# Vector Estimation of Centrifugal Force and CAD Techniques for Protection Mechanism at Over-Speed of the Low-Power Wind Turbine Rotor

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***Abstract:*** *Rotor over-speed protection for small power wind turbine requires a possible cheap and reliable mechanical system. This is because any electric or electronic device depends on the electrical power source which by an electrical damage becomes unavailable. Between these protection devices centrifugal mechanism is priced at acceptable cost, but contains some subtleties of interpretation of the mode of operation. In this paper, it is explained step by step the original solution of this mechanism put in place for a 5 kW wind turbine. After triggering the centrifugal mechanism, the rotor blades are rotated to the flag position producing aerodynamic braking of the rotor. The speed is reduced to 20 ... 30 rpm, after which the action of the centrifugal mechanism is greatly reduced (centrifugal force decreases) so that the force in the compressed spring plus the aerodynamic torque of the blades relative to the axis of the blade spindle causes the return to the initial working position. Adjusting ballast weight of centrifugal mechanism and the reliable automatically operation of the whole system should be tested repetitively on an endurance stand.*

***Keywords:*** *Vector, centrifugal force, wind turbine, centrifugal mechanism, over-speed protection*

## 1. Introduction

The centrifugal over-speed mechanism acts on the rotor blades and is activated by centrifugal force that occurs due to the rotor rotation movement. It has an apparently complex structure because only one component of the centrifugal force has an active role in triggering it, so that in order to understand its operation and utility, the nominal operating data of the turbine are taken into account.

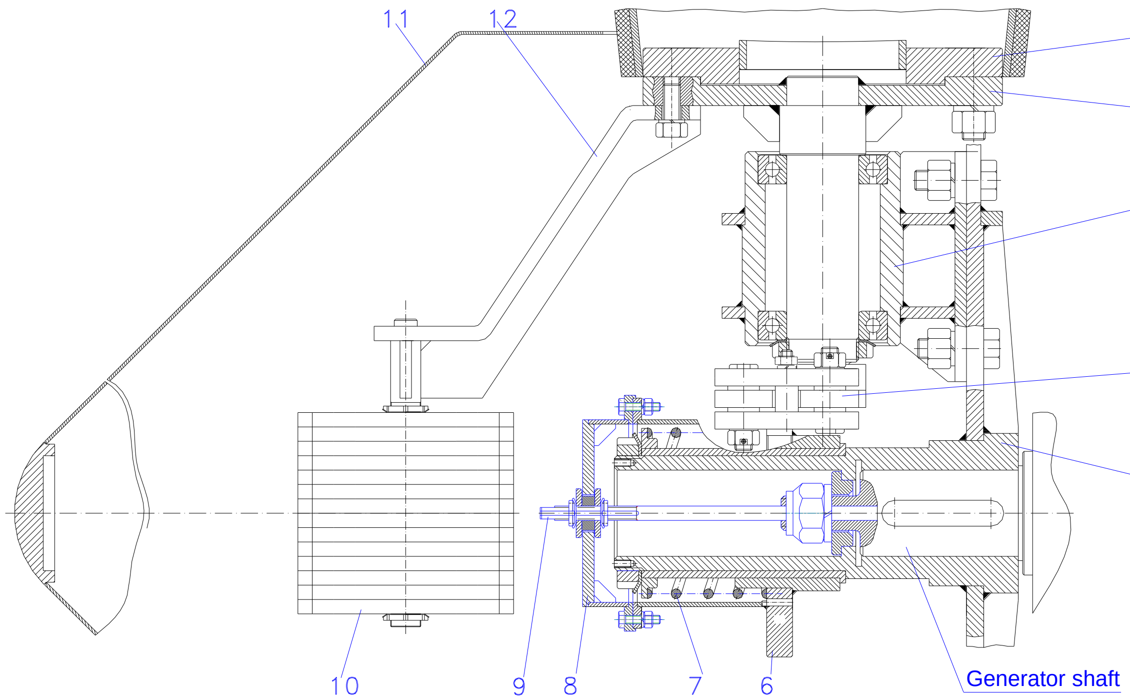
The nominal generator speed is 120 rpm, and can increase up to a maximum of 150 rpm. The turbine is coupled directly to the electric generator ("direct drive"), namely it does not contain a gearbox or speed amplifier. The rotor speed must be in the rotation domain of the electric generator.

The rotor blades can be rotated up to 45°. The pitch angle, $\beta_s$, must be maintained at the rated position and at speeds higher than the rated speed it should increase by about 45°, which means that the blades tend to be near to the flag position, in the wind direction and as a result aerodynamic braking occurs.

The problem to be solved is to automatically change the pitch angle when the rotor speed tends to exceed the limit of 120 rpm. There are two torques which acts on the blade rotating around its axis: an aerodynamic torque given by the force of the air in the direction perpendicular to the blade and a torque of a centrifugal force component of the counterweights. The centrifugal mechanism must contain an elastic element (spring) to ensure that the blades are brought back to the initial position as soon as the speed has fallen below the threshold limit. This pre-tensioned spring acts on the blade by means of a crank mechanism, producing a torque balance between the above mentioned torques.

In addition to the technical and economic aspects, it is also necessary to take into account the rotor safety, for which an automatic mechanical system is more reliable than an electric or electronic one. Occasionally, wind speeds over 30 m/s were recorded in the area.

Putting and maintaining in the wind direction is ensured by the tail of the aggregate.




1. Blade flange
2. Rotating support flange
3. Spindle blade support
4. Slider-crank mechanism
5. Disc rotor hub
6. Sliding part
7. Helical spring
8. Sliding rod support
9. Sliding rod
10. Counterweight washer
11. Protective ogive
12. Counterweight arm

**Fig. 1.** Centrifugal protection mechanism for over-speed

## 2. Vector estimation of centrifugal force

Politehnica University of Timisoara, in the framework of a multidisciplinary research program (mechanical, electrical, automation), has produced and put into operation a wind power unit of 5 kW. For protection against over-speed of the rotor, in the case of excessively high winds, a centrifugal mechanism was explored and designed, which is triggered automatically when the maximum permitted engine speed is exceeded, causing the blades to turn to the flag position followed by aerodynamic braking.
The centrifugal over-speed mechanism (Fig. 1) operates alternately in the sense that in a few seconds after triggering the rotor blades are placed in the flag position and aerodynamic braking occurs. The turbine speed decreases and the spring (7) of the mechanism returns the blades to the optimum working position, after which, if the wind persists at speeds above the permissible working limit, the rotor is operating again in load or if it is thrown away from the load, then even more, returns the rotor blades to the flag position.
The decisive part of the centrifugal mechanism is the counterweight. Attached to the rotating motion flange, it develops a centrifugal force whose tangential component with respect to the axis of the blade causes the blade to rotate towards the flag position. The geometric form obtained (Fig. 3) finally resulted from the following constraints:

- Position of the center of gravity in relation to the axis of the blade shall be at approximately maximum 300 mm, from gauge conditions;
- Contain correction elements (knobs) in order to adjust the spinning speed of the centrifugal mechanism by changing the rotating mass;
- Counterweight does not touch the rotor hub on the stroke domain, rotating between 0° and 45°.

The centrifugal force that occurs due to the rotation movement acts at the counterweight gravity center. If the entire subassembly of the counterweight is represented in 3D in the AutoCAD-Inventor design environment, then accessing the iProperties feature gives the coordinates of the center of gravity, mass, etc. The coordinates of the center of gravity are given in relation to the reference system chosen to construct the subassembly in 3D and are denoted by $X_g$, $Y_g$, $Z_g$. A Cartesian reference system is selected in which the rotation axis is reference system axis. According to the drawing, there are two perpendicular rotation axes located in the same vertical plane: the axis of rotation of the rotor, OX, and the axis of the blade rotation, OZ. The third axis, OY, will be perpendicular to the plane (XOZ) at point O and in the sense that the OX rotated clockwise to OY will give the OZ oriented vertically upward.

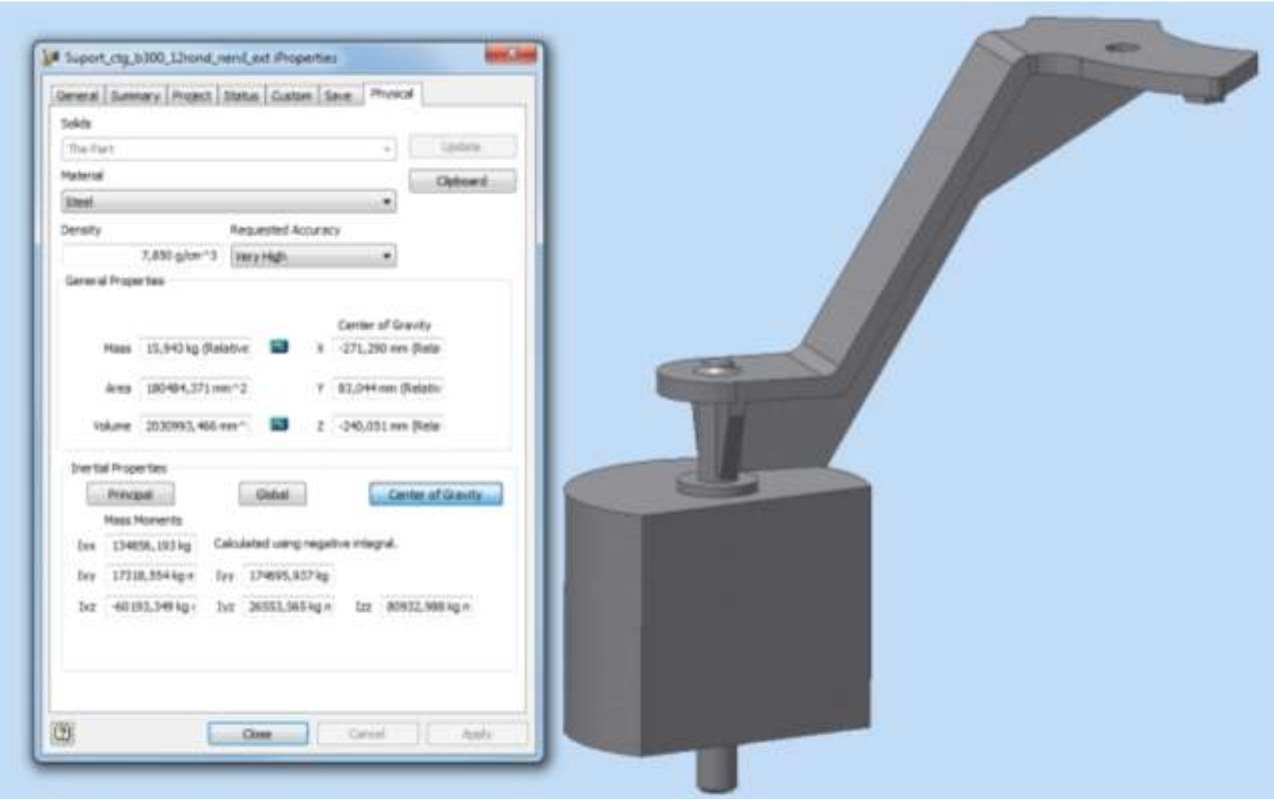



**Fig. 2.** The 3D representation of counterweight and the data plate displayed by the 3D design environment

As one can see in Fig. 2, the counterweight consists of an arm which is fixed on the blade flange and a rod to the other end, and on the rod a semi-cylindrical massive part is inserted. Because the entire system functionality is done by stand tests and in-situ, it is necessary that the weight of the counterweight to be adjustable. That's why the semi-cylindrical part was split into 10 mm thick washers.

The maximum number of washers initially provided was 14. Following the calculations and preliminary tests, an optimum of 12 washers was reached.

The projection of counterweight and center of gravity in the XOY plane is like in Fig. 3. The vector radius of the center of gravity, $\vec{r}$, projected in the plane (XOY) is denoted by $R_{XY}$.

$$R_{XY} = \sqrt{X_g^2 + Y_g^2} \tag{1}$$

The angle of this projection to the OX axis is denoted by α and is a calculation parameter for the centrifugal force and its torque. If taken as the reference plane for the angle α, the plane (XOZ), then the starting point α of the counterweight gravity center is $\alpha_{min}$ and will be calculated with the relation (2) (it is taken in absolute value because some coordinates can be negative).

$$\alpha_{min} = arctg\left(\frac{|Y_g|}{|X_g|}\right) \tag{2}$$

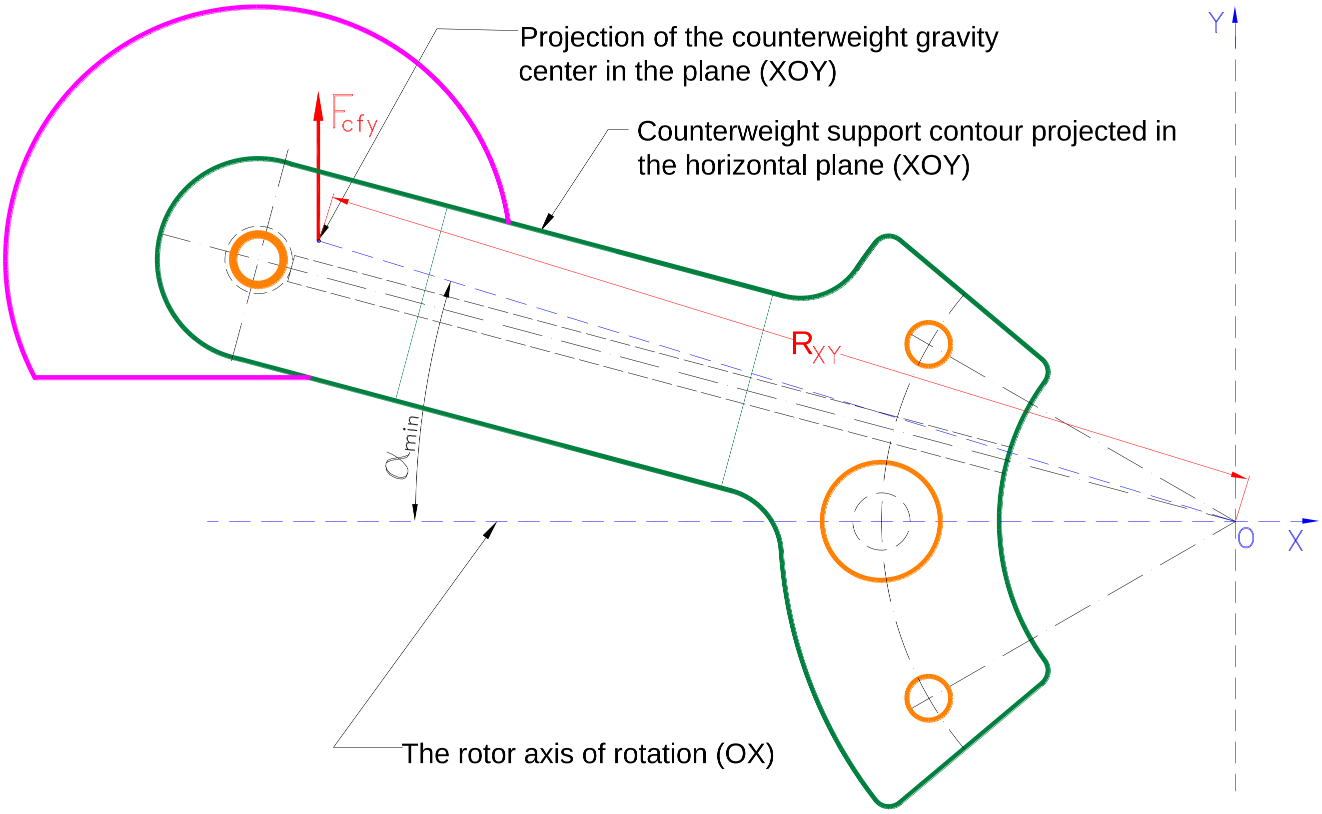



**Fig. 3.** Projection of counterweight in a plan (XOY)

According to theoretical mechanics and vector calculations the centrifugal force is determined with the relation (3):

$$\overrightarrow{F_{cf}} = m_{cg}[\vec{\omega} \times (\vec{\omega} \times \vec{r})] \quad (3)$$

Where $\mathbf{m_{cg}}$ is the mass of the counterweight, $\vec{\boldsymbol{\omega}}$ is the angular velocity of the rotor and $\vec{\boldsymbol{r}}$ is the radius of the generic gravity center. The unity axes vectors of the reference system are $\vec{\imath}, \vec{\jmath}, \vec{k}$. In this reference system $\vec{\omega}$ has a component only on the OX axis.

$$\vec{\omega} = \omega\vec{\imath} \quad (4)$$

The vector radius of the center of gravity is expressed with the relation:

$$\vec{r} = r_x\vec{\imath} + r_y\vec{\jmath} + r_z\vec{k} \quad (5)$$

Developing the double vector product (3), we obtain:

$$\vec{\omega} \times \vec{r} = \begin{bmatrix} \vec{\imath} & \vec{\jmath} & \vec{k} \\ \omega & 0 & 0 \\ r_x & r_y & r_z \end{bmatrix} = -\omega r_z\vec{\jmath} + \omega r_y\vec{k} \quad (6)$$

$$\vec{\omega} \times (\vec{\omega} \times \vec{r}) = \begin{bmatrix} \vec{\imath} & \vec{\jmath} & \vec{k} \\ \omega & 0 & 0 \\ 0 & -\omega r_z & \omega r_y \end{bmatrix} = -\omega^2 r_y\vec{\jmath} - \omega^2 r_z\vec{k} = -\omega^2\left(r_y\vec{\jmath} + r_z\vec{k}\right) \quad (7)$$

It is observed that the centrifugal force has components only on the OY and OZ axes as expected. Rotation of the blade has effect only on the OY axis component. This vector-expressed component has the expression:

$$\overrightarrow{F_{cfy}} = -m_{cg}\omega^2 r_y\vec{\jmath} \quad (8)$$

Scalar form is important for calculations. According to Fig. 3 and the notations above, the generic coordinate $\mathbf{r_y}$ is the projection $R_{XY}$ on the OY axis and thus:

$$F_{cfy} = m_{cg}\omega^2 R_{XY} sin(\alpha_{min} + \alpha) \tag{9}$$

The torque, $\overrightarrow{M_{cf}}$, made by the centrifugal force is determined, as vector, with the relation:

$$\overrightarrow{M_{cf}} = \overrightarrow{F_{cf}} \times \vec{r} = \begin{bmatrix} \vec{\imath} & \vec{\jmath} & \vec{k} \\ 0 & -m\omega^2 r_y & -m\omega^2 r_z \\ r_x & r_y & r_z \end{bmatrix} \tag{10}$$

Developing the matrix in (10) results:

$$\overrightarrow{M_{cf}} = -m\omega^2\left[\left(r_y r_z - r_y r_z\right)\vec{\imath} + r_x r_z \vec{\jmath} - r_x r_y \vec{k}\right] = -m\omega^2\left(r_x r_z \vec{\jmath} - r_x r_y \vec{k}\right) \tag{11}$$

For the intended system is important the component of the OZ axis, which causes the rotation of the blade:

$$\overrightarrow{M_{cfz}} = m\omega^2 r_x r_y \vec{k} \tag{12}$$

The scalar form of the active torque, $M_{cfz}$, using peculiar notations for each blade rotation, becomes:

$$\begin{aligned} M_{cfz} &= m_{cg}\omega^2 R_{XY} sin(\alpha_{min} + \alpha) R_{XY} cos(\alpha_{min} + \alpha) = \\ &= m_{cg}\omega^2 R_{XY}^2 sin(\alpha_{min} + \alpha) cos(\alpha_{min} + \alpha) \end{aligned} \tag{13}$$

In calculations and graphical representations is used the rotational speed, n [rpm], instead of the angular velocity, ω, where:

$$\omega(n) = \frac{\pi n}{30} \tag{14}$$

With the customizations introduced above the relationship (13) becomes:

$$\begin{aligned} M_{cfz}(n,\alpha) &= m_{cg}[\omega(n)]^2 R_{XY} sin(\alpha_{min} + \alpha) R_{XY} cos(\alpha_{min} + \alpha) = \\ &= m_{cg}[\omega(n)]^2 R_{XY}^2 sin(\alpha_{min} + \alpha) cos(\alpha_{min} + \alpha) \end{aligned} \tag{15}$$

## 3. Slider- crank mechanism

The counterweights attached to each blade cannot be left to act independently because the rotor rotates with the blade rotation axis into vertical plane, on each blade will act a variable weight force dependent from the rotor angle of rotation. That is why the three rotor blades with the counterweights on them must be connected together by a crank shaft mechanism with a sliding piece along the rotor axis of rotation. On each blade there is a slider-crank mechanism coupled with the sliding piece through a cylindrical joint. This mechanism is schematically represented in three significant positions (Fig. 4). The handle of the mechanism is rigidly connected to the rotor blade spindle at point O (Fig. 4). In turn, the sliding piece is driven by a pre-compressed helicoidally spring to return the blades to the working position after the rotor speed drops below the permissible limit, following aerodynamic braking. Through this mechanism the three blades are connected simultaneously and will rotate identically depending on the dynamic equilibrium of this mechanical system. The force in the pre-compressed helical spring acts on the rod through the Fb component. At point A of crank articulation, this component breaks down in two perpendicular directions (Fig. 5). The tangential component, $F_m$, determines the rotation torque on the blade spindle, heaving the force arm "$l_m$".

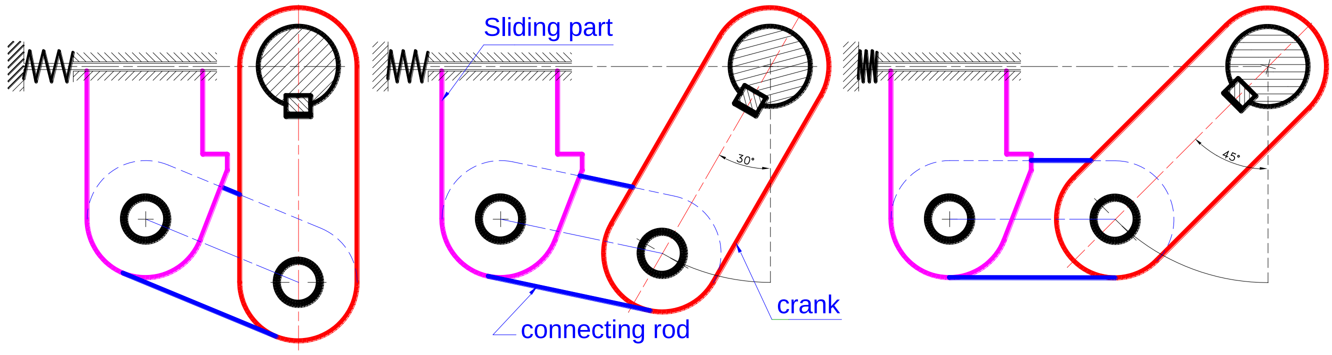



**Fig. 4.** Three successive positions of the slider-crank mechanism

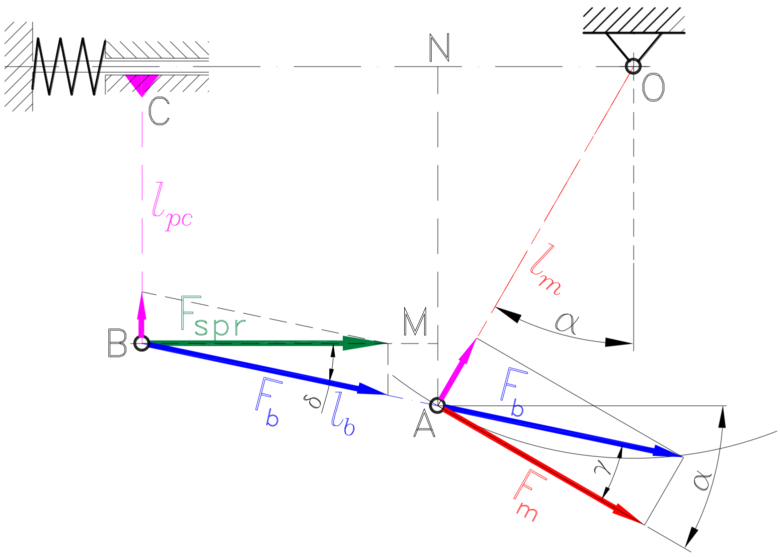



**Fig. 5.** The decomposition of forces in the slider-crank mechanism

The force given by the spring is evenly distributed over the three slider-crank mechanism (if there are 3 blades). This force acts directly on the sliding part, respectively in the joint B of the connecting rod (slider). On the connecting rod acts only one component of the force of the spring, respectively $F_b$. The spring force $F_{spr}$ is decomposed in two non-orthogonal directions, showed in Fig. 5. If note δ the angle between the force $F_b$ and $F_{spr}$ results:

$$F_b = \frac{F_{spr}}{3\cos(\delta)} \tag{16}$$

where:

$$\cos(\delta) = \frac{BM}{l_b} \tag{17}$$

The force $F_b$ acts on the crank in the joint A where it breaks into two orthogonal components. The component $F_m$ determines torque against the crank with arm "$l_m$" and observing the relationship between the angles $\gamma + \delta = \alpha$ results:

$$F_m = F_b\cos(\gamma) = F_b\cos(\alpha - \delta) \tag{18}$$

From the geometric configuration results BM and δ:

$$BM = \sqrt{l_b^2 - AM^2} = \sqrt{l_b^2 - \left(l_m cos(\alpha) - l_{pc}\right)^2} \tag{19}$$

$$\delta = arccos\left(\frac{BM}{l_b}\right) \tag{20}$$

Combining (16) and (18) results:

$$F_m = \frac{F_{spr}\cos(\alpha-\delta)}{3\cos(\delta)} \tag{21}$$

The torque of spring elastic force is determined with the scalar relation:

$$M_{spr}(\alpha) = F_m l_m = \frac{l_m F_{spr}\cos(\alpha-\delta)}{3\cos(\delta)} \tag{22}$$

### 4. The dynamics of the centrifugal protection mechanism at over-speed

The forces and torques to be calculated in function of two independent variables and the coordinates of the counterweight gravity center obtained from the 3D representation of the counterweight for the initial position of the whole assembly. These are:

- Turbine rotor speed n, for which a variation range of 50 ... 150 rpm
- The current angular position α of counterweight relative to the initial position, αmin and final (to flag), where $\alpha_{max} = \alpha_{min} + 45°$
- The first constant calculation is the mass of counterweight, $m_{cg}$
- The second calculation constant is the active crank length, $l_m$
- The following three constants are the coordinates of the counterweight gravity center: $X_g$, $Y_g$, $Z_g$

The forces and torques occurring in the operation of this automatic mechanical system are determined by the rotor speed, n, and the current angular position α of the counterweight from the initial position, $\alpha_{min}$.

Assuming that the turbine is optimally charged depending on the rotational speed and wind velocity, the torque of aerodynamic forces is calculated in relation to the axis of the blade shaft in 6 ... 10 working points. By interpolating these calculated values, the curve of the aerodynamic torque is obtained according to the rotational speed, (Fig. 6).

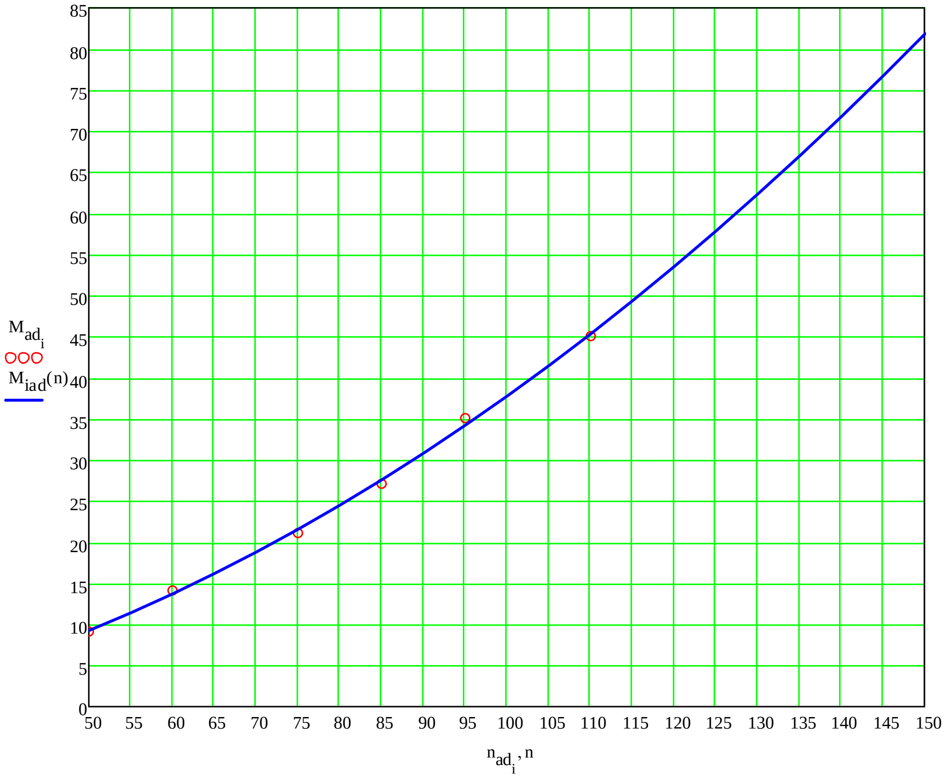



**Fig. 6.** The aerodynamic torque, $M_{ad}$, curve versus rotational speed, n

Dynamic balance with respect to the blade axis of rotation occurs between three torques: the torque of centrifugal force, $M_{cfz}$, the torque of aerodynamic forces relative to the axis of the blade spindle, Mad, and torque on the crank arm of the pre-compressed spring force, $M_{spr}$. Taking into account the sign of these torques, the relationship follows:

$$M_{spr}(\alpha) + M_{ad}(n) = M_{cfz}(\alpha, n) \tag{23}$$

By replacing the torques determined above, the relation (23) becomes:

$$\frac{l_m F_{spr}\cos(\alpha-\delta)}{3\cos(\delta)} + M_{ad}(n) = m_{cg}[\omega(n)]^2 R_{XY}^2 sin(\alpha_{min}+\alpha)cos(\alpha_{min}+\alpha) \tag{24}$$

The relationship (24) deduces the spring pre-compressed force which to be performed to maintain the balance within the optimum turbine speed range. When exceeding the maximum permissible rotational speed for the turbine, the dynamic balance is broken and counterweights turn the blades to the flag position followed by the aerodynamic braking of the rotor. The values obtained for $F_{spr}(\alpha, n)$ are very useful in spring dimensioning and enabling the mechanism to be triggered at the admissible limit speed.

$$F_{spr}(\alpha, n) = \frac{3\left(m_{cg}[\omega(n)]^2 R_{XY}^2 sin(\alpha_{min}+\alpha)cos(\alpha_{min}+\alpha) - M_{ad}(n)\right)\cos(\delta)}{l_m\cos(\alpha-\delta)} \tag{25}$$

## 5. Conclusions

All the components of this automated mechanical system can be calculated with sufficient accuracy, except for the aerodynamic torque. The friction forces were not taken into account because their size order is negligible relative to the other forces and torques which in turn are influenced by the tolerance fields of the component parts. The deduced relationships allow numerical simulation of this mechanism, modifying functional parameters, dimensions, etc. through which an optimal variant can be reached. As an automated mechanical system that needs to intervene promptly and safely, it is necessary to test it on a test bench in a repetitive manner. The approval of the system can be done only after at least one year of operation on a site with an aero-energy potential, so that all situations of climatic events can be traveled: high wind gusts, rain, snow, chill, extreme temperatures (negatives and positives), long periods of rest, etc.

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