

Experimental and Finite Element Study of Bending Fatigue Failure: A Case Study on Main Shaft of a Gyrator Crusher

Rahim Sotoudeh Bahreini, Alireza Foroughi Nematollahi, Akbar Jafari

Abstract—This study investigates the mechanism of a Gyratory crusher-located in Golgozar mining and industrial Co. specifically with a focus on stresses distribution and fatigue failure of its main shaft. At first step, the cross section of the fractured shaft is studied, and the crack growth is analyzed. Then, the rotational motion of the shaft and the oil temperature of oil circuit of equipment are monitored. Condition monitoring is used to help finding a better modification. Based on the results of this study, the main causes of shaft failure are identified, and corrective solution is offered to increase crusher performance, especially its main shaft life. To predict the efficiency of the proposed modification, finite element simulation is performed, and its results are compared with the similar modified cases. The comparison and interpretation of simulation results confirm the efficiency of proposed corrective method.

Keywords—Fatigue failure, finite element method, gyratory crusher, condition monitoring.

I. INTRODUCTION

FATIGUE failure is a very common event for mechanical equipment when they are imposed a fluctuated dynamic load. This failure begins with a small crack and occurs when the crack grows [1]. The equipment which is under simple or relatively well-known load conditions can be analyzed by analytical methods. Nonetheless, most of the equipment is operated under complicated load conditions and it cannot be analyzed just based on the theoretical methods. Hence, in such cases, using numerical methods such as finite element and experimental studies is useful and effective. As a practical example, a gyratory crusher shaft under a complex loading and supports is studied in this research. This equipment has a very important role in mineral processing factories. Therefore, study of design and analysis of its components is scientifically and practically very important. Gyratory Crusher, as an essential equipment in Golgozar mining and industrial company, is placed after the iron ore extraction area and acts as a primary mill to crushed bulk iron ore. The maximum size of output ore from this crusher is about 20 centimeters, which is suitable for upward mills. Different types of crushers are used in mining industry. But, two types of Jaw and Gyratory

crushers are more common in mineral processing lines. The Jaw crusher, as its name demonstrates, acts as mouth of some animals; it crushes the ore such like it chews it. This type of crusher is used in low to medium feed rate of material. For higher feed rate, Gyratory crushers are chosen [2]. Fig. 1 shows schematic of Gyratory crusher, and a more complete view is shown in Fig. 2. It can be seen that the central main shaft which is covered with abrasion resistant liner is rotating pendulum-like at the center of conical shell. This reciprocating movement causes changes in size of the exit area of crusher; therefore, ore crushing process is done by this mechanism. In fact, this type of crusher can be assumed as infinite number of very small jaw crushers located on the circumference of a circle. Regarding this description, it can be concluded that the efficiency of Gyratory crusher is much higher than jaw crusher. Although there is no exact equation for calculating the capacity of the crushers, it can be empirically said that, in the same condition of output size, capacity of Gyratory type is about three times of the Jaw crusher. Also, Gyratory type is recommended when the capacity of production line is more than 900 t/h. In addition, in a similar capacity due to the circular shape, the required space for Gyratory type is less than jaw crusher. Moreover, due to severe fluctuation of stress during the operation, a stronger foundation and supports are needed for Jaw crusher [2]. According to the special technology of crusher, very small numbers of manufacturers are able to manufacture and maintain the equipment. Also, providing spare parts leads to high costs of repair and maintenance. Thus, it is necessary and inevitable to study design optimization, improvement of equipment, and how to operate it better. Literature review indicates that focuses of research works are based on empirical analysis, particularly dealing with internal friction and abrasion of components of this type of equipment [4]-[6]. Accordingly, a good profile of the central shaft has been proposed for better crushing [7]. Furthermore, because of high energy consumption of this equipment, the optimal choice of that has been investigated [8].

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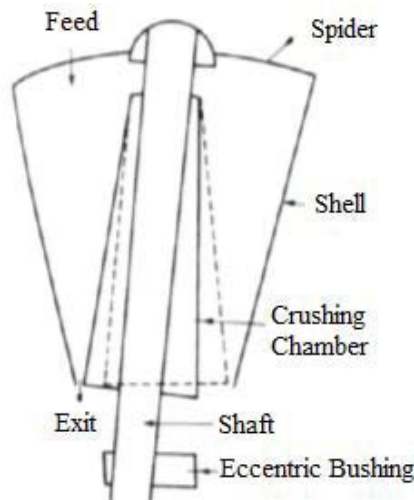


Fig. 1 A general view of Gyratory crusher [2]

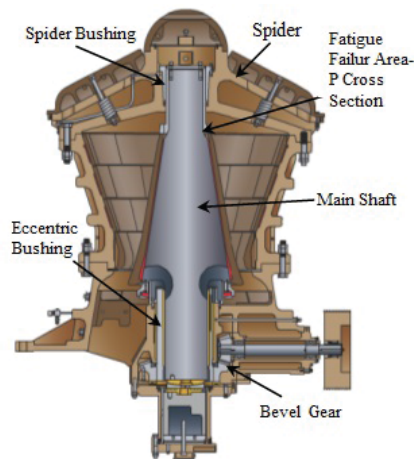


Fig. 2 Schematic of cross-section of the Gyratory crusher [3]

II. PROBLEM DEFINITION

The main shaft of Gyratory crusher is mounted inside an eccentric bushing which is located in downward of equipment. The rotational speed of shaft is equally $\Omega=120$ rpm. In fact, this bushing plays role of a crankshaft. The main shaft has two kinds of movement, first the gyratory (or oscillatory) movement due to eccentricity and second is a freely spin rotation ω around itself. The lubricant between the shaft and the bushing acts to reduce the contact and friction force, it conveys friction load from bushing to the shaft and causes the spin rotation. As it can be seen in Fig. 2, there is an arm-like support upside of crusher shell named "Spider". This heavy duty spider is made of cast steel and is of the bar type with a large hub in the center. The top sides of the spider arms are beveled shape to deflect damages from the impact, and the undersides are rounded to facilitate entry of the material into the crusher. The spider hub is bored to receive the bronze bushing [9]. The goal of present study is to analyze the main shaft of crusher and determine possible reasons for its failure. Also, this

research tries to modify bushing design in order to prevent or delay the fatigue failure. In practice, failure of main shaft leads to down time of equipment. If this failure happens when crusher is running, it may cause serious damages to the supports and bushings.

III. METHODS OF RESEARCH

Here, there are two approaches, empirical and finite element analysis, to study and analyze the issue. In the empirical method, the observation and interpretation of the fracture surface, capturing the motion of the shaft at different working conditions, and thermography analysis of lubrication system are followed. In finite element method, the commercial software ABAQUS is used.

A. Investigating the Fatigue Fracture Surface

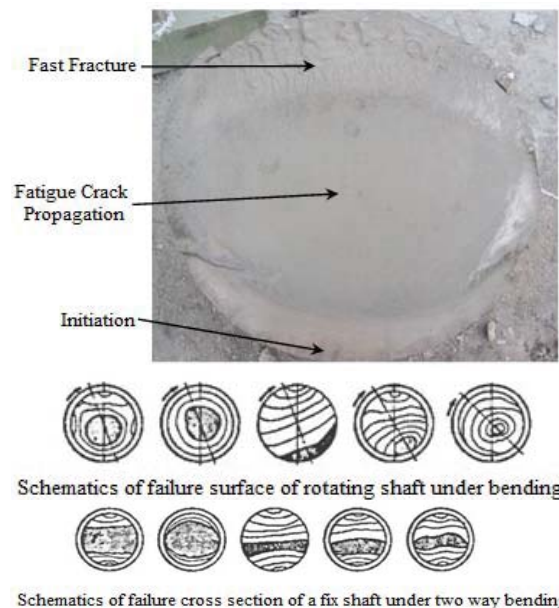


Fig. 3 Different regions of fractured section of the crusher shaft and comparison with standard schematics of fatigue failure

Fig. 3 shows fatigue fracture surface of shaft; this area is located under spider bushing and is shown by *P* in Fig. 2. Also, in this picture, images of the standard schematics of fatigue fractured surfaces are presented for comparison [10]. In fact, this area is located below the spider bushing which is circular with a diameter of 609.5 mm. At this area, stress concentration occurs due to the reduction in diameter of shaft. It can be seen that the failure begins from a point on circumference of shaft. Then, the crack growth begins in a crescent-shape area. The next area which is elliptical shape is representing fatigue growth and progressive movement of crack. Following that, the area containing the crack has grown in the same direction. Finally, rapid failure occurred; the crescent area with a rough surface and some plastic deformation indicates this final part of failure. Comparison between real cross section of failure shaft with standard schematics in which some of them are depicted in Fig. 3

shows that pattern of real fracture surface does not match the schematics of rotational fracture surface in real. Instead, it has more similarity with the schematics of fixed surface that are imposed by two-way bending load. However, this similarity is not close enough to mention certainly that this is the type of loading.

Now, the question is “what kind of loading condition is exposed to the shaft” and “which schematic is the nature of fractured surface”? Mismatch of fractured section with standard schematics indicates that the nature of shaft loading is compound and does not match standard schematics. However, the crack growth shows that bending around direction which is perpendicular to the direction of crack growth could be the main cause of failure. By comparing the ratio of the area where rapid growth occurred with the area where crack gradually grows (the area like beach marks), it can be concluded that the failure is fatigue type. In addition, it is observed that, crack initially starts from shaft circumference and gradually growth toward one side. Therefore, before shaft fails thoroughly, it was possible to discover it by non-destructive tests. So, if there exists a permanent monitoring, it would be possible to predict the failure. Such an approach can determine the approximate time of ultimate failure. Accordingly, equipment can be stopped before expensive damages. At least, it could be possible to prevent damage the other component. On the other hand, by study of crusher mechanism, it can be concluded that in case that there is only

gyratory movement and spin rotation is absent, at any moment, direction of shaft bending changes due to eccentric movement of lower bushing. Also, loading due to crushing process changes as depicted in Fig. 4. In this case a time-dependent bending load imposed to the shaft is similar to the type of loading that shaft rotates and load is fixed. Thus, the growth of crack should be similar to the schematics of fracture of rotational surface under bending load, whereas it is not. Thus, it is estimated that this type of movement does not happen in real condition. To interpret the occurrence of this situation, the shaft mechanism will be studied precisely. As already mentioned, in addition to gyratory movement of the main shaft with the speed Ω , there is another type of rotation around the axis of the shaft; spin rotation ω . As Fig. 5 illustrates, in the situation that spin movement is getting synchronized with gyratory rotation, bending always occurs around a fixed axis. For example, points A and B are always under maximum strain and stress, and points C and D are located on the neutral axis. In fact, while the shaft rotates, the point of application of load rotates too. In the other words, this is like a fixed shaft under a one-way bending load. This condition, however, does not comply with the cross-section schematic of failed shaft. But, comparing with previous condition (rotating shaft condition) shows more similarity. To evaluate this, the operation of the equipment is monitored at different times, and the results will be interpreted as follows.

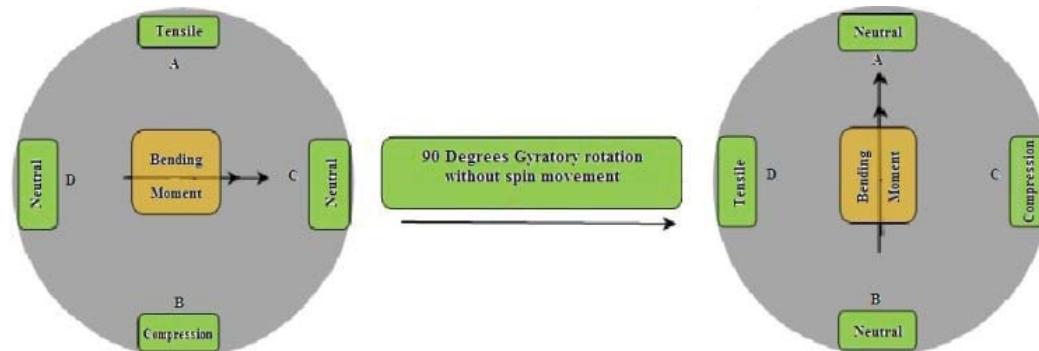


Fig. 4 Stress condition in case that gyratory movement exists and spin rotation is absent

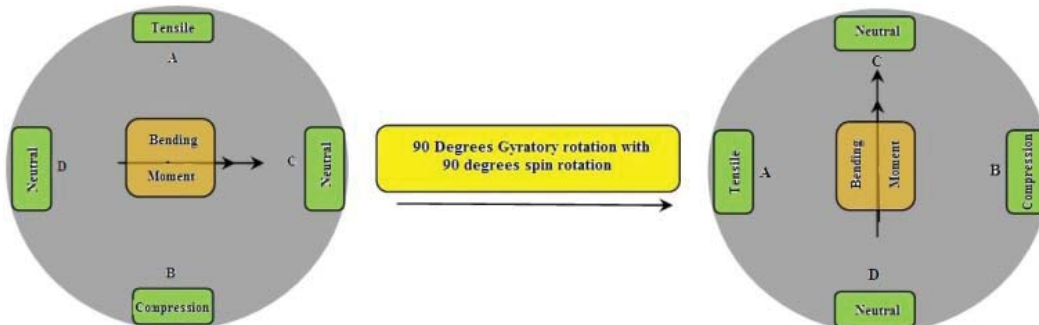


Fig. 5 Stress condition when Gyratory movement and rotational spin are synchronized

B. Condition Monitoring of Equipment

To justify our prediction on spin rotation effect, for a while shaft movement and feeding of crusher was shot in different working condition. Also, oil temperature is shot by thermo-vision camera in critical points at different times. Fig. 6 shows how the crusher is charged by iron Ore. In this study, it was found that the when crusher is charged, the spin rotation stops and during the period of crushing, almost gyratory movement dominates. However, gradually by passing the crushed Ore toward output and crusher unloads, spin rotation increases gradually. In Fig. 7, diagrams of shaft spin-rotational speed verses different times are plotted. The mentioned speed is recorded about two minutes after charge of ore simultaneous with the complete exit of ore. It can be seen when iron ore completely leaves the crushing area, spin rotational speed varies between 12 to 18 rpm on different dates. This speed approaches its asymptotic value about 30 rpm three minutes after exit of ore. It is interesting that this value is a proper fraction of gyratory speed (120 rpm). Therefore, it can be concluded that probably a direct contact happens between shaft and bottom eccentric bushing or oil shear strength between contact surfaces forms frictional torque which leads to spin rotational of shaft. Objective observation of surface of bushing and shaft circumstance indicates damages in these areas, which confirms above-mentioned hypothesis.



Fig. 6 A view of input of crusher in which charged with iron ore

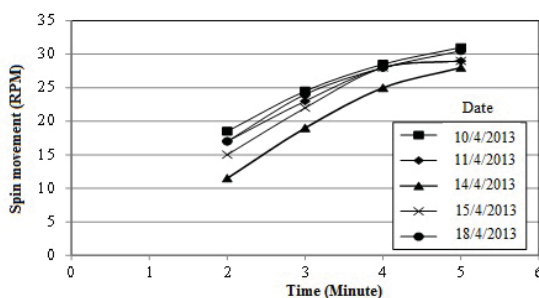


Fig. 7 Changes of shaft spin speed over the time after charging ore

C. Finite Element Simulation

Regarding finite element point of view, the end part of shaft located in eccentric bushing is modeled as a constant displacement applied into the support. This displacement equals 126 mm and leads to a bending moment on shaft. This moment maximizes at spider bushing. Therefore, P section in

Fig. 2 is the most critical point of shaft. In Fig. 8, more details of schematic and tolerances of spider bushing at upper part of shaft is shown; indeed, this is where failure happened. Diameter of spider bushing $D=609.5$ mm and its length $l=1175$ mm therefore $L/D= 1.9$. This acts as a support enduring bending moment. On the other hand, it has been shown previously that, spin movement prevents a uniform stress distribution on around shaft circumference.

According to the before-mentioned description, to improve the life of this vital equipment, restructuring its top bushing as shown in Fig. 8 is recommended. By this new bushing mechanism, the upper support acts like a spherical joint and prevents occurrence of bending moment in this area of shaft. Consequently, normal stress caused by bending does not occur.

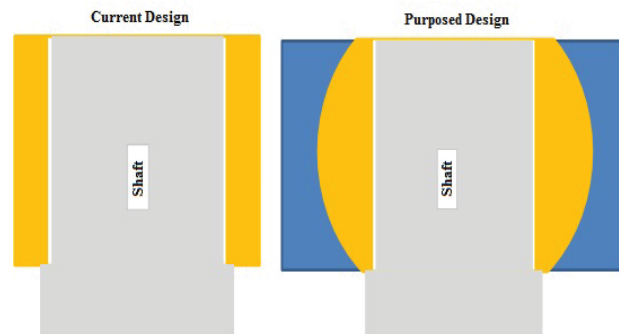


Fig. 8 Current and proposed schematic of spider bushing

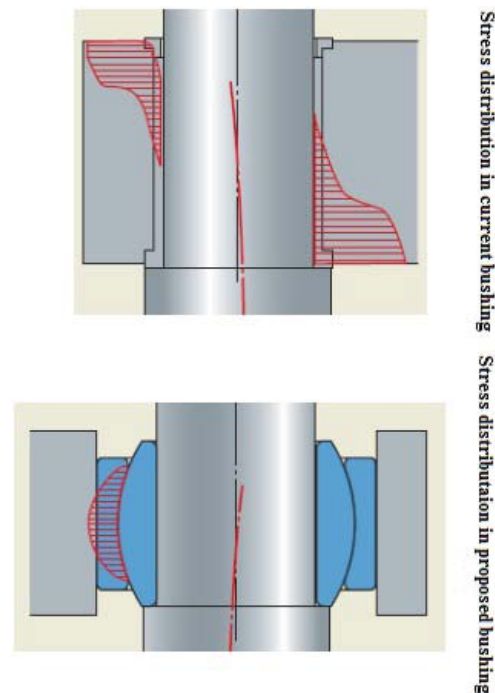


Fig. 9 The difference between stress distribution in current and proposed bushing

Fig. 9 shows stress distribution in the current and suggested

bushing. In the current bushing, stress distribution is maximum at edges, while in the offered type, it is minimized there. On the other hand, consultation with some manufacturers makes it sure that manufacturing the offered bushing is possible.

To investigate effect of new design on stress distribution in shaft, finite element modeling is performed by using ABAQUS, and the results will be discussed in brief as follows. Fig. 10 shows the model of main shaft meshed with linear hexagonal element. For modeling, elastic mechanical properties mentioned in manual of crusher are applied. Von Mises stress contours at different parts of shaft are shown in Fig. 11 based on current bushing structure. It can be seen that the maximum amount of stress occurs where the shaft diameter changes under the spider bushing. Also, there is a high similarity between lines of contours of first principal stress and lines of boundaries of crack growth areas on fractured section.

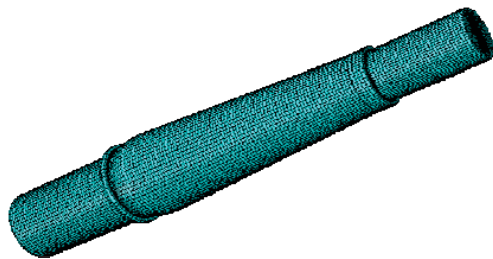


Fig. 10 Main shaft of crusher modeled by finite element

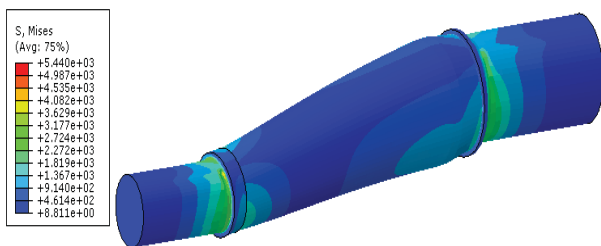


Fig. 11 Von Mises stress contours in simulated shaft

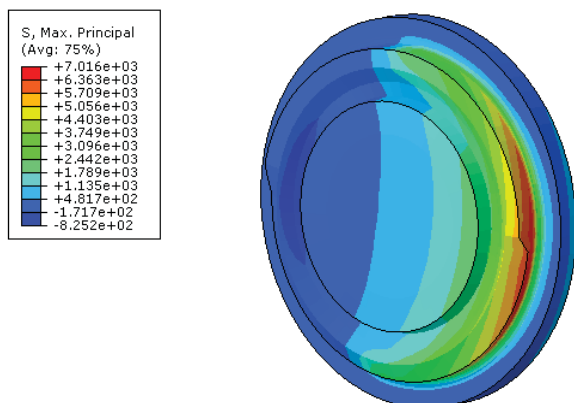


Fig. 12 First principal stress contour in critical section of shaft-section P

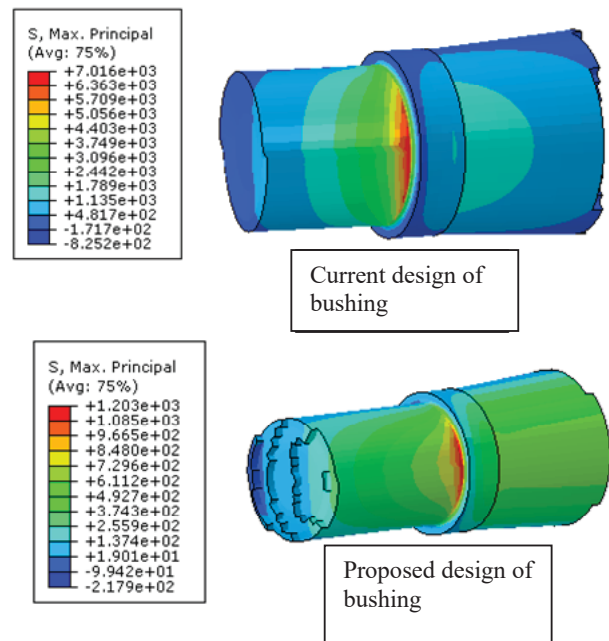


Fig. 13 First principal stress contours of current situation and after changing bushing type

Now, by applying proposed correction, finite element simulations with the same loading condition as the previous simulation carried out and the results are compared with the corresponding values. For example, in Fig. 13, first principal stress contours are shown for these two simulations.

Comparing numerical results shows that, by applying the new conditions, the principal stress which is almost equivalent to the normal stress due to bending of the shaft is reduced. The maximum value of this stress reduces from about 7.01 to about 1.2 reaching about 0.17 of the previous value. Now, loading due to crushing of iron ore should be imposed to the model then, corresponding stress on the current bushing model and the suggested model is compared. The fact is that the reaction load imposed from ore which is crushed could not be determined easily. Its actual amount and situation depends on strength and location of iron ore inside the crusher. However, to determine where the load imposed to the shaft, the crusher is monitored for a while and it is filmed. Observations confirm that the major part of crushing occurs near the exit area of crushing area where the opening is very narrow. It should be noted for large size of stones crushing load acts on upper area of shaft. However, the possibility of accurate modeling of the actual load on the shaft is not possible. Yet, in this phase of study, an assumed 1000 N load is considered and imposed to the model at two different points on shaft, 1/4 length of shaft from bottom and 1/2 length of shaft.

Results of two modeling, current and suggested bushing, are compared as will be mentioned. Consequently, Load, reaction moments and stressed created in each upper and lower bushings for mentioned loading are evaluated and compared.

Comparing stress values shows that by the implementation of the proposed method, stresses caused by crushing load

significantly reduces similar to stresses caused by eccentric movement at upper bushing. However, it is found that stress does not decrease at the lower bearing even increases. Of course, this amount of increase depends on where crushing load acts on. In fact, whatever crushing load is closer to bottom; this increase becomes more insensible at lower bushing. For example, when the crushing force is applied at 1/4 from the bottom of the shaft, stress at lower bushing, where the shaft settles, for the suggested model is 1.1 times of the current one.

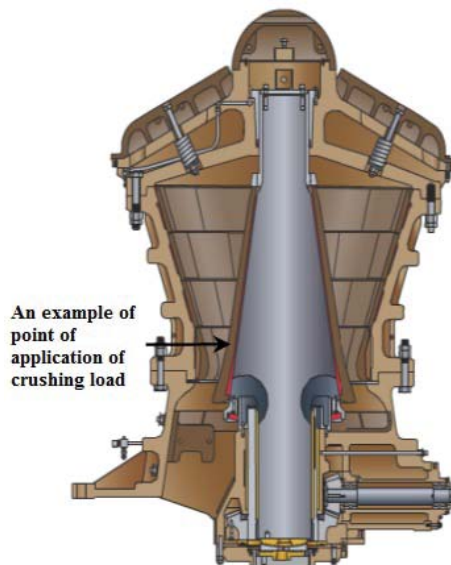


Fig. 14 A view of place of load caused by crushing of iron ore on main shaft

IV. CONCLUSION

By review of the experimental and numerical analysis performed, in this study, it was found that changing the type of bushing of equipment shaft can have a significant impact on reducing stress. It should be noted that this decline is not confined not only to the shaft but also to the bushing. Also, more suitable stress distribution happens, and subsequently, it improves the life of shaft and bushing simultaneously. To perform this change, the current cylindrical bushing must be divided into two spherical bushing that are inside each other. Further studies in references show that similar procedure performed by some manufacturers for similar equipment and the positive results are emphasized [11]. On the other hand, type of fractured surface that is monitored with non-destructive method like determining cracks by penetrating test could provide useful information about crack growth. With this method, it can be predicted when ultimate failure happens. Therefore, before reaching to ultimate failure, equipment could be stopped to prevent other component damages.

ACKNOWLEDGMENT

Authors would like to appreciate manager and experts of processing department of Golgohar mining and industrial Co.

and Gohar Ravesh Co. for their cooperation to perform this study.

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