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Parametric structural optimization of conveyor idler rollers

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Abstract. Conveyor rollers are widely used in the mining industry for ore transportation. However, due to the severe operating conditions, these components may fail prematurely and thus cause high maintenance costs. The failure of these rollers is usually directly related to the service life of their bearings. This service life is greatly influenced by the angular deflection of the bearings. Within this context, this work presents an optimization study of a metallic conveyor roller. The purpose is to find a roller with lower mass, however, maintaining the angular deflection of the bearings in an acceptable range of operation, without significant changes in the roller's shape to do not affect the current manufacturing processes. To define the optimization design problem, the study is based on a Brazilian standard, where factors such as: type of roller, load, allowable stresses and angular deflections are defined. The optimization algorithm is coupled to a radial basis functions (RBF) metamodel which predicts the structural response of the roller. The RBF metamodel is built, and iteratively refined, through finite element analyses performed in the commercial code Ansys. The optimization results indicate the possibility of obtaining a roller design with lower mass and higher stiffness than those manufactured nowadays.

Keywords: Conveyor belt rollers, Structural optimization, Metamodeling, Radial basis functions

1 Introduction

The use of conveyor belts is an efficient and widely used transport alternative in the mining sector. This means of ore transportation is carried out with the movement of a belt, supported by several rollers, over which the ore is dumped. The maintenance of these rollers, however, besides generating a high annual cost, promotes risks to the physical integrity of the operator responsible for the replacement of the component after its failure and due to its high mass. Therefore, the study of these rollers is extremely important and is the focus of this work, where ideal dimensions are searched, aiming to reduce the mass while ensuring that the roller design requirements based on the standard ABNT NBR 6678.2017 [1] are respected.

A roller commonly used in these conveyor belts is selected as the object of study. A parametric structural optimization is performed in order to find a geometry with the same original design, however changing only some dimensions of pre-established parameters, such as, for example, the diameter of the shaft where the bearings are supported. Thus, it is considered that no significant changes in the manufacturing processes are necessary, which would generate higher costs for their production.

In an optimization process, regardless of the selected solution method, several iterations are usually performed, which can require a high computational time and cost. Depending on the complexity of the problem, using only numerical simulations becomes impractical [2]. In these situations, an alternative is the use of metamodels to reduce the computational cost and the number of iterations necessary to carry out the optimization. According to [3], a metamodel, (or surrogate model) is a simplified model to approximate and replace a high-fidelity model. In practice, it corresponds to an approximate function that represents an experiment or physical phenomenon, being built from a number of sample tests from simulations. These concepts are employed here through two computational tools: the finite element commercial code Ansys Workbench and the Matlab platform. In the first, static analysis is performed simulating the operating conditions of the roller, and from them, stress distribution and displacements of the roller are obtained for different values of the design parameters. In the second one, a metamodel and an optimization scrip, which interacts with the finite element code to find the optimal design parameters, are developed.

2 Theoretical fundamentation

2.1 Conveyor belt rollers

Conveyor belt rollers structurally consist of three types of structural components: bearings, shaft and rollers. The bearings are responsible for allowing the rotational movement of the roller, while transferring the load received in the system to the shaft. The shaft supports the load of the roller and is fixed on two supports at its ends, preventing its vertical and axial movement. The rollers, on the other hand, directly receive the application of force and are constantly in contact with the belt. Regarding the selection, operational conditions and dimensioning of these components, manufacturers follow the ABNT NBR 6678.2017 recommendations. Following the standard, factors such as: load applied, type of roller, maximum bending stress, maximum deflection angle and roller dimensions are taken into account based on the belt width and the transported material.

According to [1], the idler studied is classified as a triple load roller, in which it receives this nomenclature due to its function of supporting the weight of the ore along the belt, and in its original configuration, to have three rollers positioned side by side on the same easel, as shown in Figure 1.

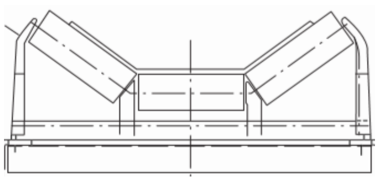

Figure 1. Positioning of triple load rollers ([1])

Among the three rollers, the one that is most susceptible to fail, conditioned to receive a greater weight, is the one that is positioned at the center of the configuration, which according to [1], for a shaft diameter of 45 mm, must support a load of 11662 N, and the maximum bending stress should be lower than 100 MPa. In the region where the bearings are positioned in the shaft, the allowable angle of misalignment is nine minutes (9’). This angle is represented by β (see Figure 2).

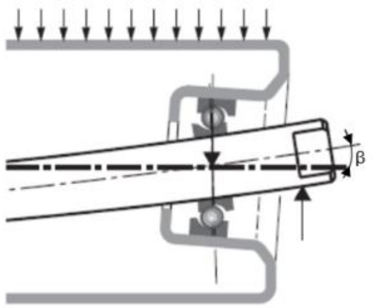

Figure 2. Shaft deflection representation (adapted from [1])

The area corresponding to the load which is transmitted from the belt to the roller is obtained through Hertz contact equations [5]. Although the belt is a hyperelastic material, the contact stress values are very low, due to this a linear-elastic behavior is considered in this calculation. Also, taking into account the Saint-Venant’s principle, a constant contact load distribution is assigned because the higher stresses are located far from the applied load. Thus, the value found for the width of the contact band is 12.4 mm. The dimensions of this band do not have much influence on the deflection caused in the shaft, however, it is important in the intensity and distribution of the stresses over the roller.

2.2 Metamodeling based optimization

According to [6], the generalized modeling process using metamodeling is performed in three stages: (i) selection of sampling points via DOE (Design of Experiments), (ii) construction of the metamodel and (iii) validation. In the first stage, “experimental” points (sample) are generated, each sample corresponding to a different combination of parameters. The choice of technique to perform these combinations has a great influence on the accuracy of the developed metamodel [6]. Here, the Latin hypercube method is employed because the distribution of points throughout the space of each variable occurs in a uniform manner. The sample points are located so that there is no superposition of the orthogonal projections of these points over the axis. As the Latin hyperbube algorithm can generate points randomly, this consequently does not guarantee that the design domain is properly covered. Hence, the more points are used, the better the representation of the original function will be.

From the points defined in the DOE and their responses after simulations, the metamodel is constructed. In this study, radial basis functions (RBF) is chosen as the metamodeling method. According to Forrester, Sobester and Keane [7], RBF technique corresponds to an interpolation in which it combines several bases, which are simple and radially symmetrical functions centered on the various points spread over the domain. In order to built a more accurate metamodel, it is customary to use new points (i.e., infill points) to be simulated and include them in the sampling points dataset to refine the metamodel during the optimization process.

3 Methodology

3.1 Numerical simulation of the idler roller

In the Ansys Workbench finite element software, static analyses of the roller are performed to build the initial metamodel and to update it using infill points. Aiming to save computational time, as shown in Figure 3, only a quarter of the geometry is modeled and symmetric conditions are imposed. An external distributed load equivalent to 2915.5 N, which corresponds to a quarter of the total load, is imposed to the model. Another feature that is used in the finite element modeling is the representation of bearings as springs, because the mesh of this type of element, independent of refining, do not guarantee convergence of results for the stress. The spring stiffness value used is 94856 N/mm (corresponding to ½ of the total value due to symmetry), which calculation is based on Gargiulo [8].

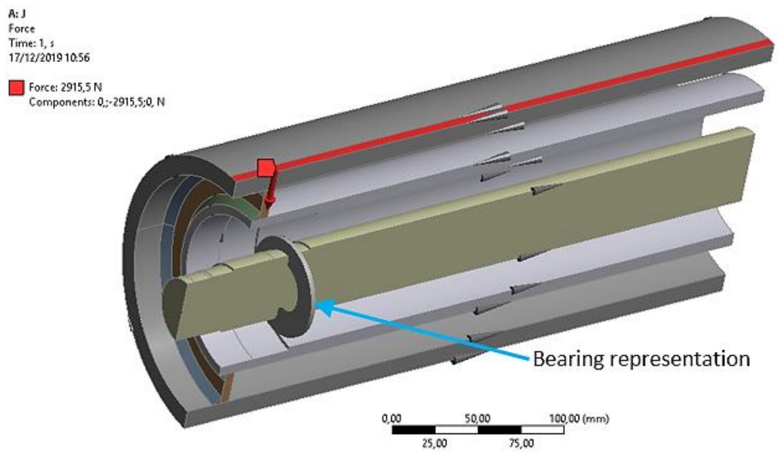

Figure 3. Solid model of the roller (1/4 of the geometry)

The finite element type defined for the mesh had both hexahedral and tetrahedral formats, of average size of 6mm, found after a mesh convergence study, and which allowed an acceptable precision for the purposes of the work.

The material of the roller and shaft is a low carbon steel, with the following properties: Poisson ratio of 0.3, Young’s modulus of 210 GPa and mass density of 7850 kg/m^3.

The results sought for the simulations are: the maximum von Mises stress (S_{VM}) in the roller, as shown in Figure 4, and the minimum and maximum displacements in the roller and in shaft in the region where the bearings are positioned. With these displacements, it is possible to obtain the angular deflection of the shaft in the bearing region (β).

These results, together with the definition of the roller mass, are interpreted in a Matlab script that uses them in the optimization algorithm calculations.

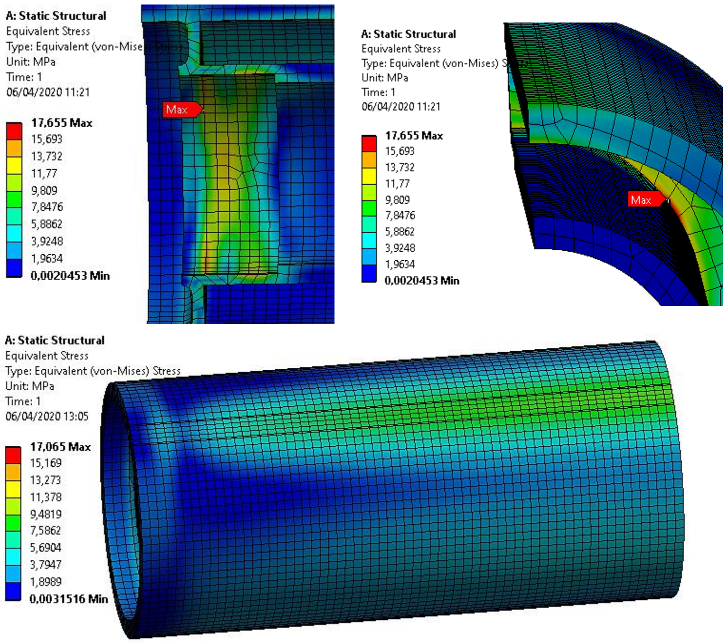

Figure 4. Von Mises stress distribution (initial design)

3.2 General aspects of the optimization

According to [4], an optimization problem is defined mathematically as the minimization or maximization of an objective function which it may be subject to constraints of equality or inequality. Within this function, there are variables (parameters) that are understood as values that are modified in the search for the optimal point. The constraints stipulate boundaries that must be met to make the design viable. The design variables of the problem here studied can be seen in a cross section representation of the roller, shown in Figure 5, and the formulation of the optimization problem is defined as:

$$\begin{cases} Minimize\ \{Mass\ (D_1, D_2, D_3, D_4)\} \\ subject\ to: \begin{cases} 134\text{ mm} \leq D_1 \leq 168\text{ mm} \\ 110\text{ mm} \leq D_2 \leq 131{,}6\text{ mm} \\ 87\text{ mm} \leq D_3 \leq 91\ mm \\ 54\text{ mm} \leq D_4 \leq 63\text{ mm} \\ S_{VM} \leq 100\ \text{ MPa} \\ \beta \leq 9' \end{cases} \end{cases} \tag{1}$$

If for a given point one or more constraints are violated, a penalty is assigned to the objective function, making its value higher for that point. The definition of the penalty function is based on the methodology described by [9], where, by adapting to the problem here studied, it can be written as follows:

$$f_p(\mathbf{x}) = f(\mathbf{x}) + \sum_{i=1}^{m} \left(\left(\frac{R_{ec}(\mathbf{x})}{g_i(\mathbf{x})} \right)^k - 1 \right) \cdot f(\mathbf{x}) \cdot \delta_i \tag{2}$$

where, $f_p(\mathbf{x})$ is the penalized objective function, $f(\mathbf{x})$ is the non-penalized objective function, δ_i a parameter that has value 1 if, the ith constraint is violated, or value 0 if the constraint is not violated, $R_{ec}(\mathbf{x})$ is the constraint value at each point studied and $g_i(\mathbf{x})$ the reference values of each constraint, for example: $S_{VM} \leq$ 100 MPa. In the equation, k is a user-defined exponent.

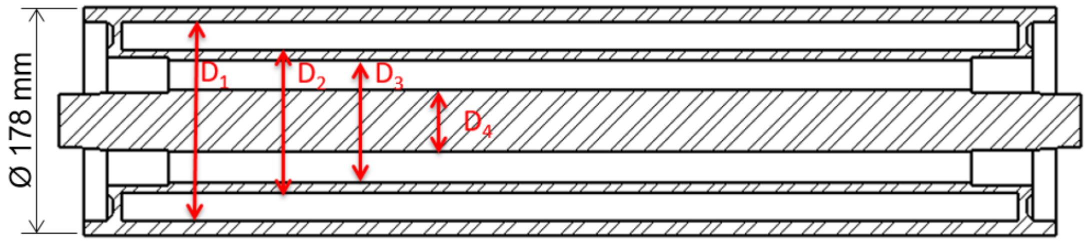

Figure 5. Representation of the design variables D_1, D_2, D_3 and D_4 in mm

Figure 6 shows a flowchart of the metamodeling based optimization strategy used here. As already mentioned, the finite element analyses are performed using Ansys Workbench and the optimizer used over the metamodel function is the Globalized Bounded Nealder-Mead (GBNM) [10].

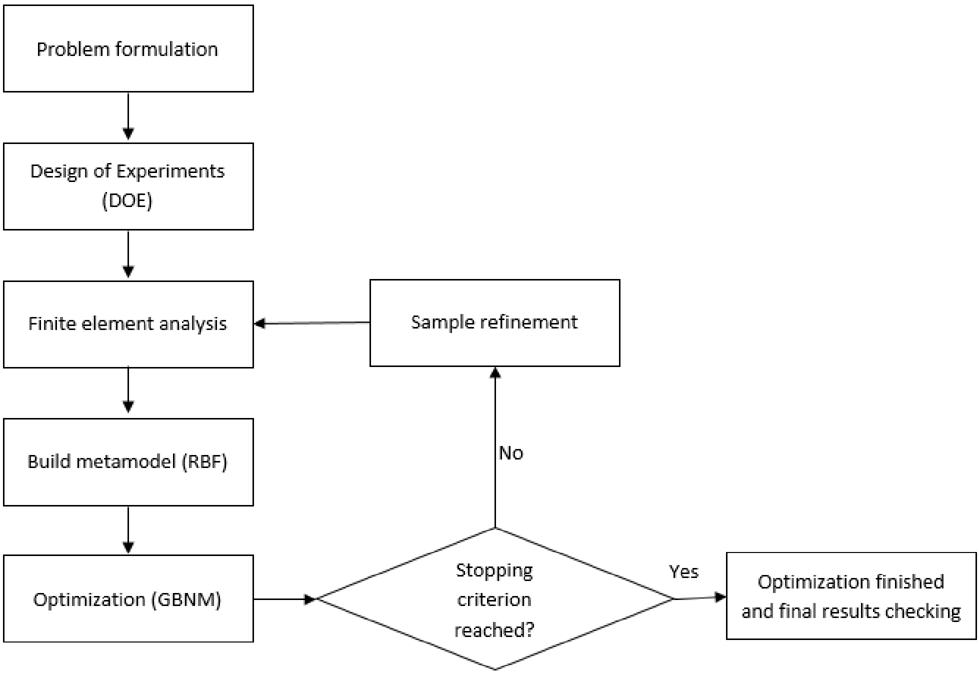

Figure 6. Optimization process based on metamodeling

The process is started with the development of the DOE, where from the points generated the simulations are carried out. With the results of each simulation, the constraints are evaluated and depending on the value obtained the mass may or may not be conditional on the penalty. Based on these results, the metamodel is developed and then submitted to the optimization algorithm. After obtaining the optimal point of this initial metamodel, a new point is created randomly. With this, the finite element simulations are carried out for these two points (the best one found and a random one), and, in this way, the metamodel is updated and so on until a previously defined total number of iterations is reached (stopping criterion).

4 Results

Three optimization cases were carried on a 6 GB RAM computer. The first one has 20 simulations (case 1), the second one 50 simulations (case 2) and the third one 80 simulations (case 3). The time required to perform the optimization in case 1 was approximately 1h, being 15 simulations used in the DOE to build the metamodel, three simulations with the optimal points found in the developed metamodels and the other two using random values for the variables, alternating between the metamodel's optimal point and the random choice. In case 2, the optimization served only as a test to verify the behavior of the metamodel in relation to the violation of the stress constraint, since in case 1 this constraint is not violated. In case 3, similarly to case 2, it served as a metamodel test but focused on violating the angle constraints. In order to analyze the constraints, it was stipulated for case 2 that instead of 100 MPa, the stress should not exceed 60 MPa, and for case 3, instead of 9', the deflection angle should not exceed 5'.

In case 2, the required time to perform the optimization was approximately 2:18h, using 30 simulations in the DOE, and the latter 20 simulations were obtained with the 10 optimal points found in the developed metamodels and the other 10 using random values for the variables, alternating between the metamodel's optimal point and the random. In case 3, the time was 2:12h, with 30 simulations for the DOE, and for the other 50, 25 simulations of optimal points and 25 simulations of random points.

The roller mass of the initial design is 69.136 kg and, after the optimization in case 1, it reached 46.795 kg. The evolution of the mass reduction during the optimization process can be observed in Figure 7.

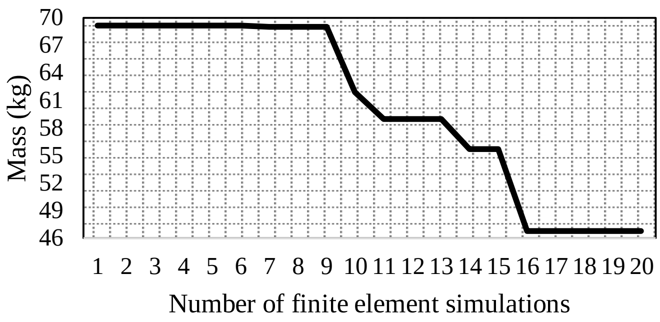

Figure 7. Roller mass evolution in case 1

In case 2, after the optimization, the value of 49.119 kg is found for the objective function. The evolution of the mass for this case can be observed in Figure 8.

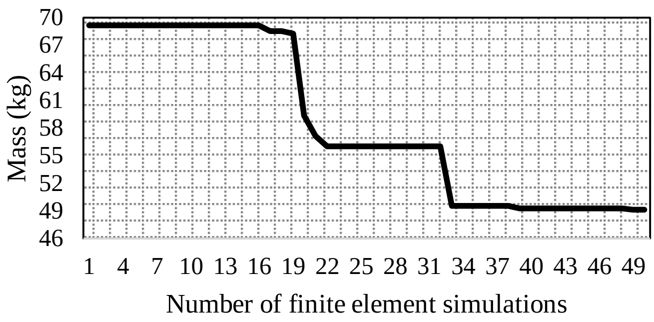

Figure 8. Evolution of the roller mass in case 2

In case 3, the value found of the objective function is 47.269 kg at the end of the optimization process. The evolution of the mass in this case is present in Figure 9.

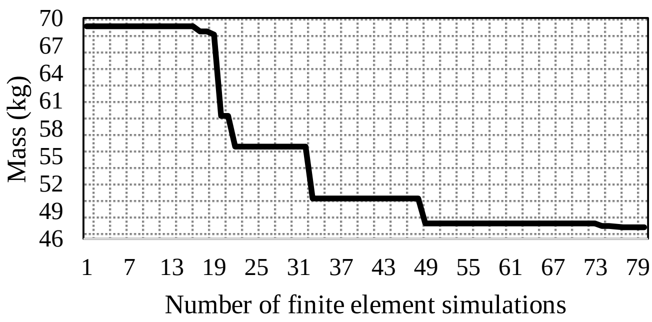

Figure 9. Evolution of the roller mass in case 3

The values obtained for this final roller configuration compared to its initial version are present in Table 1. In this table, the mass values do not include the bearing masses and the components of the roller labyrinths seals.

Table 1. Initial and optimal roller's configurations

Propriety	Initial design	Final design (case 1)	Final design (case 2)	Final design (case 3)
Nº of simulations	-	20	50	80
Mass (kg)	69.136	46.795	49.119	47.269
D_1(mm)	154.00	168.00	167.04	167.92
D_2 (mm)	111.60	110.00	110.00	110.00
D_3 (mm)	91.00	91.00	89.82	91.00
D_4 (mm)	54.00	54.00	54.04	54.86
S_{VM} (MPa)	17.065	69.272	57.260	67.980
β (’)	4.962	5.116	5.091	4.860

5 Conclusions

With the optimization strategy employed here, respecting the design constraints, a reduction of the roller's mass around 32.3% was achieved in case 1. For this case, a small amount of simulation points was required to build an accurate metamodel compared to the other two cases. This occurred because there were no constraints violations during the optimization process. For case 1, the corresponding values of the safety factors found for S_{VM} and β were: 1.443 and 1.759. In cases 2 and 3, it was possible to demonstrate the functionality of the penalty function if the constraints are violated during the optimization process.

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