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PROJECT OF A TANGENTIAL FLOW MACHINE OF SIMPLIFIED MANUFACTURING

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Abstract: *The growth of the Brazilian productive chain generates an increasing demand of energy to supply the needs of the industrial processes. This growth makes it necessary to invest more in research to optimize production processes, making them self-sustaining. It is in this sense, that the main objective of the present study is the realization of a project, its execution and analysis of a tangential flow machine of simplified manufacturing. In order to do so, the main theories of fluid mechanics and fundamentals of thermodynamics were used, as well as the main stages of a Mechanical Engineering project, aiming at the optimization of the system. Using the developed methodology, it was possible to achieve in all the tests carried out the main objective of this work, which was the generation of mechanical power in the 1000 W shaft. In this way a comparison was made with the initial design, presenting an efficiency of 18.63 %, which in this case was considered satisfactory.*

Keywords: *Flow machine. Cogeneration. Sustainability. Simplified manufacturing.*

1 INTRODUÇÃO

The high investment required to development the existent flow machines, the need to apply advanced technology and specialized labor, arises the necessity to design a tangential flow machine of simplified manufacture, which has simple process of working and can use the wasted energy of other process. Souza (2011) explains that the project of a flow machine, as in other projects, has the goal to attend the initially set expectation. In this case, the expectation is to transform energy of quantity of motion on a fluid into mechanical energy. To meet this need, it is necessary to have a good interaction between theories and given data of the project, providing a suitable scale plan that will permit executing the proposed project. This model is going to be strictly tested for validation purposes.

A flow machine has the main concept of energy transformation. Every machine flow, theoretically, is designed considering the principles of an ideal fluid that enable the use of the same methodology for calculation. The theory indicates that important fluid proprieties, such as specific volume and viscosity, are characteristics of the considered fluid and they influence the scope of each project (HENN, 2006).

The exergy enables a critical evaluation of the systems and a better use for energy flows of an energy converter machine, improving the ability to generate energy from the mechanical work of a system. Therefore, a robust method characterizes the real energy flow in a process. This theory identifies where occurs the bigger energy losses, informing the real factor to invest, looking to improve global efficiency in a process (ROJAS, 2007).

In this way, this article contemplates the development of a tangential flow machine, through the realization of the main phases of the project, which are understanding of requirements, basic calculations, 3D and 2D design, fabrication project, fabrication process and testing and validation. For that, were applied the principal concepts of thermodynamics and fluid mechanical, seeking for the higher efficiency of the system.

According to Souza (2011), the constructive elements of a flow machine are divided in two sections. One named fixed or carcass, and the other, rotating, called rotor or rotors. Some elements are directly related with hydraulic flow path and are made by those components: injectors, diffusers, fins and blades. The injectors are the components responsible for the increase in the fluid velocity on the flowing direction, so, injectors are components that convert pressure energy to kinetic energy. Diffusers are components with opposite characteristics of injectors. In diffusers, velocity decreases in the flowing directions. The fins are part of the components pre-distributions and reconductors, elements that have as main objective to direct the flow of the system, keeping the geometric characteristics of the carcass and, by that, making possible the transmission of efforts.

According to Fox (2013), flowing trajectory through the rotor is a determinant factor for choosing the application of a flow machine. Considering a tangential flow machines, fluid trajectory is tangential to the rotor and its main use is at Pelton turbines, where the rotor blades act as deflectors to the flowing fluid, generating tangential force in the rotor, providing power for the rotating element.

Klonowicz et al (2017) explain that choosing the fluid consider not just performance, but also depends on price, toxicity, explosiveness, availability, environment impact and other considerations. It was proved that is viable to project turbines for a small power that reach great efficiency. One of the most important tasks, which influences in efficiency and on economic aspects is the definition of the fluid, considering that greater efficiency is related to high fluid temperature. The First Law of Thermodynamics, according to Van Wylen et.al (2012), relate all types of energy present on flows. For most of engineering analysis, energy exists in different forms, which are potential, kinetic, pressure or heat. When steady state and isothermal flow is considered, the First Law of Thermodynamics is as Eq. (1).

$$\frac{p_1}{\rho} + \frac{v_1^2}{2} = \frac{p_2}{\rho} + \frac{v_2^2}{2} \tag{1}$$

For flows, which potential energy is negligible, such as heat transfer and work, the simplified version of the First Law of Thermodynamics is Eq. (2).

$$v_2 = \sqrt{\frac{(p_1 - p_2)2}{\rho}} \tag{2}$$

According to Çengel (2011), exergy is in equilibrium when the system doesn't have any condition to perform any type of work. It means that the relation between temperature and pressure with the surroundings cannot supply kinetic and potential energy to the system. The exergy balance is the relation between total energy provided to the system and the energy leaving the system, known as destroyed exergy. The result for this relation is the maximum energy of the system, which can be converted in work, and is obtained using Eq. (3).

$$\Delta\varphi = \left(h - T_0 s + \frac{v^2}{2} + gz\right) - (h_o - T_0 s + gz_0) \tag{3}$$

As reported by Çengel (2015), the principle of mass conservation is one of the most important principles in nature. Every physical and chemical systems interact with the energy of their surroundings, and the amount of energy involved in this system is equivalent to a small portion of mass, comparing to the global mass of the entire system. For closed systems application, this principle must be validated, and to do so, the mass of the system must be constant during the whole process. For control volumes, where mass surpass the boundaries of the system, is essential to monitor the quantity of mass that enter the system and the amount of mass leaving the system, to guarantee the values are equal, during a specific time. This equation is simplified as Eq. (4), considering steady state flow and uniform velocity profile on the flowing sections.

$$0 = \frac{\partial}{\partial t}\int_{vc} \rho d\forall + \int_{sc} \rho\vec{v}\, d\vec{A} \tag{4}$$

In consonance with Fox (2013), the principle of conservation of angular momentum explains that the sum of all external torques that act in the system must be equal to the torque done by linear momentum of the flow. Considering a steady state flow and uniform velocity profile in the flowing sections, this principle is described mathematically at Eq.

(5). In this equation, $\vec{T}$ representes external torque acting in the system $\vec{r}$ is position of the analized point until the flowing and e $\vec{V}\rho\vec{V}A$ repreents the quantity of linear momentum of the flow.

$$\vec{T} = \sum \vec{r} \times \vec{V}\rho\vec{V}A \tag{5}$$

For compressible flow, the specific mass of the fluid, according to Fox (2013), can be calculated through the Ideal Gas Law, as Eq. (6).

$$\rho = \frac{p}{RT} \tag{6}$$

Halliday (2002) presents that angular velocity of a rotation body informs the angle traveled per unit of time. This angle can be determined by Eq. (7).

$$\omega = \frac{2\pi}{60} n \tag{7}$$

Damirchi (2016) and Arashnia (2015) show that a promising technology is the Stirling engine, powered by hot smoke produce by a boiler. Those devices permit generation of 16% electrical energy even for a very small unity. Those are concepts based on idea of cogeneration of energy.

2 MATERIALS AND METHODS

In this section, will be approached the main stages of project, beginning on basic calculations. After, using a CAD software, was possible to design the 3D project, following by the fabrication process and validation of mechanical power of the machine shaft.

2.1 Basic sizing

After a careful analysis of the existents theories and study of economic viability of the project, machine tools and available raw material, it was determined, a low cost tangential flow machine with a rotor of 300 mm in diameter, 3500 rotations per minute on the shaft and generation of 1000 W of mechanical power in the shaft.

The theoretical torque and theoretical minimum force to generate 1000 W can be obtained by adjusting Eq. (1) and (2). In this case, considering just one entrance and one exit on the system, the theoretical minimum torque needed to generate 1000 W of power on the shaft can be calculated using Eq. (8), through the ratio between power, in (W) and angular velocity, in rad/s.

$$T = \frac{P}{\omega} \tag{8}$$

The determination of minimum force and torque to generate 1000 W of mechanical power is given at Eq. (9), by the ratio between torque (T), in Nm, and radius of the rotor (r) in meters.

$$F = \frac{T}{r} \tag{9}$$

For determination of the available force on the feeding air flow of the injector is necessary to know the specific mass of the fluid. For the Test 1 experiment, which a manometric pressure of 517.10 kPa (75 PSI) was used, the specific mass was calculated using absolute pressure, it means, the sum of manometric pressure and atmospheric pressure.

Using Eq. (10), the calculation of specific mass (ρ), in (kg/m³), was done for each pressure used to validate the project. For each situation there was a different pressure, hence, that will be a specific mass for each test condition, considering that the determination of specific mass is the ratio between absolute pressure with the product of the ideal gas constant (R) and ambient temperature (T). For this condition, was considered an ambient temperature of 20 °C. The results of the other tests done using the same calculations will be presented at Tab. 1.

$$\rho = \frac{p\,(\mathrm{kPa}) + 101.32\ \mathrm{kPa}}{RT} \tag{10}$$

Besides the specific mass, it is also necessary to know the flowing velocity on the injector. The calculation to determine the velocity used in Test 1 was done using Eq. (2).

With the values of velocity and specific mass, was possible to calculate available force on the flow, for this case using Eq. (11).

$$F = v^2 \rho \frac{\pi D_{injector}^2}{4} \tag{11}$$

To determine the thermodynamic efficiency of the machine, the exergy balance on the system must be analyzed. For it, the same equation was applied in the entrance and exit of the turbine, fitting it for the ideal conditions to this study, the referred equation can be rewritten as Eq. (12).

$$\Delta\varphi = \frac{v^2}{2}\dot{m} \tag{12}$$

To make the testing and verification of this prototype, it was pre-determined the realization of three tests, using three different pressures. Every other condition of tests done were made using the same sequence of math presented previously, in which the results can be find at Tab. 1

Table 1. Results of the calculations for three different tests.

Test	**Manometric pressure (kPa)**	**Density (ρ)** $\frac{kg}{m^3}$	**Speed** $\frac{m}{s}$	**Force (N)**	**Ideal torque (Nm)**	**Exergy variation Δφ (W)**
1	517.10	7.353	375.04	29.24	4.38	5383.19
2	861.86	11.45	387.96	48.73	7.31	9453.69
3	1206.58	15.55	393.93	68.22	10.23	13438.67

Through the parameters presented at Tab. 1, theoretically, the specific objectives presented for this study have been achieved. Expressive variations of the system exergy have been seen, that is, the maximum exergy available on the system can be converted into power.

2.2 3D and 2D Project

After the basic sizing stage, all the constructive elements were design using a Computer Aided Design (CAD) software, as shows Fig. 1. One of the input data used for basic sizing was the rotor diameter of 300 mm, characteristic used to determine the number of blades on the rotor, considering also the angle of the milling cutter (60°) used to machine the rotor, the height limit of the blade, fixed in 10 mm until hitting a blade edge of 0.5 mm. Hence, there are 56 blades in the rotor, according to Fig. 1b. The Fig. 1a presents the main constructive elements of the system: 1 – structure; 2 – alternator; 3 – template for entrance angle; 4 – Rotor; 5 – carcass; 6 – bearings.

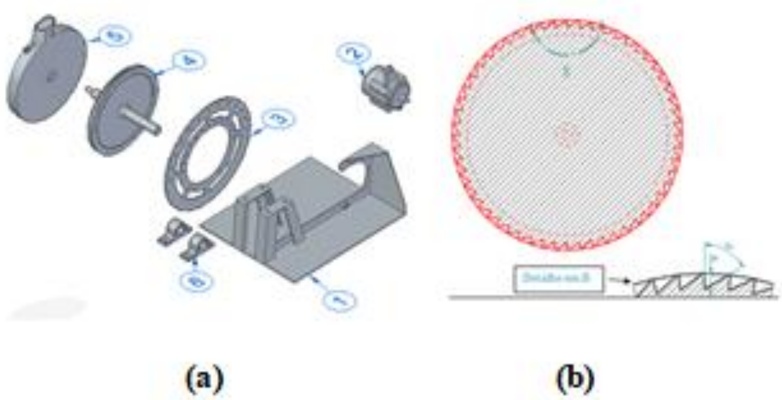

Figure 1. a) Constructive elements of the testing bench; b) detailing of the rotor blade.

2.4 Testing and validation

In the testing and validation stage, were developed templates to analyze all the variables involving efficiency and fluid dynamics behaviors of the system. Therefore, it was possible to make tests to determine the best work conditions for each element of the project, correlating in out data with the variable response obtained.

The test that has been done had as main goal to determine the best conditions to the following items: the gap between rotor and carcass; carcass entrance angle; validation of the injector diameter; carcass exit angle and system power validation. To make the experiments, was used, as power supply, a compressor of the brand Pressure Onix Press, which has a theoretical shift (flow rate) of 1133 1/min and maximum work pressures of 1206.58 kPa (175 PSI). This information were requisite to make the basic sizing and testing.

2.4.1 Determination of gap between carcass and rotor

For better understanding of a tangential flow turbine, first was developed a project according the financial resources available and structure at hand. Based on basic sizing, as shows Fig. 2a, a first carcass was developed to understand the course of the fluid for the proposed geometry. This stage was necessary because it was not possible to use the simulation software first available to flowing simulation, considering that it is made just for laminal flow, and the flow of this project is turbulent.

These devices also have as main goal the realization of experimental testing, considering that the bibliography consulted didn't explain, in a clear way, practical problems and interactions between constructive elements, besides many variables of flow machine projects that influence on a significant way on the dynamic effects of the system. Therefore, it was developed a first bench to simulate dynamic tests and better understanding of the flow course between carcass and rotor. It was adapted at the physical structure an alternator of 12 V and 50 A, seeking to load in the shaft, making possible to test on a free shaft mode (without alternator load) and loaded shaft (with alternator load). On the initial test, it was found that even on the best conditions, at the higher rotation levels, when the alternator was turned on, the shaft lost power, returning to a static condition a few seconds after the load application. It was decided to make a destructive test, increasing the gap between carcass and rotor, as presented in Fig. 2b. The tests were repeated, and presented a great improvement. The gap between carcass and rotor is a variable that has not been seen at the consulted bibliography, and its understanding was very important to the study sequence. After, was made test with the second compressed air entrance (Injector 2) on the carcass, working simultaneously, expecting to help on the system fed and analyze its influence on the system effectiveness. At Tab. 2, it is possible to understand the tests behavior in this experiment.

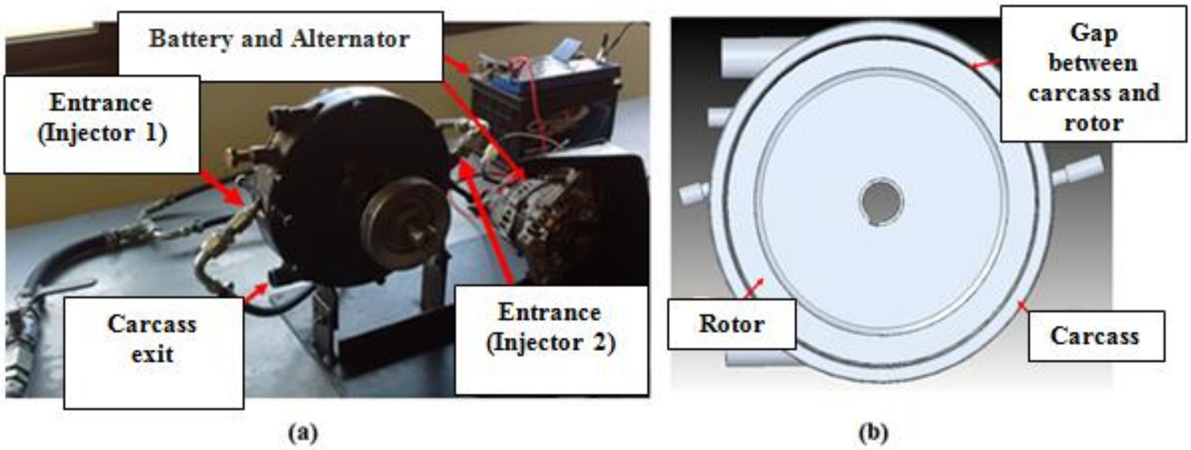

Figure 2. a) testing bench b) Gap between carcass and rotor.

Table 2. Tests to analyze the influence of the gap between rotor and carcass.

Test	Entrance n°	Pressure range in PSI 50 PSI – 334.7 kPa 100 PSI – 689.4 kPa 175 PSI – 1206.5 kPa	Gap rotor radius/carcass (in millimeters)	RPM free shaft	RPM load shaft
1	1	100 – 50	0,2	3000	0
2	2	100 – 50	0,2	4000	0
3	1	175 - 50	0,2	3500	0
4	1	100 – 50	0,5	3900	0
5	2	100 – 50	0,5	4500	0
6	1	100 – 50	1,5	5000	750
7	2	100 – 50	1,5	5500	1250
8	1	175 – 50	1,5	7000	3325

According to Table 2, is possible to understand that the gap between rotor and carcass influences in a significant form to the system efficiency. It is noticed that on tests 1,2,3,4 and 5, regardless of the used pressure, the haft didn't have any efficiency, stopping quickly after being loaded, and on tests 6,7 and 8, there is a good improvement, due a bigger gap. So, the tests have been made without a carcass, evidencing the percent operation of the system, proving that the carcass cannot influence on the system flow.

With the tests, it is possible to conclude that, for the presented settings, as lower the gap between carcass and rotor, higher the possibility of the feeding pressure jet act as positive and negative force in relation to the rotor blade, this way, stopping the rotor. The Fig. 3a presents, in a detailed form, the action of two vectors, in which the vector 1 acts as the injector feeding jet, being a positive force, and vector 2 a negative force. This phenomenon occurs when the fluid is in a confinement state, flowing just in the tiny gap between carcass and rotor, impairing the efficiency of the system. The Fig. 3b was made to determine the best work angle for the injector, using a template, that helped to get the best exploitation of the fluid energy. The position of the disc installed aside the rotor was determined by the physical space available, making possible the realization of the tests in 13 defined positions.

It is evident that, for the conditions used in this study, the carcass cannot influence on the fluid flow, it is, by the lower density of the compressed air, after the pressure jet act in the rotor blade, it will always have the trend to leave quickly the carcass. This way, the exit angle of the carcass must help the fluid to leave, avoiding the confinement of the fluid. So, the ideal condition for this model is an open system, which has been applied on the tests and presented on this study.

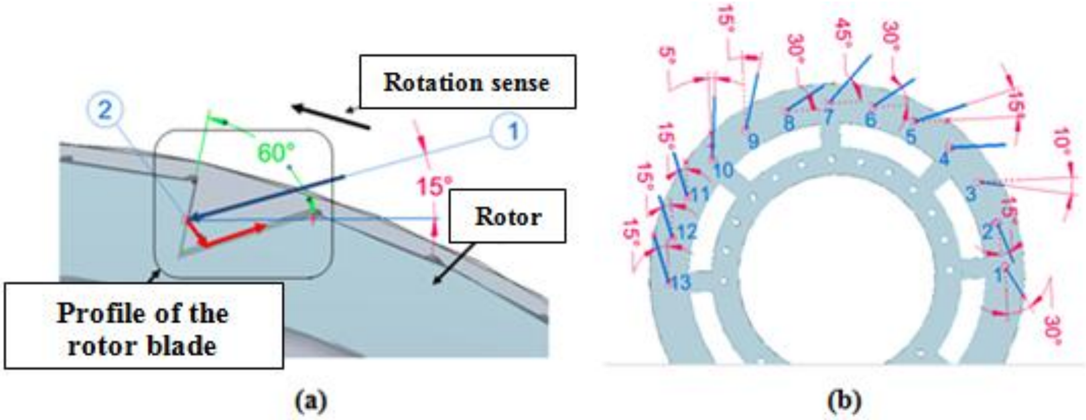

Figure 3. a) Positive Vector 1 and Negative Vector 2 acting on the rotor blade; b) Number of positions to test the better condition of entrance angle.

2.4.2 Determination of the entrance and exit angles of the carcass

It was installed a disc aside the rotor with holes at each 15°, as presented at Fig 4a. For each position of the disc, was made tests using one or two angles, always positioning the injector entrance in a tangential form in relation to the rotor. The device that fixed the injector was designed with a template, also with axial displacement, enabling a bigger approximation of the injector to the rotor.

After setting the entrance angle, was possible to establish an exit angle. To do so, were considered the best conditions obtained on the tests. Helped by a support, some wires were tied, positioning the support in the path of the fluid flow, making possible a better looking at the direction the fluid takes after hitting the rotor blade. This test helps to determine the exit angle of the carcass. The test done is presented at Fig. 4b.

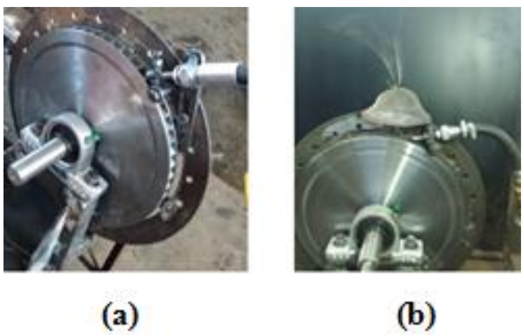

Figure 4. (a) Determination of the entrance; (b) Exit angles of the carcass.

In this drilling template, begging on position 1 presented at Fig. 3b, for every position was installed the injector, always seeking to tangential the rotor. In this form, was possible to analyze the best condition to the entrance angle of the carcass. The obtained results can be analyzed at Tab. 3.

Table 3. Comparative between input angle versus rotation on the rotor shaft, considering rotor diameter of 300 millimeters and diameter of the injector of 6 millimeters.

Position	**Angle**	**RPM**
1	30º	3300
2	15º	4280
3	10º	4400
4	90º	3919
5	**15º**	**4889**
6	30º	4549
7	45º	4474
8	30º	4850
9	15º	4815
10	5º	4800
11	15º	4513
12	15º	4717
13	15º	4404

Based on the values of Table 3, the best condition is using an angle of 15° in position 5. So, was determined position 5 as work position for the injector on the system validation test. This study has not been found on the literature used.

2.4.3 Validation of the injector diameter

To validate the best internal diameter for the injector, which is directly linked to flow rate and kinetic velocity of the flow and is an indispensable propriety for a better exploitation of the available energy on the flow, were used terminals of commercial hydraulic systems, that have standard diameter and easy to find on the market-place. The selected internal diameters selected were 4, 6 and 9 millimeter. In the pre-determined position of the input angle and helped by the join template, shown at Figure 4a, was possible to adapt three different terminals of internal different diameters.

As presented previously, the compressor used for the test don't have technology to keep the pressure constant, therefore, the same pressure variation was used for every test. The Tab. 4 shows the tests made.

Table 4. Test to validate the internal diameter of the injector

Test	**Pressure variation kPa**	**Internal Diameter of the injector (mm)**	**maximum rotation on the shaft**	**Time**
1	689.47 – 334.73	4	3000	04:52 min
2	689.47 – 334.73	6	6400	48s
3	689.47 – 334.73	9	4090	21s

According to the data at Tab. 4, for each diameter, when the manometer on the injector reached 689.47 kPa, the system was started with free load shaft. Then, was monitored the time and rotation until the manometer reached 334.73 kPa. Using these information, the conclusion obtained was that diameter of Test 2 presented the best condition of exploitation of the fluid energy. Although the time is lower, which means higher velocity of the flow, the test 2 present the best conditions for power generation on the shaft due the high rotation. This setting was selected and used on the following tests.

2.4.4 Validation of the mechanical power on the shaft

For the validation of the mechanical power of the low cost tangential flow machine, was used an electromagnetic brake Delorenzo DL 30300 and a digital device to measure the mechanical energy Delorenzo DL 10055N. This equipment is used on industries to measure torque on engine shafts, using a load cell, rotation velocity and mechanical energy measurements provided to the system. A test bench was mounted to coupling the magnetic brake in the tangential flow machine. A coupler was used to connect the machine shaft to the electromagnetic brake. The Fig. 5a shows the test bench. The digital device to measure mechanical energy has a little screen that shows the measure of the main information relevant to validate this project, as can be seen at Fig. 5b.

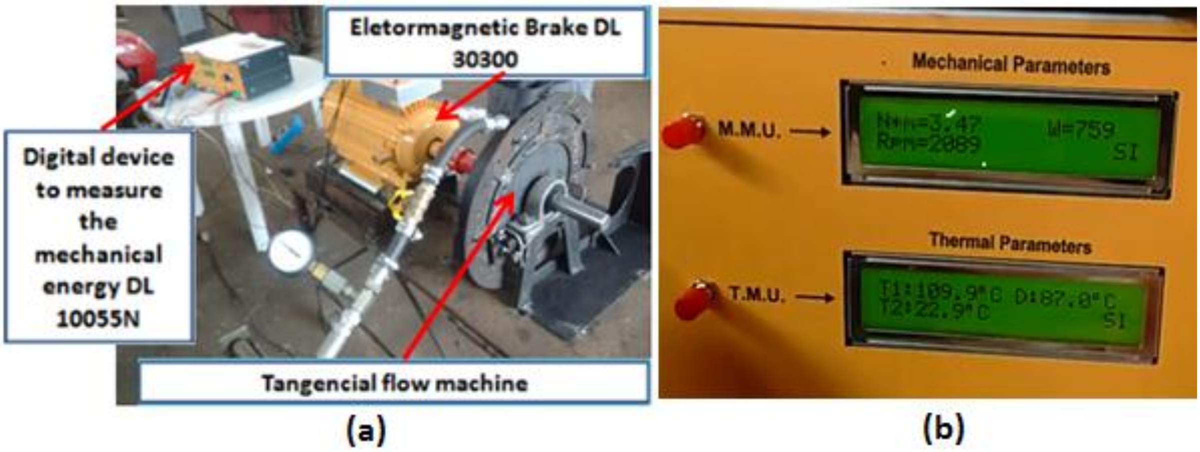

Figure 5. a) Test bench to validate mechanical power; b) Screen showing the mechanical parameters.

3 RESULTS E DISCUSSIONS

According to project requires and basic sizing that have been done, each test was made using 3500 rotations per minute. The quantity of three tests were made using just a coupled shaft on the electromagnetic brake, as shown at Figure 6a. Also, was used three pressure ranges for every test; Test 1 (517.10 kPa (75 PSI)); Test 2 (861.64 kPa (125 PSI)); Test 3 (1206.58 kPa (175 PSI)).

The tests were done according to the following steps: 1. Starting of the injector; 2. Monitoring on the screen until reaching 3500 RPM; 3. Data collecting; 4. Stop of the injector. Whit it, was possible to make an analysis of the interactions of pressure and RPM; Pressure and Mechanical Power; and Pressure versus Torque on the shaft, that are presented at Fig. 6a, 6b and 6c.

The Figure 6a presents the three tests done establishing the relation between pressure variation and shaft rotation. The graph behavior shows that as higher the pressure, higher the shaft rotation.

Also, according to the values in the Fig. 6b, the graph shows that on the three tests made, the power increase is proportional to pressure increase. This characteristic is result of the fact that as higher the pressure of the feeding air of a tangential turbine, higher is the available energy to be converted in power. With it, the initial goal of the study was achieved, since the project was validated in all the situations, as it reaches values higher than 1000 W of mechanical power on the shaft, even at the lower pressure test (Test 1). The Fig. 7c presents a graph with the values of the shaft torque according to the applied pressure. The shaft rotation for all the cases was standard, fixed at 3500 RPM.

The efficiency for the flow machine can be obtained relating the measured torque through the electromagnetic brake with the ideal torque, calculated on the basic sizing. The values of the ideal torque generated on the shaft, in a no-loss situation, were determined multiplying tangential force and the radius of the rotor ($\mathrm{T} = \mathrm{Fr}_{\mathrm{rotor}}$). For each test, the values of this multiplication are at Tab. 1. The measured torque are values collected during the tests, presented at Fig. 6c.

$$\text{Efficiency Test 1} = \frac{\text{Mensured Torque}}{\text{Ideal Torque}} = \frac{3.56\ \text{Nm}}{4.38\ \text{Nm}}\ \text{x}\ 100 = 81.11\ \%$$

The same math was used for the measured torque of 3.96 Nm and 3.97 Nm, resulting in an efficiency of 54.17 % and 38.79 %, respectively. According to the efficiency values based on the First Law of Thermodynamics, it can be observed, even knowing that are energy losses on the system, that Test 1 presents the better work condition.

The efficiency, based on the exergy balance, relate the available energy on the system, which can be converted to power, with the real energy converted by the system, measured through experiments. To this study, the aspect observed is the mechanical power on the shaft. The available energy is the kinetic energy of the flow, so, for this flow machine, the efficiency based on the exergy balance can be calculated by the ratio between measured power and the exergy variation rate, calculated previously on the basic sizing stage, presented at Tab. 1. The efficiency values for each test defined by the following calculations:

$$\text{Efficiency Test 1} = \frac{\text{Mensured Power}}{\Delta\varphi} = \frac{1022\ \text{W}}{4323\ \text{W}}\ \text{x}\ 100 = 18.63\ \%$$

The same math was used for measured power of 1218 W and 1351 W, resulting in efficiencies of 12.88 % and 10.01 %, respectively. Based on Fig. 6d, which has a graph of comparison between the efficiency based on the First Law of Thermodynamics and the exergy balance, can be seen that the best conditions to this prototype is also the Test 1, that has the lowest pressure, but reaches the initial goal of the present study, that was the generation of a mechanical power of 1000 W on the shaft by the better use of the power offered by the system.

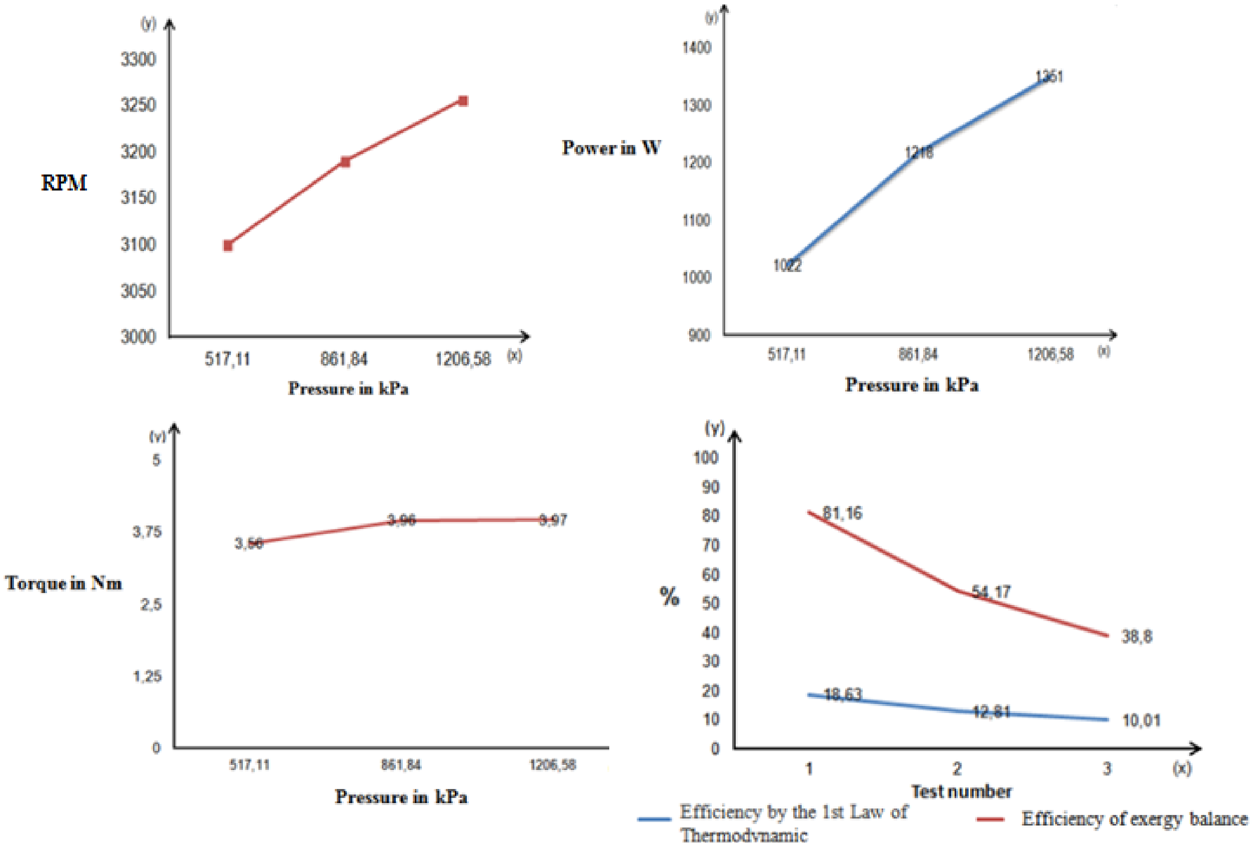

Figure 6. (a) Shaft rotation behavior according to pressure variation; (b) Mechanical power in the shaft (W), according to the pressure variation; (c) Behavior of torque on the shaft according to pressure variation; (d) Comparison of efficiency based on the First Law of Thermodynamics with exergy balance efficiency.

Comparing the obtained results with the studies of Kliniwicz (2017), Damirchi (2016) and Arashnia (2015), it can be seen the necessity of more research to evaluate the best fluid and blade angle of the rotor, because, according to the books, these are factors that contribute to increasing on efficiency of the system. A bigger investment on research for the related items can increase the efficiency of the system. Anyways, the reached results are considered satisfactory, considering the simplicity of the system and the low cost of the project.

4 CONCLUSION

This study presented the main stages of projects for a low cost tangential flow machine, since the conceptual stage, basic sizing, manufacture process and test bench development to validate the project. I was possible to analyze and understand the best conditions of the main constructive elements for a tangential flow machine, which were validated through practical experiments, using templates and developed equipment, that were very important for optimization of the proposed model. Future work exists for this project, with more investment on research to study aspects to increase the efficiency of the system, since the definition of fluid and rotor blade influence significantly the efficiency of flow machines.

Based on the data that was obtained and the practical tests made, can be concluded that the project reached expressive results, low investment considered. All the goals determined in the begging were reached, which the main goal was to generate 1000 W of mechanical power om the machine shaft. According to the tests that have been done, was noticed that the turbine generates 1022 W when air pressure getting into the system is at 517.11 kPa. In the related condition, the turbine presents an efficiency of 18.63 %. In the moment pressure changes to 861.84 kPa, the turbine generates 1218 W, with efficiency of 12.88%. For the higher air pressure, 1206.58 kPa, the generate power is 1351 W and efficiency drops to 10.05 %. These results were possible to get only using the presented methodology, which is proper to attend the goals proposed on this study, besides making possible to understand the stages of a mechanical engineering project.

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