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DESIGN AND MANUFACTURE OF ATTRITOR LABORATORY MILL FOR LOW COST

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Abstract. *The objective of this work was the development of a Programmable Attritor Mill to be used in the Powder Metallurgy Laboratory of the Federal University of Itajuba. This project should meet some technical requirements such as: grinding volume 30g, speed of rotation adjustable between 150 to 600 rpm, number of adjustable cycles maximum 100 cycles, and cooling using water, and liquid nitrogen. The design was performed using the Tresca criteria and some critical components in the grinding process and was validated through simulation of finite elements. The project proved to be technically and financially viable.*

Keywords: *Attritor mill, grinding speed, milling container cooling.*

1. INTRODUCTION

The powder metallurgy process (MP) is used in the production of components with complex geometry, because the processing cost is lower comparing to forging processes in large scale production, MP can be used to form multiple compositions by mixing the primary or pre-attached powders (Wong-Ángel et al., 2014).

To obtain metallic powders, it requires precise control of chemical and physical compositions, also to meet requirements inherent to application and cost. It is usually produced by chemical and mechanical methods. Commonly, these techniques include: Chemical processes (reduction of oxides), physical processes (water or gas atomization), processes by electrolytic decomposition, and mechanical processes (reduction in solid state), for the production of

metallic powders, mechanical comminution is the most applied due to the ductility of the material (ASM, 1998). Usually using grinding process.

Milling is a technique used to reduce certain particles to smaller particles by mechanical shocks between the milling element and the powder.

One of the equipment used in the milling process is the Attritor mill.

1.1 Moinho Attritor

Attritor mills are the ones that have the highest capacity among all the available mills. The material capacity of the grinding container ranges from 0.5 to 40 kg. According to Fig. 1, its operation consists in introducing the material and the grinding balls into the grinding container with the desired atmosphere, where rotary impellers move these components at a desired speed, which can reach up to 250 rpm. Its main advantages are the high capacity of the material to be ground and the speed of milling, being 10 times superior to the conventional mill (Suryanarayana, 2001).

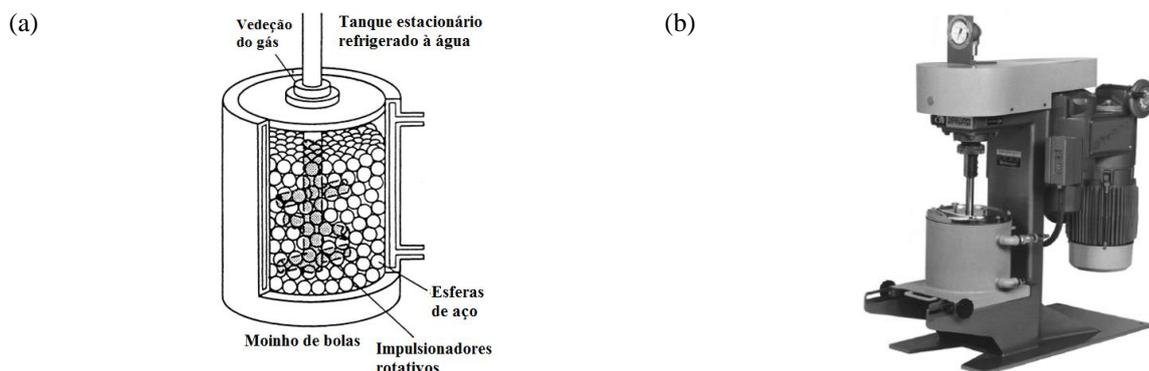


Figure 1.(a) Stationary tank, (b) Attritor Mill.

The project was requested to meet the technical requirements of Attritor mill according to Tab. 1 to be used in the Powder Metallurgy Laboratory of the Federal University of Itajuba. In the market there are several manufacturers of Attritor mill but with high cost and milling containers with much greater capacity than the need of this institution.

This project was financially feasible due to the use of national equipment and will use labor available at the university.

Table 1. Technical input requirements for the Attritor mill

Technical input requirements	Requested
Milling volume (g)	30
Rotations speed range (rpm)	150 to 600
Cooling	Water and liquid nitrogen
Number of cycles	60
Grinding in controlled atmosphere	Yes
Milling container material	Austenitic stainless steel
Impellers material	Austenitic stainless steel

2. DIMENSIONING OF EFFORTS

The determination of the efforts is paramount for the dimensioning of the components directly involved in the milling process.

2.1 Shaft Loading

The shafts will be dimensioned to resist the maximum moment that will be provided by the gear motor and the maximum flexion that will be imposed by the belt tension. This avoids any possibility of overload failure.

The tension in the belt is determined according to Shigley, Budynas, Nisbett (2008).

2.2 Static analysis

For the static analysis, the Tresca criteria will be applied taking into account the shock and fatigue factors applied to the torsion moment acting on it.

In this method we have the maximum shear stress in the components (Shigley, Mischke, Budynas, 2008).

K_m and K_s are factors related to shock. Also we can solve by the equivalent stress method for the value of the diameter incorporating the safety coefficient.

2.3 Fatigue analysis

2.3.1 Factors influencing fatigue

The properties of the materials are determined by laboratory tests under controlled conditions and environments. It is expected that the resistances presented by a mechanical component have to be corrected by factors that influence the resistance limit of the material in the conditions of manufacture and use in which it will be inserted. Factors that influence such resistance limit involve characteristics related to material, manufacturing, environment and design conditions (Shigley, Mischke, Budynas, 2008).

The criterion used for fatigue analysis was the Soderberg criteria.

3. ATTRITOR MILL PROJECT

3.1 Project input requirements

The developed project met all the technical input requirements and the number of cycles exceeded the need due to the flexibility of the equipment to be programmable according to Tab. 2.

Table 2. Requirements requested and attended technical to the Attritor mill

Technical input requirements	Requested	Attended
Milling volume (g)	30	30
Rotations speed range (rpm)	150 to 600	150 to 600
Cooling	Water and liquid nitrogen	Water and liquid nitrogen
Number of cycles	60	100
Grinding in controlled atmosphere	Yes	Yes
Milling container material	Austenitic stainless steel	Austenitic stainless steel
Impellers material	Austenitic stainless steel	Austenitic stainless steel

3.2 Attritor mill design

The model was developed with the software PTC CREO Parametric 3.0 according to Fig. 2.

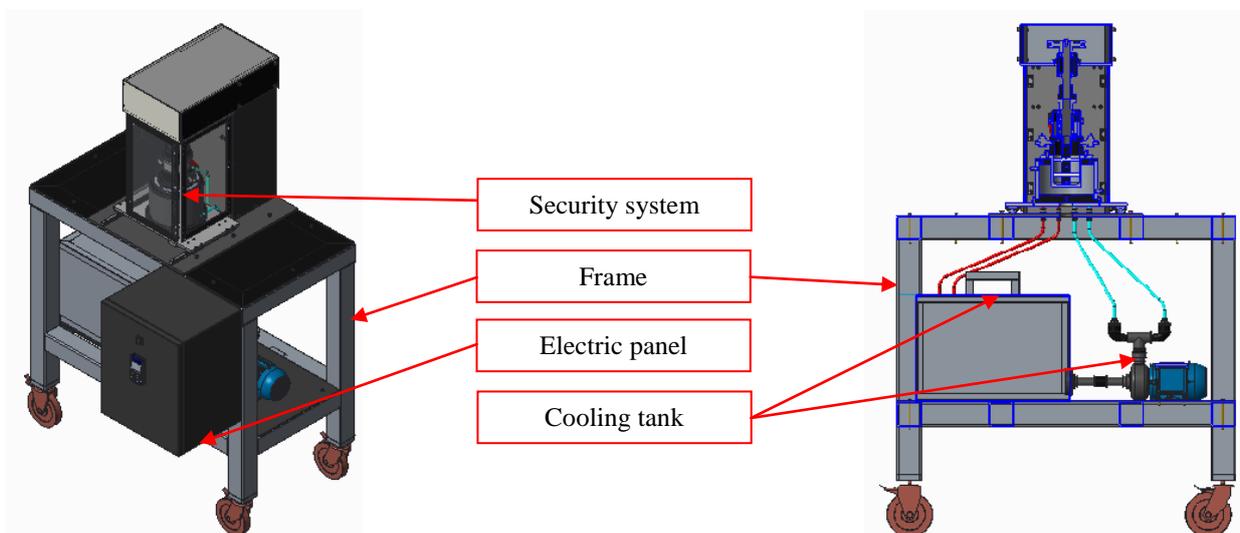


Figure 2. CAD drawing of Attritor mill

The mill will include a frame made of welded steel tubes and steel sheets. There will also be a milling container, cooling tank, grinding axis and transmission system, gear motor, electrical panel and cooling system. Its project was carried out following the considerations of NR12 safety standards. The frame is fixed in castors in order to increase the mobility of the equipment.

An electrical panel shall include mill control systems such as frequency inverter and its interface and components related to its electrical supply

The 3D model of the mill was carried out in conjunction with the dimensioning of the mechanical components.

Dynamic calculations of the motor and rotating components will be performed in this work to verify the safety factor (adopted $N = 2$) of the components directly involved in the grinding as well as the minimum diameters for each section of the shafts in order to optimize the geometry.

3.2.1 Milling container, cooling tank, agitator system and refrigeration system

Considering the initial proposition of the mill, the use on the study of the effect of high energy grinding, the set although interchangeable, it will be initially made in stainless steel AISI 316.

Inner milling container wear due to the friction and impact of the grinding balls can lead to contamination of the material to be milled. Stainless steel is commonly applied as a composite material for the milling container because it exhibits resistance to wear and, in this case, will have little influence on the contamination of the material to be milled. Both containers will be in contact with refrigerant liquid during the milling, the fluid used will be mainly water, so the material of its composition must have some resistance to oxidation.

The milling container of grinding will have a maximum capacity of 490 cm³. For the atmosphere control, connections for gas insulation were incorporated into the lower bearing of the grinding shaft.

The agitator system consists of a main shaft, intermediate shaft, drive shaft impeller including 4 impellers positioned transversely to the milling axis. The components cited in this topic are shown in Fig. 3 (a).

The electric motor is positioned at the rear of the mill with its shaft in the upright position. This will be coupled to the pulley drive system and "V" belt with 1: 1 ratio as per Fig. 3 (b).

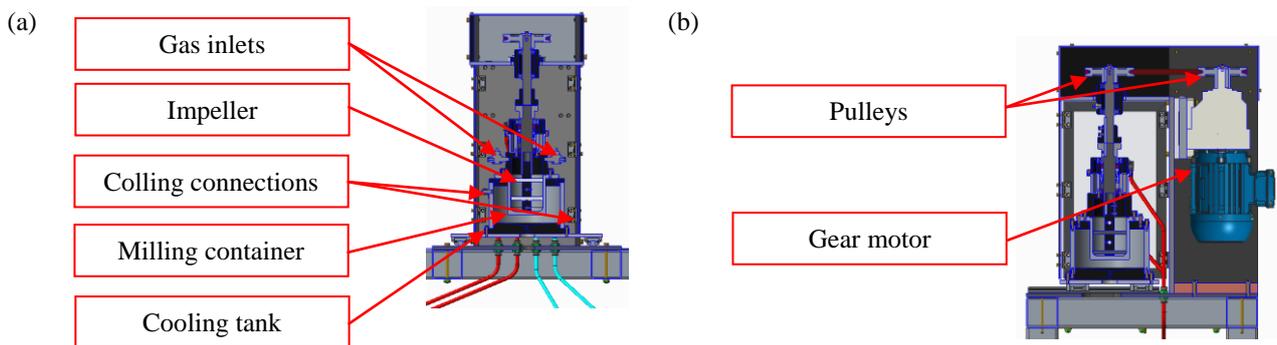


Figure 3. (a) Section view of the milling container, cooling tank and impeller system, (b) Section view of the transmission system

3.3 Dimensioning of the grinding system components

3.3.1 Breakdown of the grinding system into components

The grinding system is composed of 4 parts, and it will be analyzed separately. The parts will be divided according to Fig. 4.

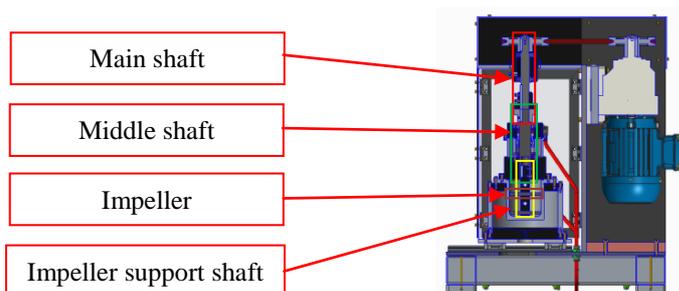


Figure 4. Section view of the transmission system

Where each of the parts can be described as follows:

- Main shaft;
- Middle shaft;
- Impeller support shaft;
- Impeller;

Due to the discontinued geometry of each of the shafts, each of these parts will be sectioned, and each section will receive a letter for marking as shown in Fig. 5.

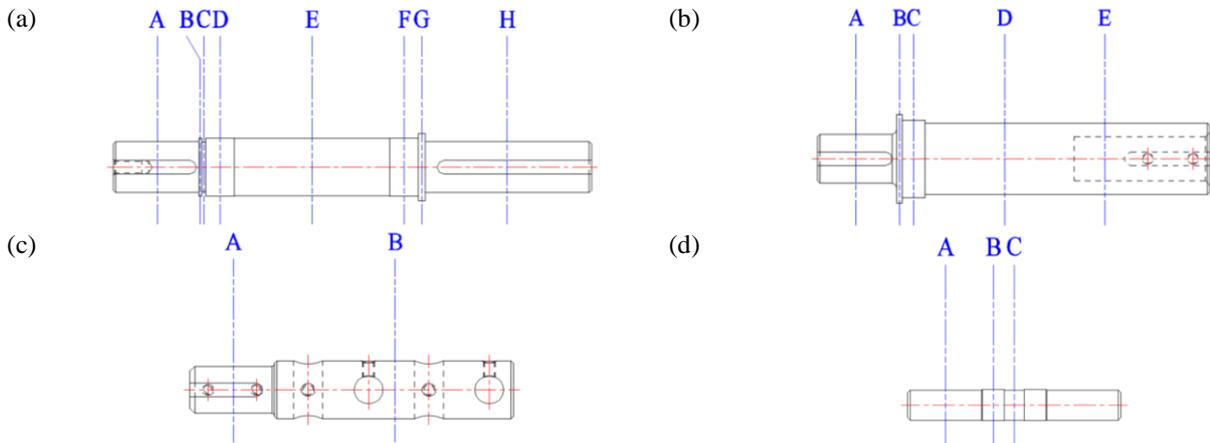


Figure 5. (a) Main shaft, (b) Middle shaft, (c) Impeller support shaft, (d) Impulsor

3.3.2 Loading on the main shaft of the grinding system

The shafts of the grinding system will be sized to resist the maximum load available on the gear motor. This eliminates any possibility of overload failure according to Tab. 3.

Table 3. Technical specifications of the Attritor mill gear motor

Technical Specifications	Value
Power (cv)	1.0
Number of poles	4
Torque (Nxm)	17
Rotation (rpm)	428

3.3.3 Loading grinding system

The transmission configuration has a ratio of 1:1, only translating the rotation of the drive shaft to the main shaft of the grinding system. In this configuration, the belt presents an embracement angle of $\theta = 180^\circ$. For a "V" profile, the face-to-face angle of $\phi = 38^\circ$ is used, and Shigley, Budynas, Nisbett (2008) can adopt a coefficient of friction of $f = 0.35$.

The tension of the belts according to Shigley, Budynas, Nisbett (2008), has the transverse loads applied on the shaft due to the tension in the belt according to Tab. 4.

Table 4. Transverse loading due to the transmission system

Loading	Value
Tensile side (N)	10.45
Loose side (N)	306.23
Total cross loading	316.69

These loads must be taken into account for analysis of the main shaft plus the effects of the maximum torque of the 17 Nxm gear motor.

3.3.4 Static analysis

a) Main shaft

The main shaft, due to the tension in the belt shows a bending moment to be analyzed. The transverse loads will be canceled on the bearing. The bearings are in sections D and F, the reactions in each of them is demonstrated in Tab. 5

Table 5. Reactions of main shaft bearing areas

Reaction loading	Value
Force in section D (N)	448.24
Force in section E (N)	-131.55

From this data, it is already possible to plot the bending moment diagram of the shaft discussed, as Fig. 6. Due to the small size of the main shaft, the loads were treated as distributed loads.

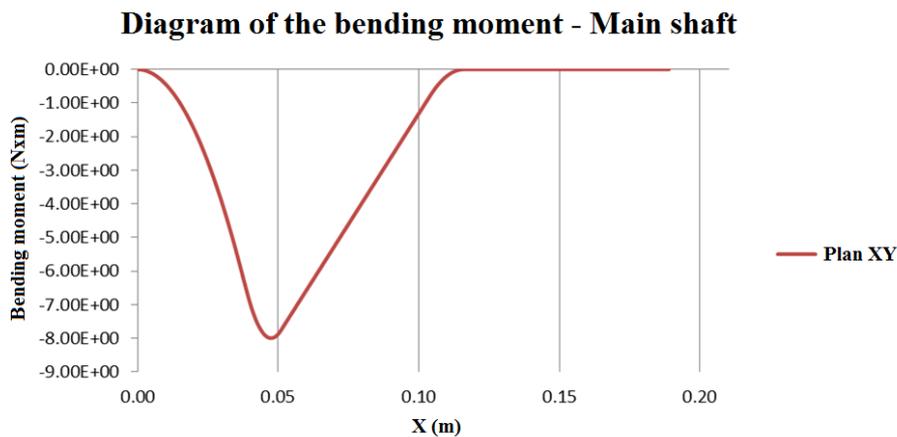


Figure 6. Diagram of the bending moment on the main shaft

In addition to the bending moments that generate axial loads, we have the torque that generates shear loads. The torque is continuous for all sections of the main shaft, and consequently for the entire grinding system, so for its dimensioning its magnitude will be 17 Nxm, equivalent to the maximum torque that the gearmotor could provide.

The main shaft will be working in a neutral environment, without risk of corrosion or temperature effects influencing its work. The material chosen for its manufacture was SAE 1020 steel.

Using the equations of the Tresca criteria with the values of bending moment, and torque for each of the sections, we obtain the values of the safety coefficient for each section of the main shaft, as well as the minimum diameters shown in Tab. if the loading factors $K_m = 2.0$ and $K_s = 1.5$.

In the design, the main shaft sections have the following diameter values as shown in Tab. 6.

Table 6. Outer diameter, safety factor and minimum diameter for main shaft

Specifications	Section							
	A	B	C	D	E	F	G	H
Outer diameter (m)	2.20E-02	2.50E-02	2.10E-02	2.50E-02	2.45E-02	2.50E-02	2.90E-02	2.20E-02
Safety Factor	13.01	18.89	10.99	17.72	16.86	20.89	32.65	14.26
Min. diameter (m)	1.18E-02	1.18E-02	1.19E-02	1.21E-02	1.20E-02	1.14E-02	1.14E-02	1.14E-02

b) Middle shaft

Proceeding in an analogous way to the previous section for analysis of the main shaft the conditions of the middle shaft are verified. Some changes must be taken into account, the middle shaft will work in a more aggressive environment, and although it is not directly linked to the reduction of the particles, can occur its contact with gases or work at high temperature.

The material specified for the middle shaft is AISI 316 stainless steel, which has good corrosion and work resistance at temperatures above ambient. Its properties are:

It is also considered that the transverse loads do not act on this branch of the milling shaft because they have been neutralized in the main shaft bearing. Therefore, the only working load is the continuously torque of 17 Nxm, and the bending moments are zero. The following results are obtained for the secondary tree according to Tab. 7.

Table 7. Outer diameter, safety factor and minimum diameter for middle shaft

Specifications	Section				
	A	B	C	D	E
Outer diameter (m)	2.20E-02	4.00E-02	3.50E-02	3.20E-02	3.20E-02
Safety Factor	12.61	75.77	50.76	38.79	32.87
Min. diameter (m)	1.36E-0,2	1.36E-02	1.36E-02	1.36E-05	1.36E-02

c) Impeller support shaft

The impeller is located inside the milling container of grinding and will be in contact with the material to be ground. It should have good wear resistance and chemical composition similar to the powder to be reduced in the process. It will be made of stainless steel AISI 316, and the results of its static analysis are obtained and presented in Tab. 8, in a way analogous to the intermediate axis.

Table 8. Outer diameter, safety factor and minimum diameter for impeller support shaft

Specifications	Section	
	A	B
Outer diameter (m)	2.00E-02	2.50E-02
Safety Factor	6.31	12.33
Min. diameter (m)	1.36E-0,2	1.36E-02

d) Impeller

The impellers will also be manufactured with AISI 316 stainless steel for the same considerations as the impeller support shaft. The results obtained are according to Tab. 9.

Table 9. Outer diameter, safety factor and minimum diameter for impeller

Specifications	Section		
	A	B	C
Outer diameter (m)	1.20E-02	1.25E-02	1.20E-02
Safety Factor	2.55	2.88	2.55
Min. diameter (m)	1.10E-02	1.10E-02	1.10E-02

3.3.5 Fatigue analysis

Resistance limit modifiers will be determined using the stress concentration factors that will be analyzed separately for each part of the milling system. According to Shigley (1984), the values of were considered Tab. 10.

Table 10. Factors modifying the fatigue resistance limit

Resistance limi tmodifying factors	Value
K_a	0.90
K_b	0.85
K_c (rotary bending)	1.00
K_c (torsion)	0.59
K_d	1.00
K_e	0.81

Then, for the materials of the shafts it presents the following values for the modified fatigue strength limit listed in Tab. 11.

Table 11. Modified fatigue strength limit

Resistance limit	Value
SAE 1020 Steel (Pa)	1.47E+08
AISI 316 Stainless steel (Pa)	9.53E+07

a) Main shaft

To finalize the fatigue analysis procedure, it remains to identify the fatigue stress concentration coefficients. In main shaft geometry, most stress concentrators occur by projections or recesses, it is recommended to use rounding at the sharp corners with a minimum radius of 0.5 mm. In sections A and H, the keyway is the main stress concentrating agent, so rounding is also recommended for sharp edges to reduce stresses. As there were no specifications in the project, a rounding of 0.5 mm.

According to Dieter (1981) we obtain the most critical values of q , K_t and K_{ts} for each of the main shaft sections presented in Table 12. Then it is possible to estimate the values of K_f and K_{fs} .

Table 12. Stress concentrators, safety factor and minimum fatigue diameter on main shaft

Specifications	Section							
	A	B	C	D	E	F	G	H
q	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6
K_t (bending)	2.14	1.90	1.70	1.70	1.90	1.90	1.95	2.14
K_{ts} (torsion)	3.40	1.35	1.10	1.10	1.30	1.30	1.60	3.40
K_f (bending)	1.68	1.54	1.42	1.42	1.54	1.54	1.57	1.68
K_{fs} (torsion)	2.56	1.23	1.07	1.07	1.20	1.20	1.39	2.56
SafetyFactor	7.26	16.90	10.54	15.73	13.87	25.81	34.96	8.29
Min. diameter (m)	1.43E-02	1.23E-02	1.21E-02	1.26E-05	1.28E-02	1.07E-02	1.12E-02	1.37E-02

b) Middle shaft

In the same way as in the main shaft analysis, the middle shaft is used and the results are shown in Tab. 13.

Table 13. Stress concentrators, safety factor and minimum fatigue diameter on middle shaft

Specifications	Section				
	A	B	C	D	E
q	0.65	0.65	0.65	0.65	0.65
K_t (bending)	2.60	2.40	2.40	2.40	2.20
K_{ts} (torsion)	3.40	1.50	1.50	1.50	3.20
K_f (bending)	2.04	1.91	1.91	1.91	11.78
K_{fs} (t torsion)	2.68	1.35	1.35	1.35	2.54
SafetyFactor	4.70	56.12	37.60	28.74	12.94
Min. diameter (m)	1.65E-02	1.32E-02	1.32E-02	1.32E-05	1.62E-02

c) Impeller support shaft

Similar to the middle shaft analysis, the impeller support shaft is used, and the results are shown in Tab. 14.

Table 14. Stress concentrators, safety factor and minimum fatigue diameter on impeller support shaft

Specifications	Section	
	A	B
q	0.6	0.6
K_t (bending)	2.60	1.90
K_{ts} (torsion)	3.40	3.40
K_f (bending)	1.96	1.54
K_{fs} (t torsion)	2.56	2.56
SafetyFactor	3.70	7.23
Min. diameter (m)	1.63E-02	1.63E-02

d) Impeller

Analogously to the impeller support shaft analysis, we proceed to the impeller and the results are shown in Tab. 15.

Table 15. Stress concentrators, safety factor and minimum fatigue diameter on impeller

Specifications	Section		
	A	B	C
q	0.6	0.6	0.6
K_t (bending)	2.00	2.00	2.00
K_{ts} (torsion)	3.40	3.40	3.40
K_f (bending)	1.60	1.60	1.60
K_{fs} (t torsion)	2.56	2.56	2.56
SafetyFactor	2.1	2.4	2.1
Min. diameter (m)	1.17E-02	1.17E-02	1.17E-02

3.3.6 Simulation

Attritor mills were simulated in static linear analysis in Altair Hyperworks 2017 software and are presented in Fig. 7.

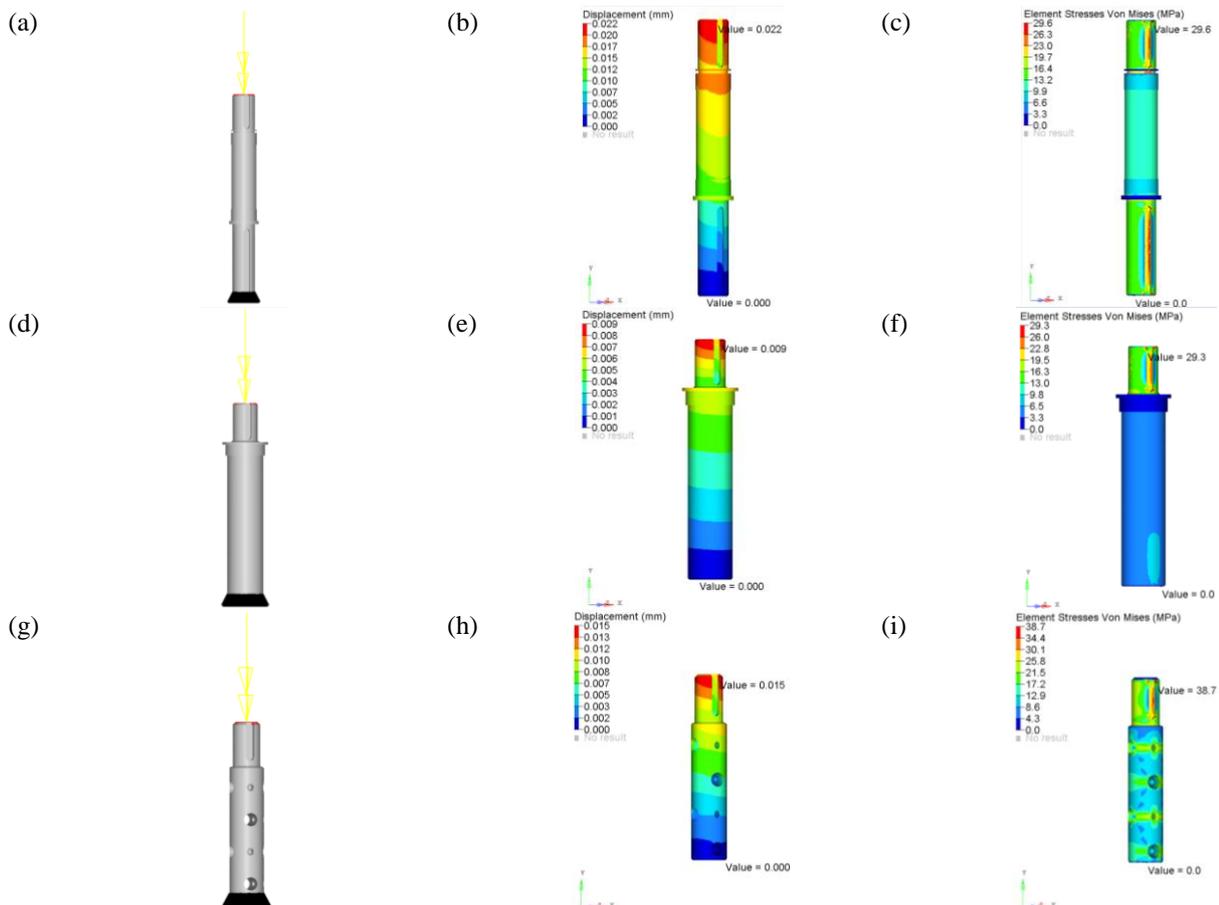


Figura 7. (a) Loads applied to main shaft, (b) Displacement of main shaft, (c) Stress of main shaft, (d) Loads applied to middle shaft, (e) Displacement of middle shaft, (f) Stress of middle shaft, (g) Loads applied to impeller support shaft, (h) Displacement of impeller support shaft, (i) Stress of impeller support shaft

4. DISCUSSÕES

All the technical requirements requested for the Attritor mill project were met to meet the needs of the use of the equipment at the university.

It was verified that the design through static analysis, fatigue analysis and simulation analysis of the main shaft has a safety factor of 363% higher than stipulated in the project. In section "A" the design has a diameter of 22 mm and it could be up to 14.3 mm in diameter.

It was verified that the sizing by means of static analysis, fatigue analysis and simulation analysis of the middle shaft has a safety factor of 235% higher than stipulated in the project. In section "A" the design has a diameter of 22 mm and it could be up to 16,5 mm in diameter.

It was verified that the design through static analysis, fatigue analysis and impeller support shaft simulation analysis has a 185% higher safety factor than stipulated in the project. The section "A" the design has a diameter of 20 mm and it could be up to 16.3 mm in diameter.

It was verified that the sizing by means of static analysis, fatigue analysis and impeller simulation analysis has a 5% higher safety factor than stipulated in the project. Sections "A" and "B" the design has a diameter of 12 mm and this could be with a diameter up to 11.7 mm.

5. CONCLUSIONS

All the technical requirements requested for the Attritor mill project were met in order to meet the need to use the equipment at the university. It was verified that the sizing by static analysis, analysis by analysis and analysis by simulation of the main axis is with a security factor 363% greater than stipulated no project. The "A" section of the design has the diameter of 22 mm and the existence of a method with a diameter up to 14.3 mm. It was verified that the sizing by means of static analysis, analysis by analysis and analysis by simulation of the intermediate axis is with a 235% higher safety factor not stipulated without design. In section "A" the design has the diameter of 22 mm and the existence of a method with a diameter up to 16.5 mm.

It was verified that the sizing by static analysis, analysis by analysis and analysis by simulation of the propeller port axis has a 185% higher safety factor not stipulated without design. Section "A" the design has a diameter of 20 mm and the existence of a method with a diameter up to 16.3 mm

6. ACKNOWLEDGEMENTS

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