# Design of an actuation system for a variable pitch axial fan

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## ABSTRACT

This paper investigates the mechanical design of an axial fan with automatically controlled variable pitch blades, focusing on kinematic and dynamic synthesis of the fan. In addiction it is presented a case of an axial-flow fan with 16 adjustable blades able to settle the air flow rate in a power plant. The dimension of the external hub of the fan is 3.2 [m] and its maximum work velocity up to 1000 rmp.

**keywords:** axial fun, variable pitch blade.

## 1. INTRODUCTION

In the last few years the energy problem has become more and more relevant. Besides to the research and the development of new energy sources, the optimization of the existing tools has became a strategic element, especially the optimization of performance of fluid machine. Many examples of axial-flow fans (single or multi stage design) with variable blade pitch angle can be found in energy production field such as in thermal power stations: the two induced-draft fans at the Weiher and Bexbach power stations are among the largest in the world [1] [2].

Fluid-Dynamic analysis of axial-flow fans has been studied for about 30 years [3] [4] [6]. The decision to develop this type of fan was prompted not only by its easily integrable design into overall plant configurations but also and primarily by the operating cost benefits that it offers, specifically when compared with centrifugal fans with variable inlet vanes [5].

This paper investigates the mechanical design of this kind of fans focusing with the kinematic and dynamic synthesis of variable pitch axial flow fans. Different mechanisms to achieve blades regulation are compared and the effects on the overall design of the fan are evaluated. The mechanical design of the fan and the actuation system is considered, neglecting the aerodynamic aspects which are considered only to define the forces applied to each blade. We present a case of an axial-flow fan with 16 adjustable blades to settle the rate of flow. The fan operates as an air supplier system in combustion chamber for energy production. The most relevant operating conditions regard the angular speed 1000 rmp, the starting time of the fan (less than 10 s) and the blade pitch regulation ($\pm 30°$).

Great attention has been paid to the design of the actuation system. It is able to move the blade quickly and accurately, therefore the position of the actuator and the type of supply are evaluated. Moreover, the mechanisms which transform the motion of the actuator in the blade rotation are investigated. An important issue is represented by the choice of blade-shaft bearings and their position, since the centrifugal force can reach more than 50 tonnes for blade. Once the best solution to assembly each blade-shaft on the impeller and to allow an easy maintenance to be made periodically is defined, it is studied the balancing of the blade-shafts by means of appropriate counterweights disposition which is aimed to reduce inertial effects on the system.

Finally in order to compute the transmitted forces and then to size the components, a multi-body model of the fan has been created. This model allows to investigate the problem of balancing blade-shaft and the influence of friction on bearings and aerodynamic forces in the determination of loads on the system. Numerical simulations have been carried out to support the design and highlight project criticalities.

## 2. ACTUATION SYSTEM: KINEMATIC SYNTHESIS

The pitch blade regulation of an axial-flow fan can be realized with different solution, which can be classified according to the way the motion of the actuator is transformed into rotation of the blades. Each blade is connected to the impeller, so they rotate with it; the blade actuation system can be joined to the impeller or to the ground. Depending on this choice, we can describe these situations:

- in the case the actuator is connected to the impeller, the problem of suppling the actuation system must be solved;
- in the case the actuator is connected to the ground, the relative velocity between the actuator system and the impeller must be added up to zero.

If we choose to have the actuator rigidly linked with the impeller, the force between the impeller and the mechanism to rotate the blades is internal; in this case, it is easier to design the structure and balance the fan, since the forces of the mechanism are not transmitted to the basement. The motion of the actuator can be linear or rotative; in the first case it is necessary to create a mechanism to

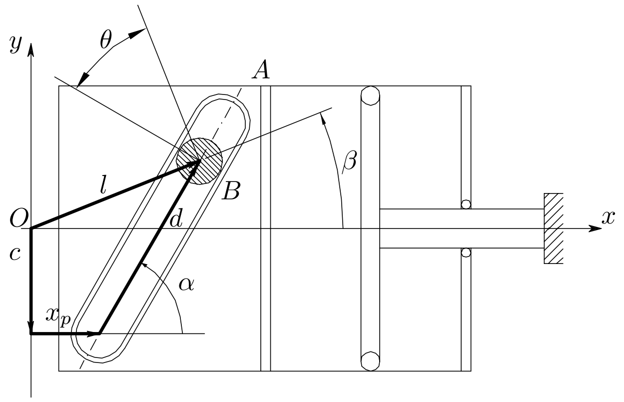


Figure 1: Cinematic draft of *plate and cam configuration*.

convert the linear motion of the actuator in the rotation of the blades. From this analysis, it is clear that two possible kinematic solutions can be used: a rotative actuator linked to the impeller (in this case a motor for each blade or a system of gears is needed) or a linear actuator rigidly constrained to the impeller and a mechanism to convert the motion. Using a motor for each blade, the system can be subject to failure; while, using a system of gears to move all blade with only one motor, it is necessary remove the backslash and plenty lubricate.

The simplest and most suitable choice for this application is a linear actuator with a system to convert the motion. The available technologies for the actuation are electrical, pneumatic or oil-hydraulic. Pneumatic and electromechanical systems play virtually no role in power plant fan engineering, while mechanical control systems of blade pitch used to be employed specifically on smaller units; oil-hydraulic control systems have emerged as the most suitable solution for this purpose. They operate with less hysteresis since they use fewer mechanical power transmission elements; in addition, they are capable of transmitting higher actuating forces. The oil-hydraulic solution grants greater dynamic performances and a better rate weight-power than the other two solutions. The chosen solution for the design of the fan is then a linear oil-hydraulic actuator rigidly linked to the impeller with a mechanism to convert the motion.

### Mechanism of motion conversion

The choice of the mechanism of motion conversion is important to determinate the amplitude of the forces between the blade-shaft and the actuator; moreover it defines the relationship between the regulation parameter of the actuator and the angular position of the blade. For this purpose, a mechanism which behaves as a speed reducer (and therefore of position) could minimize the load on the actuator and at the same time it can modulate the action of the actuator itself.

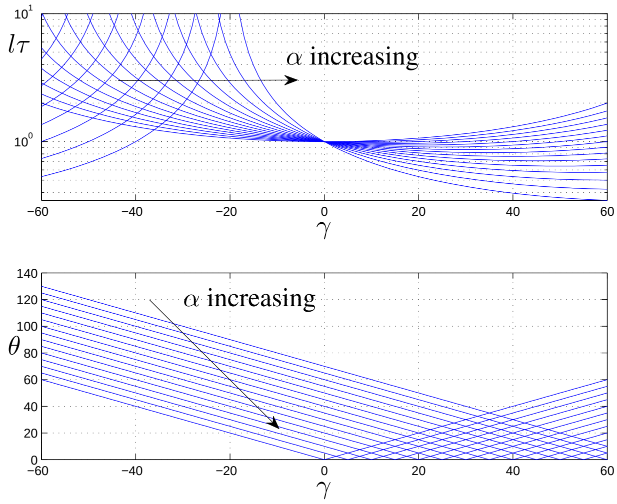


Figure 2: Absolute value of the generalized transmission rate and pressure angle.

The simplest system to convert a linear motion (actuator) to a rotative motion (blade) is obtained by a Scotch yoke mechanism. The piston is directly coupled to a sliding yoke with a slot that engages a pin on the rotating part. In figure 1 it is shown a generic kinematic draft. The cursor $A$ which is rigidly linked to the oil-hydraulic cylinder imposes a rotation to the crank which is connected to the blade-shaft through a prismatic link $B$. The direction which the point $B$ can move determines if the mechanism is centered ($\alpha = 90°$) or deflected one ($\alpha \neq 90°$). If the Scotch yoke mechanism is centered, the shape of the motion of the link $l$ is constant over the time given a pure sine wave speed of the piston. With reference to figure 1, we can describe the following kinematic relations:

$$x_p = le^{i\beta} - ci - de^{i\alpha} \tag{1}$$

$$\dot{x}_p = i\dot{\beta}le^{i\beta} - \dot{d}e^{i\alpha} \tag{2}$$

$$\ddot{x_p} = il\ddot{\beta}e^{i\beta} - l\dot{\beta}^2e^{i\beta} - \ddot{d}e^{i\alpha} \tag{3}$$

where $O$ is the rotational axis of the blade, $x_p$ is the advance direction of the plate, $l$ is the length of the crank and $\beta$ is the rotation angle of the blade.

From the relation (2) it is possible to derive the generalized transmission rate $\tau$, that is the rate between the angular velocity of the blade and the translational piston speed:

$$\dot{x_p}\left[\frac{-\sin\alpha}{l\cos(\alpha-\beta)}\right] = \dot{\beta} \quad \rightarrow \quad \tau = \frac{-\sin\alpha}{l\cos(\alpha-\beta)} \tag{4}$$

In the upper graphic of the figure 2, the Absolute value of the generalized transmission rate $\tau$ adimensionalized with length $l$ is plotted versus $\gamma = (90° - \beta)$ for $20° < \alpha < 90°$; this graph using a base 10 logarithmic scale

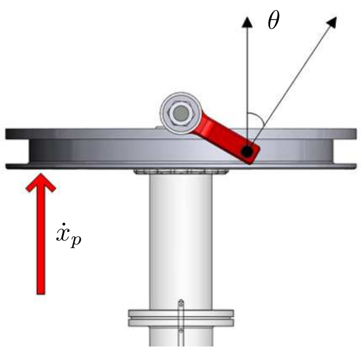


Figure 3: *Plate mechanism.*

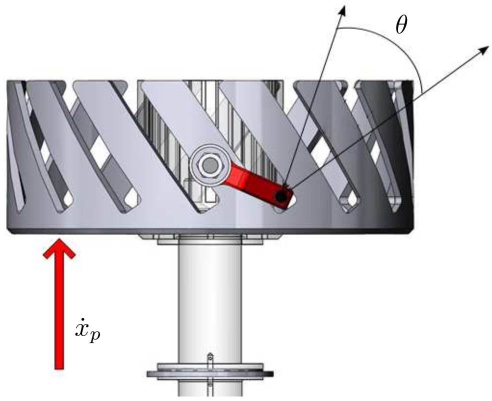


Figure 4: *Cam mechanism.*

for the y-axis. In the lower graph the absolute value of the pressure angle is plotted versus $\gamma$ for the same previous configurations. The pressure angle is given simply by: $\theta = \alpha - \beta$. In figure 2 the influence of the angle $\alpha$ on the transmission rate and therefore on the pressure angle can be observed.

Figure 3 shows the kinematic solution with the centered Scotch yoke mechanism which has been designed for this fan. The blade-axis is designed with a crank at the end of which a small wheel is mounted. This wheel is constrained between two plates which can translates along the rotational axis of the impeller; this solution is called *plate mechanism* because two plates, by a crank, guides the blade shaft position. The angle $\theta$ in figure 3 is the pressure angle.

In order to realize the kinematic solution with a uncentered Scotch yoke mechanism, the crank linked to the blade-shaft is moved through the contact with a conjugate profile, as shown in figure 4. In the proposed solution, the conjugate profile is simply a straight line with a slope of $150°$ from the rotational axis of the impeller. It can be observed, during the design phase, the cam profile can be modified to obtain a variable transmission rate as in the case of spatial cams. For this reason this solution is called *cam mechanism*.

We can immediately observe that with the plate solution the blade rotation has to be symmetrical with respect to the position $\beta = 90°$ and it is impossible to have a rotation greater than $60°$ without worsening the pressure angle. The cam solution turns out to be more flexible and it is possible to set the global rotation $\Delta\beta$ without limitations for symmetry. Moreover the maximum achievable rotation can be increased defining the profile shape. On the other side, the cam solution is more complex to be realized, as shown in figure 4, and the complete mechanism is more cumbersome.

Because of the dimension of the fan and the number of blades, the *plate solution* has been preferred. It can be observed that the *cam mechanism* is more suitable for fans with high number of blades, since the cranks would be aligned with the rotational axis of the impeller, reducing the lateral dimensions that they would have during the motion.

## 3. Blade dynamic balancing

The chosen solution leads each blade-axis to be unbalanced, with regards to its axis of rotation, due to the presence of the crank: it is impossible to create a crack which is opposite to the command one, since it would interfere with the command crank of the adjacent blade axis. As consequence the bearings are subjected to loads and torques which tend to rotate the blade. To reduce this two effects, first of all it is necessary to move the center of mass on the rotational axes of the blade; in this way it is possible to align a principal axis of inertia with the rotational axes of the blade. The second topic we can focus on regards differences between the other two inertial axis which are responsible for a torque around the rotational axes. To compute this torque it is possible to integrate the effect of each point $P$ of the blade, which is characterized by a mass of $\rho dV$, where $\rho$ is the density of the blade and $dV$ is the infinitesimal volume. The velocity of the generic point $P$ is:

$$\underline{v_p} = \underline{\dot{r}_0} - [\hat{r}]\,\underline{\omega} \tag{5}$$

where $\underline{\dot{r}_0}$ is the velocity of the center of coordinate system "0", against which we compute the moments of inertia, $\underline{\omega}$ is the angular velocity of the point $P$ and $[\hat{r}]$ is the matrix which contains the position of the point $P$ in the reference system "0", matrix $[\hat{r}]$ is defined as:

$$[\hat{r}] = \begin{bmatrix} 0 & -r_z & r_y \\ r_z & -r_y & -r_x \\ -r_y & -r_x & 0 \end{bmatrix} \tag{6}$$

Since the body is rigid, the terms of the matrix$[\hat{r}]$ are constant. Forces and torques are calculated through the inte-

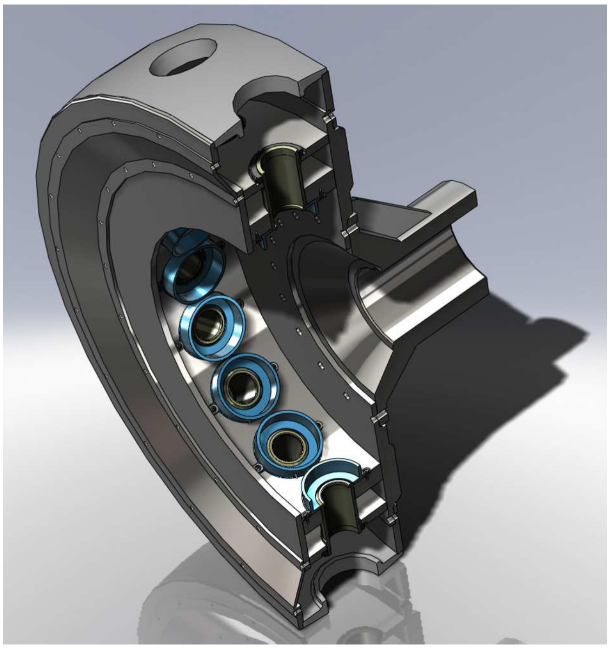

Figure 5: The impeller.

gration of infinitesimal element of mass $\rho dV$, multiplied by the acceleration $a_P$ of each point $P$; thus it results:

$$\left\{ \begin{array}{c} \underline{F} \\ \underline{C} \end{array} \right\} = \left[ \begin{array}{c} \int_V \rho \underline{a}_p \\ \int_V \rho \left[\hat{r}\right] \underline{a}_p \end{array} \right] \tag{7}$$

These can also be written as:

$$\left\{ \begin{array}{c} \underline{F} \\ \underline{C} \end{array} \right\} = \left[ \begin{array}{cc} [M] & [S] \\ [S]^T & [J] \end{array} \right] \left\{ \begin{array}{c} \ddot{r_0} \\ \underline{\dot{\omega}} \end{array} \right\} + \left\{ \begin{array}{c} [\underline{\hat{\omega}}]\,[S]\,\underline{\omega} \\ [\underline{\hat{\omega}}]\,[J]\,\underline{\omega} \end{array} \right\} \tag{8}$$

Referring forces and torques to the center of mass, one gets:

$$\left\{ \begin{array}{c} \underline{F} = [M]\,\{\underline{\ddot{r}}_{CG}\} \\ \underline{C} = [J]\,\{\underline{\dot{\omega}}\} - [\underline{\hat{\omega}}]\,[J]\,\underline{\omega} \end{array} \right\} \tag{9}$$

$$= \left[ \begin{array}{cc} [M] & [S] \\ [S]^T & [J] \end{array} \right] \left\{ \begin{array}{c} \ddot{r_0} \\ \underline{\dot{\omega}} \end{array} \right\} + \left\{ \begin{array}{c} [\underline{\hat{\omega}}]\,[S]\,\underline{\omega} \\ [\underline{\hat{\omega}}]\,[J]\,\underline{\omega} \end{array} \right\} \tag{10}$$

Considering the matrix $[J]$ diagonal and one of the principal axises of inertia ($z^*$) aligned with the rotational axis of the blade ($z$), can be defined the angle $\alpha$ as the angle between the principal axis ($x^*$) and the rotational axis of the impeller ($x$). Calling $\dot{\Theta}$ the rotational velocity of the impeller around axis $x$ and $\dot{\Psi}$ the rotational velocity of the blade around axis $z$.

The angular velocity vector of the blade-shaft body is:

$$\underline{\omega} = \left[\dot{\Theta} cos\alpha, \quad -\dot{\Theta} sin\alpha, \quad 0\right] \tag{11}$$

The vector of torques applied in the center of mass is:

$$\underline{C} = -\,[J]\,\{\underline{\dot{\omega}}\} - [\hat{\omega}]\,[J]\,\{\underline{\omega}\{ \tag{12}$$

where:

$$[\hat{\omega}] = \left[ \begin{array}{ccc} 0 & -\omega_z & \omega_y \\ \omega_z & -\omega_y & -\omega_x \\ -\omega_y & -\omega_x & 0 \end{array} \right]$$

Deriving $\underline{\omega}$ and replacing $[J]$ with the diagonal matrix of inertia:

$$\left[ \begin{array}{ccc} I_{x^*} & 0 & 0 \\ 0 & I_{y^*} & 0 \\ 0 & 0 & I_{z^*} \end{array} \right] \tag{13}$$

one gets: $C_z = \dot{\Theta}^2 \sin\alpha \cos\alpha (I_{x^*} - I_{y^*})$ and $C_x = Cy = 0$. The difference between the two moments of inertia of the blade in the rotational plane of the blade itself generates a torque that makes the blade rotate. This torque tries to align the greater moment of inertia between $I_{x^*}$ and $I_{y^*}$ along the rotation axis of the impeller. Because of the complex geometry of the blade-shaft, the definition of the balancing mass is not easy; it is necessary to identify suitable areas to place the counterweights and to define their mass with numerical optimization algorithms.

## 4. MECHANICAL SYSTEM DESIGN

Once the kinematic solution and the actuation system have been chosen, the mechanical design of the axial fan can be discussed. The designed solution includes an impeller made of three concentric rings. Each of them is provided with holes, in which the blade-shaft are inserted, as shown in figure 5.

The inner ring is the most critical component, because it has to bear all the centrifugal load of the blades. Moreover the inner ring together with the intermediate one has to bear the radial loads which are responsible of keeping the blade shaft in its bearing housing. The external surface of the third ring has a spherical shape with a diameter of 1920 [mm] letting the blade to rotate without interfering with the impeller and keeping the distance from the base of the blade constant.

Figure 6 shows the designed blade-axis. The blade is constrained to the shaft with six screws M20. On the same

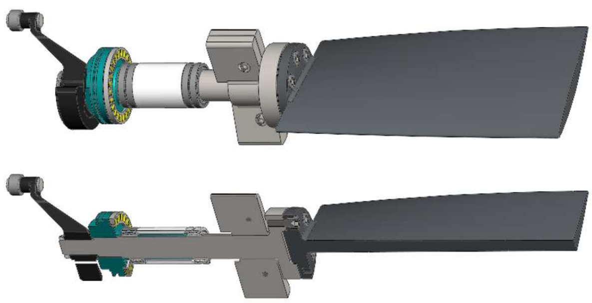

Figure 6: The blade.

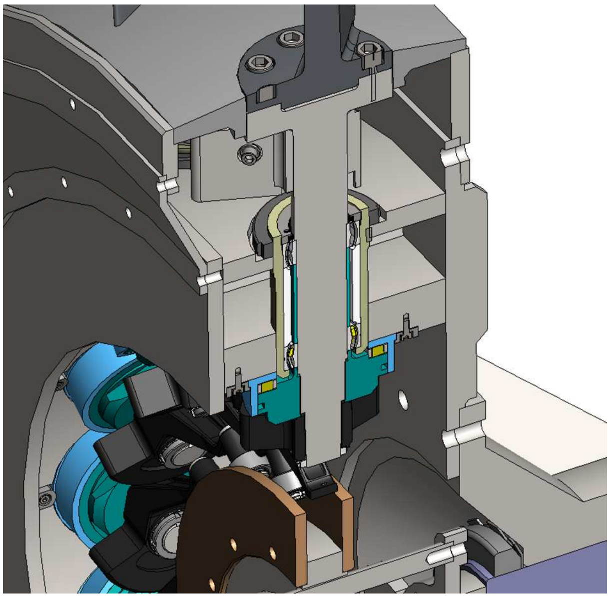

Figure 7: Blade assembly axis.

shaft a counterweight, made by two shells, and two radial spherical roller bearings are assembled. At the bottom of the shaft, an axial roller bearing and its supports are screwed to the shaft itself. At the end of the shaft the crank is assembled. The blade shaft, without the crank and the support of the axial bearing, is inserted in a cylindrical element, in which the radial bearing housings are obtained. This solution makes the assembly of the blade independent from the assembly of the impeller. Moreover it is possible to balance each blade-shaft individually and do maintenance in a simple way. The position of the cylindrical element is defined by a mobile mechanical stop which

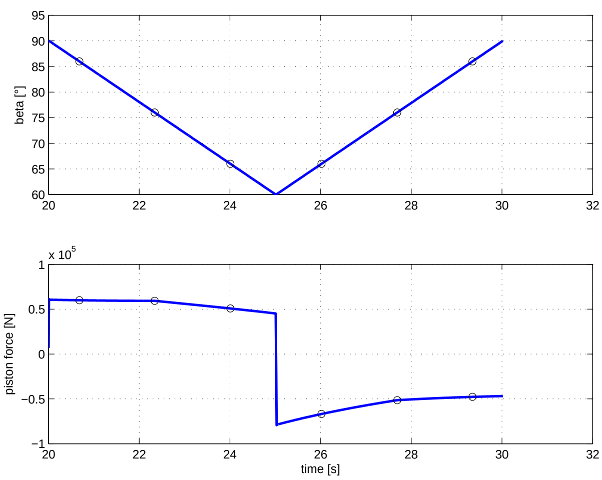


Figure 8: Piston force.

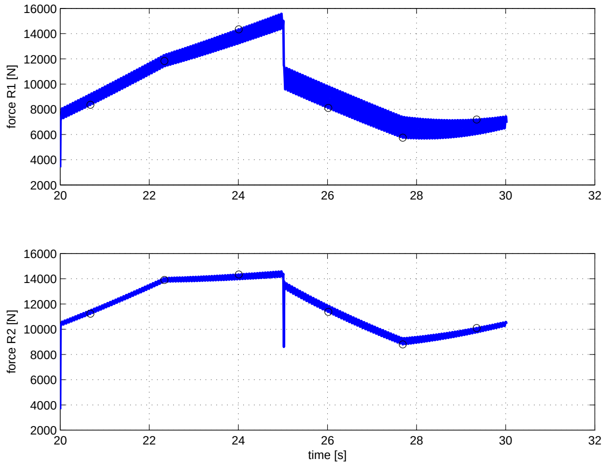


Figure 9: Absolute value of the reaction force $R_1$ and $R_2$ (internal and external bearing of the blade shaft).

is made by a threated ring which preload the axial bearing as shown in figure 7. It can be also observed that the lubrication system is realized thanks to a single grease nipple. The blade is made of aluminum and it is 640 [mm] high; between the base and the tip there is a twisting of 20○. The weight of the blade is 16 [kg], while the blade and the shaft with the radial bearings housings weighs 62 [kg]. Figure 10 shows the full-assembled fan. The stem of the oil-hydraulic actuator is rigidly linked to the impeller, while the sleeve is mobile. On the sleeve two plate are mounted: they have to guide the wheel on the tip of each crank.

## 5. MULTIBODY MODEL AND RESULTS

To evaluate the loads on the bearings on the most stressed mechanical parts and to estimate the force that the actuator must be able to supply, a multibody model of the fan has been created with the simulation environment Matlab-SimMechanics, in which the aerodynamic forces were included. These are available [1] for some areas of operation and for some values of the blade rotation. Intermediate values were obtained by interpolation.The friction model due to the bearings have been obtained applying a torque in opposition to the blade rotation in correspondence of the axial bearing. Friction torque is calculated with relation and coefficients of the chosen bearings manufacturer: $C_{\text{att}} = K \cdot f_{\text{SKF}} \cdot r_m \cdot F_a$. Where $f_{\text{SKF}} = 0.005$ is the friction coefficient obtained from the catalogue; $r_m$ is the internal radius of the axial bearing of 60 [mm]; $F_a$ is the force in the direction of the blade-shaft axis. It can be observed that in the previous relation a safety coefficient has

[1] The blade profile has been obtained from the dynamic optimization on the base of the fan requirements

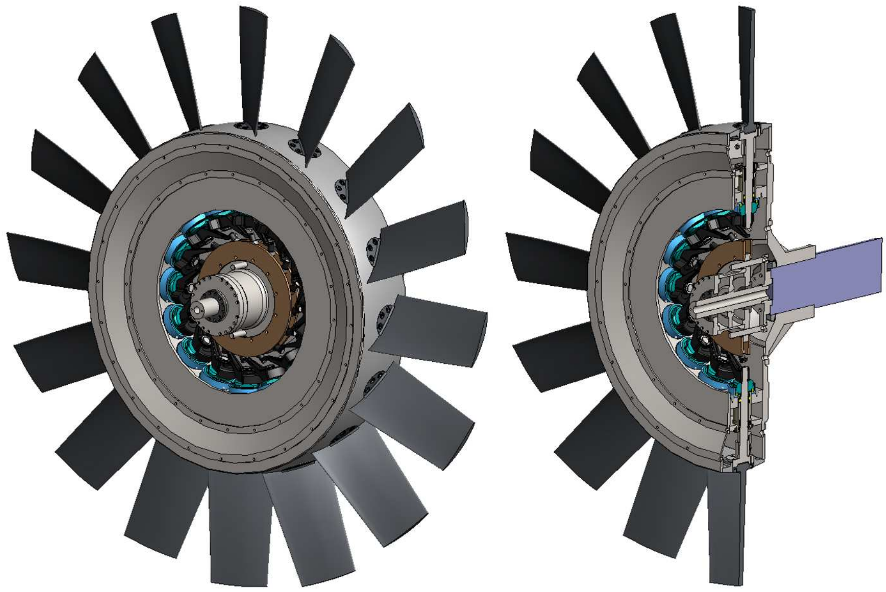

Figure 10: Complete draw of the fan.

been introduced ($K = 3$) to consider friction phenomena due to assembly inaccuracies or due to unconsidered deformations or friction (such as frictions between the crank and its wheel or between the wheel and the plate).

Some results obtained by the multibody model are reported in this section. In particular in the simulations the impeller starts from a velocity of 0 [rpm] and in 10 seconds it reaches 1000 [rpm]. In this phase the blades are kept closed, until the operating speed is reached, in this way aerodynamic forces can be neglected and the reaction forces on the impeller bearings are constant.

When the rotor has reached the operating velocity, the blades are opened. For example we report a regulation phase around the operating point (figure 8), which corresponds to the orientation of the crank $\beta$ of $76°$. In the range of $\pm 30°$ it is possible to interpolate the values of the known aerodynamic forces. Since the regulation of the angle of incidence of blade is very slow, the inertial torque which is generated by the rotation of the blade among its own axis is negligible if compared to the torque due to the aerodynamic effects and the friction torque.

The graphics in figures 8 and 9 show respectively the force provided by the actuator and the loads on the bearing of the blade-shaft in the radial direction. The axial load is about of 560000 [N] on the inner bearings mostly due to the centripetal force. Thank to the results obtained from the simulation with the multibody model, it was possible to calculate correctly the size of the bearings and evaluate the loads on the impeller to define the size of the structural parts.

## 6. CONCLUSION

This article presented the design process and the computation tools that are useful for the project of a variable pitch axial fan. Particular importance was given to the kinematic synthesis and the dynamic analysis of the command mechanism of the rotation of the blades. the implementation of a multi-body model allowed us to simulate the inertial effects, friction and aerodynamic loads to assess the most critical components and appropriately size them.

## REFERENCES


[1] Th Billotet and U. Johaenntgen. Bexbach ii power station - advanced electricity generation from bituminous coal. *VGB-Kraftwerkstechnik*, 75(1):21–25, 1995.

[2] Abroell G. Electricity from pit coal - bexbach power plant. *BBC-Nachrichten (Brown, Boveri and Cie)*, 66(4):141–148, 1984.

[3] Gurbin and Florian. Adjustable pitch fan. *Patent, Canada*, 1978.

[4] W.B. Kolb. Lubrication of fan blade bearings. *Patent, Germany*, 1974.

[5] L. Mulluer. Variable-pitch axial flow fans for thermal power station.

[6] A. Takata. Construction of adjustable blade shaft bearing in axial-flow fan with adjustable blades. *Patent, Japan*, 1979.