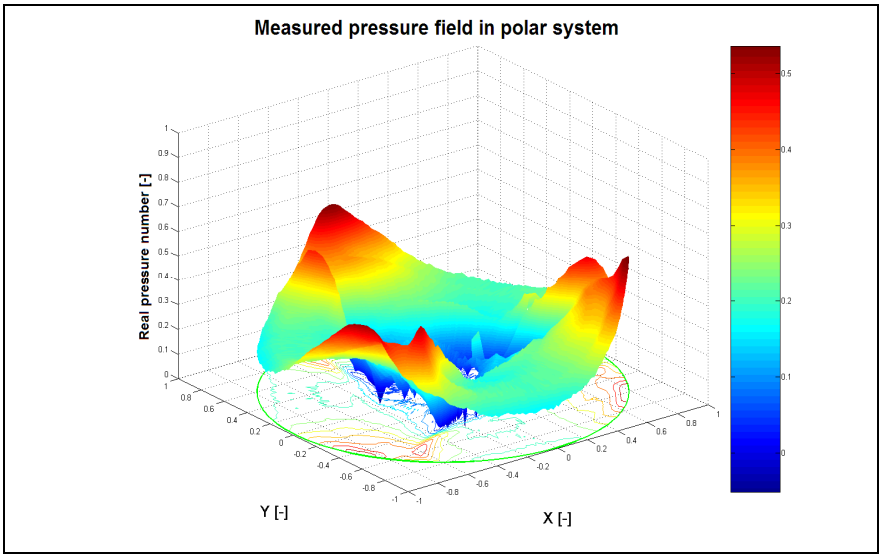


Figure 3.16 Pressure field behind the impeller of the „Benji" fan

The periodicity according to the blades can be seen unambiguously in Fig. 3.16. Pressure values were measured along the radius until y = 0.2 (which is already smaller than the hub size of the fan, ν = 0.39). It can be seen very well, that there is huge scale alteration along the circumstance; the passing of the blades occurs pressure rise, and the pressure drops when the blades go away. Drop of the angular momentum plays a key role in the establishment of the pressure.

Pressure behind the impeller was measured, as well. Axial and tangential velocity distributions behind the impeller can be seen in Fig. 3.17. Velocity distribution along the circumstance shows the leaving edge line, which locates closer to the gauge. Contour of the entering edge has impact just along the circumstance. These upcoming apices can be seen in the pressure distribution charts, too, but blade contours can not be recognized as sharp as in this case.

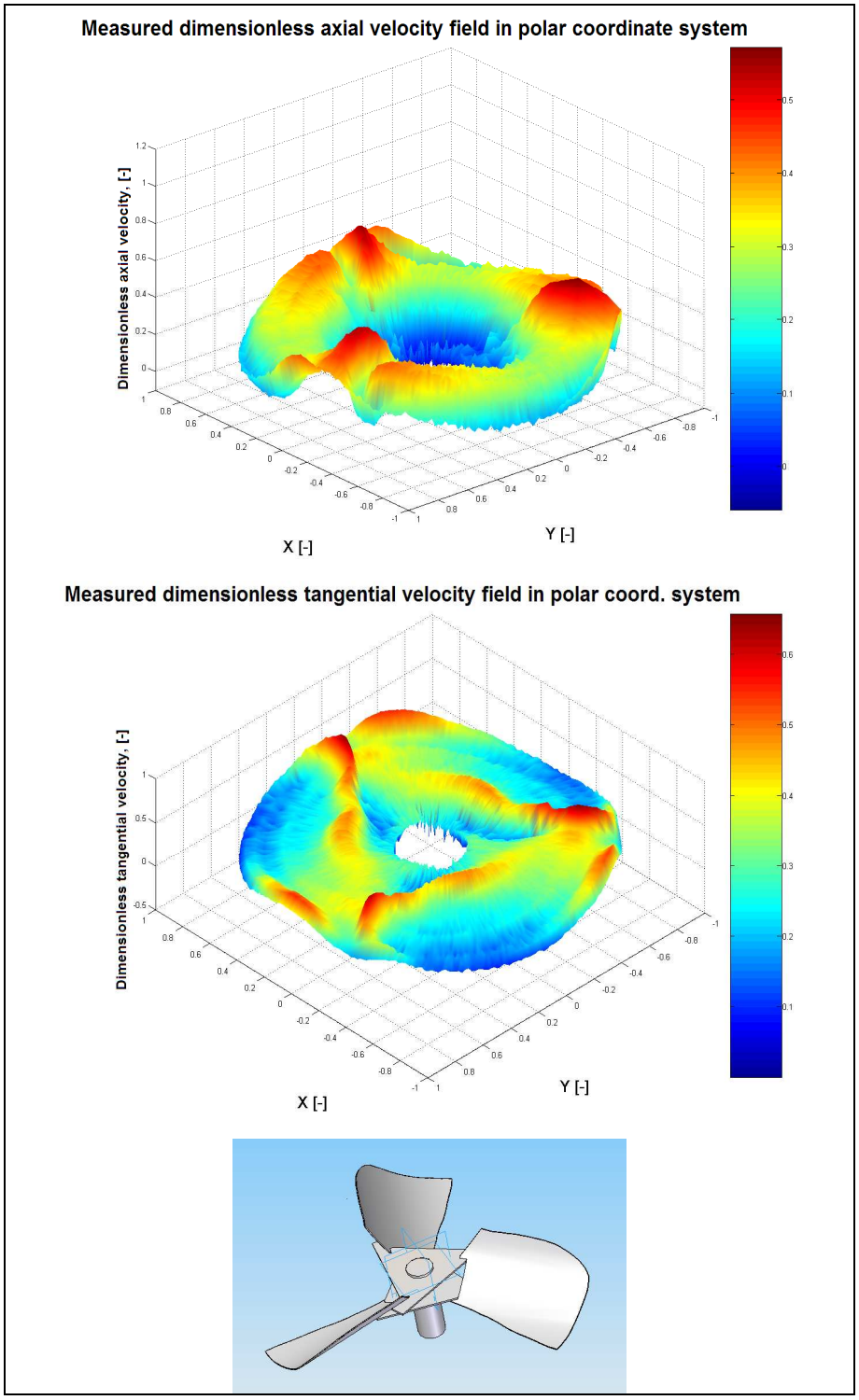


Figure 3.17 Measured velocity field behind the impeller of the „Benji" fan

Simulation results (made by ANSYS) of the impeller of the “Benji” fan with the previously mentioned operation point can be seen in Fig. 3.18-3.20.

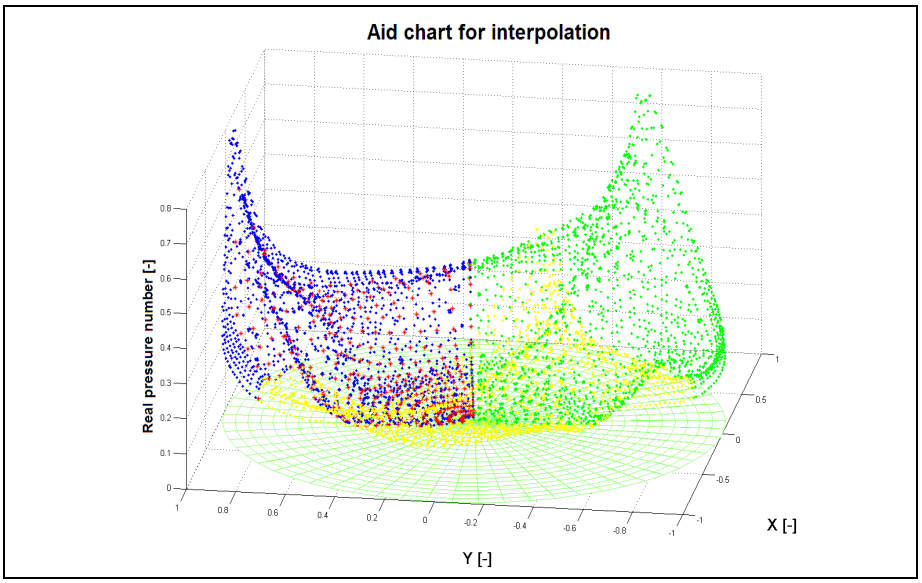


Figure 3.18 Calculated pressure field in polar coordinate system

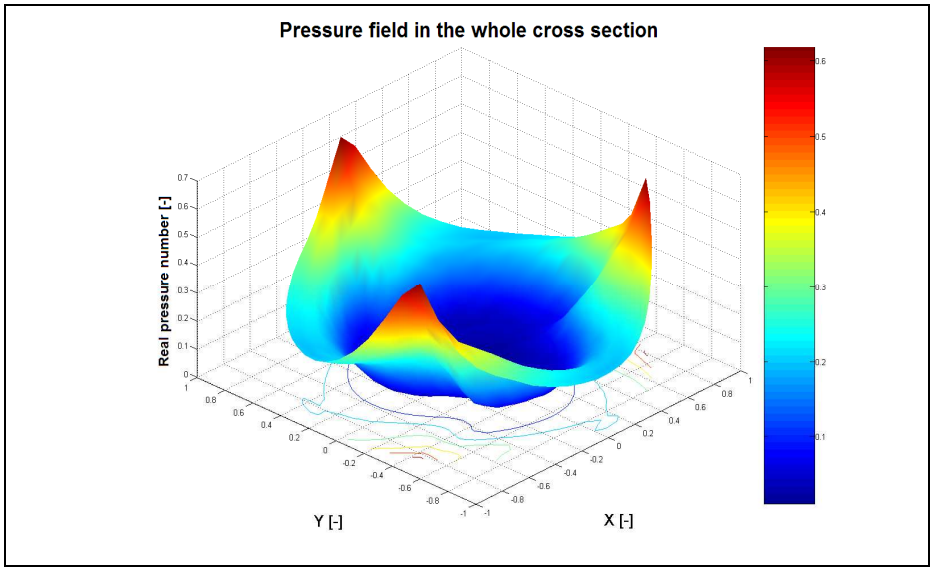


Figure 3.19 Pressure field behind the „Benji” fan (ANSYS result)

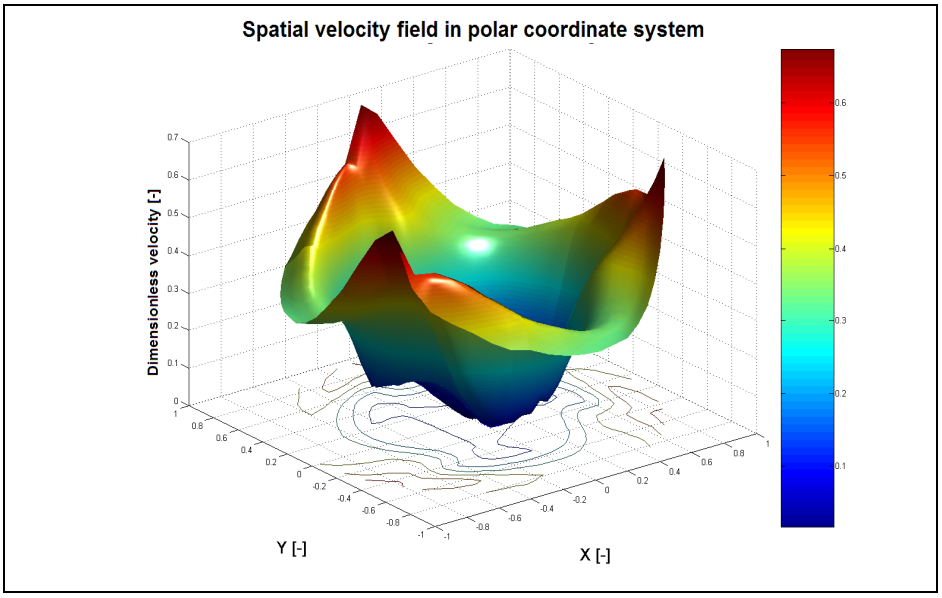


Figure 3.20 Velocity field behind the „Benji" fan (ANSYS result)

Comparison of the measurement data and the simulation results can be seen in Fig. 3.21 (loose grid represents the simulation result of ANSYS, and the dense one is for the measurement data), where both are shown in the same coordinate system. Henceforce this is the biggest advantage of the work with MATLAB, that the demonstration of data from different data sources could be unified, as shown. The difference between the two surfaces can be seen at the blade edges, which rise from its basis.

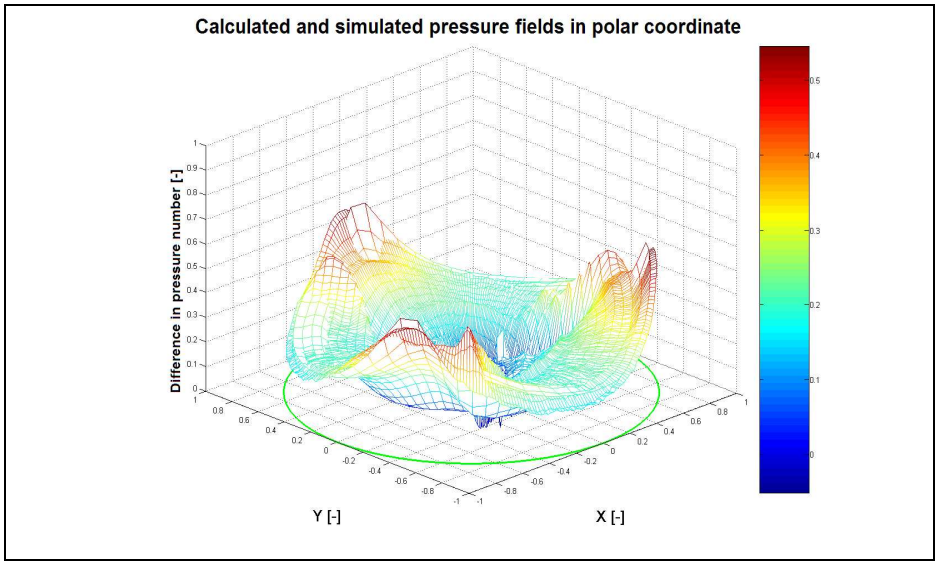


Figure 3.21 Comparison of measurment data and simulation results (k-ε turbulence model in ANSYS)

The k-ε turbulence model was applied during the ANSYS simulation in Fig. 3.21, and that one, where the SST turbulence model was used, can be seen in Fig. 3.22 (the scale of the vertical axle was changed for the sake of better understanding).

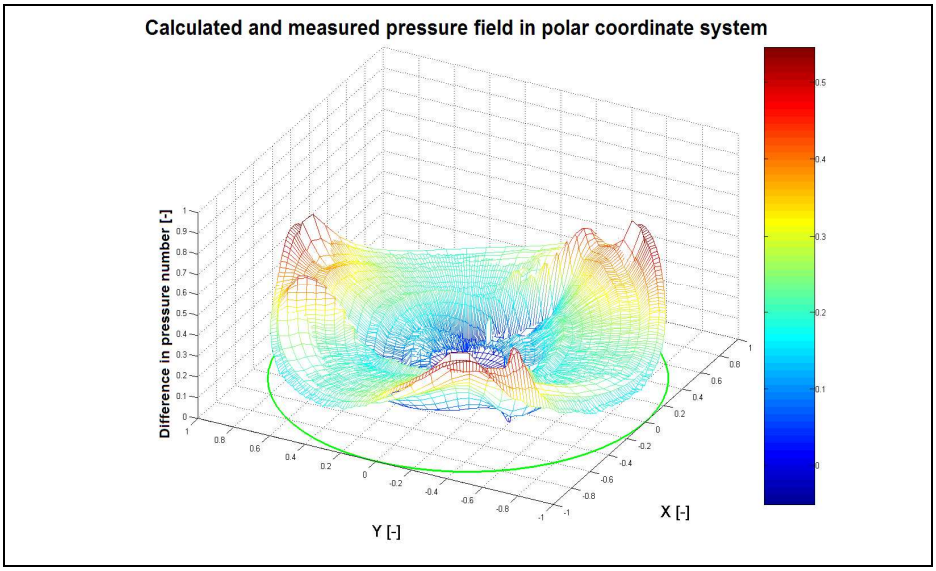


Figure 3.22 Comparison of measurment data and simulation results (SST turbulence model in ANSYS)

Until the measured or calculated pressure number reaches 0.7, the difference between the two values in the same location remains below 0.06, which means, that the difference between the measurement data and simulation results stays mainly under 10%. This error is acceptable respect on the given conditions, because only between the results of the different turbulence theories result in such an error.

4. SUMMARY

4.1 Summaty of my research activity

In my dissertation the corresponding professional literature was summarized, than the solutions for the aims were detailed.

Velocity- and pressure field mapping near to the impeller of an axial flow fan was realized by the help of the improved measurement method. The measurement method allows the determination of the pressure field close to the impeller, too. Spatial distribution and direction of the static- as well as total pressure were determined right in front of and behind the impeller by the above method (coordinate system binded to the blade channel). There is even the possibility to record the pressure maps in all the blade channels.

Knowing the losses of aerodynamic processes within fans (pressure losses in each point, hydraulic efficiency) the nowadays used design method has been improved. Advantages of the previous design processes were kept, and the change of the hydraulic efficiency along the radius was taken into consideration, furthermore the varying total pressure coefficient was fitted with a complicated polinom. Hence it allows designing blades with lower losses compared to that of the conventional method.

Stream simulation around the fan was developed in ANSYS-CFX. The simulation results were compared to the measurement data. Basic parameters, initial and boundary condition of the measurement serve as starting point for the simulation, as well; furthermore the results of the numerical simulations can be compared to the measurement data in a given location.

Another aim was to design such a blade geometry, which can be produced easily (cutting from cylindrical band sheet metal) and has constant string length, too. The geometrical model was developed in SolidEdge, where the applied parameters allow us to change its geometry quickly. The main feature of the design product is the two blade rows, which can be rotated for eac other (divisions between the blades are not stable). Position of each impeller part can be set freely by the user.

As it can be seen, this method is useful to analyze the change of the hydraulic efficiency along the radius. Measurement values allow drawing conclusion for the features of the impeller, moreover making proposal for improvement of the sizing method. Deviation of the pressure number depends on the initial sizing functions even somewhere outside of the previously set operation point.

4.2 New Scientific Results

1.) Measurement method of velocity- and pressure field behind the impeller blades was improved in this dissertation. Measurement sensors (total- and static pressure) were analyzed in point of accuracy, direction sensitivity and positionablility, as well. Tubular sensor proved to be more reliable then Pitot-tubes. The worst determination coefficient of Pitot-tube was R^2=0.7675, and that of the tubular sensor was R^2=0.9624.

2.) Maximal value and direction of the total pressure behind the blade was determined by my own algorithm, which gives a more accurate result then the previous applied solution. There were two chances to do the method more accurate: on the one hand the pressure sensor could be turned with lower steps and on the other hand a continual function could be fitted on the existing 10 points and a suitable regression analysis could result in the location and value of the maximal pressure. The number of the points as well as the measurement time would have been risen in case of the first solution, therefore the second one was chosen. A cubic regression method was used to determine the maximum value accurately, because its determination coefficient was the least. Substraction of the total- and the measured static pressure values gives the dynamic pressure value, then the velocity can be already calculated. Regression was carried-out with the following equation:

$$\Delta p = A\alpha^3+B\alpha^2+C\alpha+D$$

3.) In this dissertation the hydraulic efficiency – against several authors in the literature – during the measurements was not assumed constant. Measurement results prove that the efficiency varies along the radius. Efficiency variation was described by the following equation:

$$\eta = -A\left(R - \frac{1+\nu}{2}\right)^2 + \eta_{max}$$

4.) A design method for axial flow fans was developed, where the method was expanded with the respection of the hydraulic efficiency, which varies along the radius, indeed the function of the pressure number variation with two indexes was applied as follows: $\psi_{öi} = K_1 \cdot R^n - K_2 \cdot R^m$. This method gave appropriate solution. The developed computer program, which was developed for the above metioned calculations, could make the design phase shorter and easier.

The program solves the following differential equation:

$$2\varphi_3 \cdot \frac{\partial \varphi_3}{\partial R_3} = -\frac{\partial \psi_{öi}}{\partial R_3} \cdot \frac{\psi_{öi}}{2R_3^2} + \frac{\partial \eta}{\partial R_3} \cdot \psi_{öi} + \eta \cdot \frac{\partial \psi_{öi}}{\partial R_3},$$

and it calculates the geometrical data of the blade.

5.) During the solution the variation of the efficiency was taken into consideration, which resulted in lower decrease of the quantity number (φ) along the hub, where φ_v=0.11 was in case of varied efficiency and φ_v=0.09 in case of constant efficiency, so the difference was 20% (danger of stripping is higher at lower quantity number ranges). The design method of varied circulation with the use of the efficiency varied along the radius resulted in lower load on the blades near the hub.

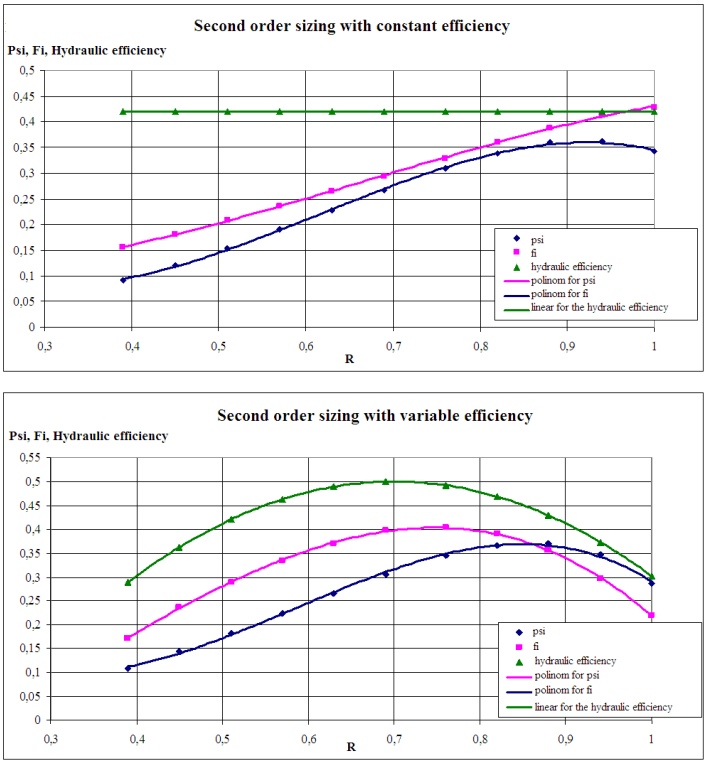


Figure 4.1 Variations of the $\varphi(R)$, $\psi_{öi}(R)$, $\eta_h(R)$ functions

6.) A parametrized method in the SolidEdge solid designing program was developed, which permits of stringing of the blades onto a cylinder. This method results in a foreward bended blade quite all the time, which can be produced easier, and it possesses better acustic properties, then the blades stringed along the radius. Earlier a special regression method was used for this sake, however the developed new design method is able to design and

draw automaticly the shape of the blade by the help of some corresponding previously calculated parameters.

7.) A new measurement method was developed; where the input torque of the fan was estimated by strain gauges through the deformation of the equipment. Industry could use it, as well, for on-site measurement purposes, because the fan does not require to be disassembled from the unit, moreover there is no need for a measurement shaft with strain gauges or other accessories. On-site calibration of this equipment was solved within the frame of this dissertation, too.

4.3 Practical applicability of the scientific results

Design method suggested by this dissertation calculates the varying total pressure coeffitient with a complicated polinom, which considers the change of the hydraulic efficiency along the radius. This method results in lower loss level of the impeller than the gap losses considered by the conventional sizing method.

Main tasks of this dissertation were the measurement and analysis of different impellers, sizing of simply fans, as well as carrying-out of Computational Fluid Dinamics (CFD) calculations. Further improvement of the designing method is possible by the CFD itself, which allows making proposals without any further measurement, indeed final test before production is necessary. There is just only 10% difference between the laboratory scale measurements and that of the FEM, which is an acceptable approach under the given conditions.

Industry does always FEM analyses between the design and test phases, which could reduce extremely the development time of the given product. A very detailed description about the FEM analysis of an axial flow fan is included in my dissertation, which could be very helpful for such people, who have the task to solve the above problem by his own. Therefore applying of FEM such way could reduce time and save money (design and production).

The presented analyses are useful on the field of food production, and could be more helpful in case of cooling houses, where mainly axial fans are used.

5. PROFESSIONAL PUBLICATION

Proofed articles in one of the World languages
MOLNÁR I. - DR. SZLIVKA F.: CONCLUSIONS OF THE MEASUREMENT DATA OF THE VELOCITY- AND PRESSURE DISTRIBUTION OF AN AXIAL FLOW FAN, HUNGARIAN AGRICULTURAL ENGINEERING, N°19/2006 PP. 41-42P.
FERENC SZLIVKA – **ILDIKÓ MOLNÁR**.: MEASURED AND NON-FREE VORTEX DESIGN RESULTS OF AXIAL FLOW FANS, JOURNAL OF MECHANICAL SCIENCE AND TECHNOLOGY, SPRINGER, 22 (2008) PP. 1902-1907P. (**IMPACT FACTOR 0.21**)
Proofed articles in Hungarian
MOLNÁR I.: VIZSGÁLATI ÉS MÉRÉSI MÓDSZEREK A MEZŐGAZDASÁG GÉPESÍTÉSE TÉMAKÖRÉBEN JELENTKEZŐ ÁRAMLÁSTANI PROBLÉMÁK MEGOLDÁSÁRA, MEZŐGAZDASÁGI TECHNIKA, XLVII. ÉVF., 2006, 40-41P.
MOLNÁR I.: EREDMÉNYEK A MEZŐGAZDASÁGBAN ALKALMAZHATÓ AXIÁL VENTILÁTOROK FEJLESZTÉSÉBEN, GÉP, LVII. ÉVF., 2006, 41-44P.
MOLNÁR I. - DR. SZLIVKA F.: AXIÁL VENTILÁTOR MÉRETEZÉSI ELJÁRÁSÁNAK KORREKCIÓJA, XXII. ÉPÜLETGÉPÉSZETI ÉS GÉPÉSZETI SZAKMAI NAPOK, DEBRECEN 2006
MOLNÁR I.: AXIÁL VENTILÁTOROK ÖSSZHATÁSFOKÁNAK MEGHATÁROZÁSA NYOMATÉKMÉRÉS SEGÍTSÉGÉVEL, GÉP, LVIII. ÉVF., 2007, 40-44P.
MOLNÁR I. – KAJTÁR P - DR. SZLIVKA F.: AXIÁLIS VENTILÁTOROK TERVEZÉSE, MEZŐGAZDASÁGI TECHNIKA, MEZŐGAZDASÁGI TECHNIKA, L. ÉVF., 2009, 2-4P
International Conference Proceedings
MOLNÁR I. - DR. SZLIVKA F.: OPERATIONAL PLANNING AND AERODYNAMICALLY CALIBRATION OF A CONVECTIVE DEEP BED GRAIN MODEL-DRIER, 4TH YOUTH IMEKO SYMPOSIUM, ON EXPERIMENTAL SOLID MECHANICS, CASTROCARO THERME, ITALY, 2005, 59-60P.
MOLNÁR I. - DR. SZLIVKA F.: PRESSURE AND VELOCITY FIELD PROPERTIES AROUND AN IMPELLER OF AXIAL FLOW FAN, PHD. HALLGATÓK V. NEMZETKÖZI DIÁKKÖRI KONFERENCIÁJA, MISKOLCI EGYETEM, 2005, 361-366P.
MOLNÁR I. - DR. SZLIVKA F.: MEASUREMENT METHOD OF VELOCITY AND PRESSURE FIELD AT AXIAL FLOW FANS, 5TH YOUTH SYMPOSIUM, ON EXPERIMENTAL SOLID MECHANICS, PÚCHOV, SLOVAKIA, 2006
MOLNÁR I. - DR. SZLIVKA F.: ANALYSIS OF DESIGN METHOD OF AXIAL FLOW FAN SIN THE VIEW OF THE MEASUREMENT RESULTS, INTERNATIONAL CONFERENCE ON FLUID AND THERMAL ENERGY CONVERSION, JAKARTA, INDONEZIA, 2006, 107-1-107-12P.
MOLNÁR I. - DR. SZLIVKA F.: DETERMINATION OF THE AXIAL FLOW FANS' EFFICIENCY, 6TH YOUTH SYMPOSIUM, ON EXPERIMENTAL SOLID MECHANICS, VRNJACKA BANJA, SERBIA, 2007, 229-230P.
MOLNÁR I. - DR. SZLIVKA F.: COMPARISON OF THE RESULTS OF MEASURED METHOD AND NON-FREE VORTEX DESIGN, JEJU, KOREA, 2007, 9-303P.

Proceedings in Hungarian
(discourse)
DR. SZLIVKA F. - **MOLNÁR I.**: AXIÁL VENTILÁTOROK MÉRETEZÉSÉNEK ÉS MÉRÉSÉNEK ÖSSZEHASONLÍTÁSA, DUNAÚJVÁROSI FŐISKOLA KÖZLEMÉNYEI, A MAGYAR TUDOMÁNY HETE 2005 – TUDÁSKÖZPONT DUNAÚJVÁROSI FŐISKOLA, 2005, 51-56P.
MOLNÁR I. - DR. SZLIVKA F.: AXIÁL VENTILÁTOR MÉRETEZÉSE A MÉRÉSI EREDMÉNYEK TÜKRÉBEN, XI. FIATAL MŰSZAKIAK TUDOMÁNYOS ÜLÉSSZAKA, KOLOZSVÁR, 2006, 351-354P.
MOLNÁR I. - DR. SZLIVKA F.: AXIÁL VENTILÁTOR KÜLÖNBÖZŐ MÉRETEZÉSI MÓDSZEREINEK ELEMZÉSE, A MAGYAR TUDOMÁNY HETE 2006 – TUDÁSKÖZPONT DUNAÚJVÁROSI FŐISKOLA, 2006
(poster)
DR. SZLIVKA F. - DR. BALLÓ B. - **MOLNÁR I.** POSTER SECTION, AXIÁLIS VENTILÁTOROK SEBESSÉG- ÉS NYOMÁSTERÉNEK MÉRÉSE, MTA AMB, K+F TANÁCSKOZÁS GÖDÖLLŐ, SZENT ISTVÁN EGYETEM 2005
MOLNÁR I. - DR. SZLIVKA F.: AXIÁL VENTILÁTOR KÖRNYEZETÉBEN MÉRT SEBESSÉG- ÉS NYOMÁSELOSZLÁSBÓL LEVONHATÓ KÖVETKEZTETÉSEK, MTA AMB, K+F TANÁCSKOZÁS GÖDÖLLŐ, SZENT ISTVÁN EGYETEM, 2006
MOLNÁR I. - DR. SZLIVKA F.: AXIÁLIS VENTILÁTOROK VÁLTOZÓ CIRKULÁCIÓRA TÖRTÉNŐ MÉRETEZÉSI ELJÁRÁSÁNAK TOVÁBBFEJLESZTÉSE, MTA AMB, K+F TANÁCSKOZÁS GÖDÖLLŐ, SZENT ISTVÁN EGYETEM, 2007
Staff member of the organization comitee of a conference
KONZULENS **MOLNÁR I.**: AXIÁL VENTILÁTOR NYOMATÉKMÉRÉSÉNEK KIDOLGOZÁSA – GODA ADRIENN, TDK DOLGOZAT, 2006