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Failure Analysis and Prevention of a Heavy-Duty Industrial Vibrating Feeder

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Abstract Vibrating feeders play a vital role in numerous industries by facilitating the transfer of materials. Achieving the ideal material flow rate depends on the efficient operation of the feeder. Vibrating feeders are standard in material handling applications across multiple industries. This article deals with finite element model (FEM) analysis, and an investigation aimed at failure analysis and prevention of heavy-duty industrial vibrating feeders to: (i) investigate the causes of failure experienced during the dynamic phases of vibrating feeders, (ii) provide comprehensive information about the system, (iii) foresee the potential improvements to eliminate the cause of failure, and (iv) extend the lifetime of the system. The failed vibrating feeder was studied using ANSYS to investigate the stresses and deformations. Taking into account the analysis results, the location of the failure was identified, the design of the failure was improved, and the causes of the failure were eliminated by modifying the original design of the vibrating feeder. The study also clarifies the theory behind vibrating feeder technology and the importance of its design. Complete FEM analyses were conducted to calculate the stress and strain distributions together with vibration, fatigue, and frequency analyses in the original and modified vibrating feeders to reveal the differences between the two designs.

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Introduction

The vibrating feeder plays a vital role in material processing in industrial and mining enterprises. Within the mineral processing sector, applications range from sizing to feeding and washing. Key components are vibrating feeders, grills, side plates, electrical motors, and helical springs "feed" material to a process or machine. However, design errors can lead to the failure of vibrating feeders. Recently, there has been a significant surge in the focus on optimizing both the efficiency and the geometry of vibrating feeders. This increased attention highlights the unique interest in and dedication to enhancing performance.

During the operation of the vibratory feeder unit, due to the moving masses and vibration created by the motor installed at the system base, fatigue cracks can occur at some critical locations [1–4]. It is emphasized that an analysis of the vibrating screen's structural durability and lifespan is necessary to increase the efficiency of the screen [5]. Designing machines can be complex if the dynamic loading operating on the feeder structure needs to be adequately characterized, including evaluating the stress distributions [6]. Moreover, feeders work in three different operating modes: start-up, steady state, idle run, and breaking phase. The stress state will differ for each mode [6]. The transportation and separation effectiveness, together with the fatigue life of the vibrating feeder, are impacted by the acceleration, frequency, and amplitude of the screen motion; as a result, the design optimization of the vibrating feeder is a vitally important consideration [7].

Any unexpected failure on any of its components will stop the plant operation; they are considered as one of the critical equipment in the line of plant downstream equipment (Fig. 1).

The dynamic characteristics of large vibrating feeders are highly complex due to the fully coupled load effect caused by the intense excitation load initiated by exciters, the inertial load of the screen structure, and the impact load of the transferred materials. This often results in structural damage such as beam fracture or lateral plate crack [8, 9]. Modal analysis in FEM was carried out by Wang and his co-workers [10], and Chandravanshi and Mukhopadhyay [11, 12], Yy and Zhang [13], and Rao and Ratnam [14] in order to determine the weaker components of the vibratory feeder model. Based on the analysis results, their initial design was modified using the modal characteristics discovered, resulting in less stress in the trough structure and improved vibration feeder operational reliability over an extended period of time. Because of their high operating frequencies, vibrating feeders must undergo a very high number of cycles in order to meet their necessary lifespan. A life of 10^9 cycles is attained in less than three years, with an average operating rate of roughly 10^6 cycles per day. Consequently, they offer the perfect framework for assessing fatigue design requirements [15].

Fatigue Crack in Vibrating Feeder

Despite extensive efforts to optimize vibrating feeders, more research is needed to explore the potential of utilizing a feeder model to provide insights into its performance and aid in identifying failures or damages.

This article is devoted to the stress analysis of a vibrating feeder using the finite element method (FEM), optimizing the design, and overcoming failures encountered in this type of feeder. The stress distribution and deformation under both static and dynamic loads on a given vibrating feeder are investigated and analyzed. In particular, the vibrating feeder experiences forceful vibrations under high-intensity loads, making it susceptible to fatigue failure. Due to the intricate nature of its complex structure and shape, traditional design methods must be revised to assess the resulting stress levels accurately. In this article, the FEM analyses are conducted on the initial design of a failed vibrating feeder; then, under the light of the analysis, the necessary design revisions are made, and finally, the analysis is repeated on the modified feeder to make sure that the revised design meets the design requirements.

A fatigue crack is initiated and propagated after about 1150 working hours in the field at the vibrating feeder's side plate under investigation, as shown in Fig. 2. The figure also shows that the fatigue crack emanated from the stress concentration at the side plate. Furthermore, it is also worth pointing out that there was also a weldment at the zone where the fatigue crack originated. These two parameters contributed to the fatigue crack initiation from this location. The current study aims for a full detailed analysis of this failure and proposes design improvements to the current feeder to overcome the re-occurrence of this

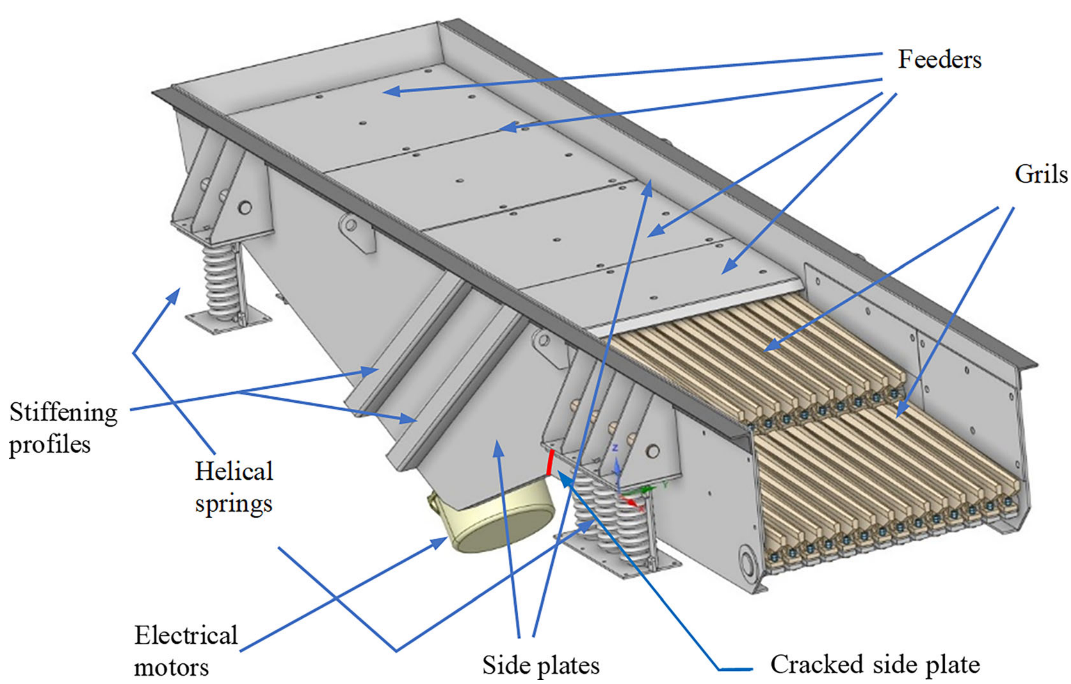

Fig. 1 A vibrating feeder and its main sections

Fig. 2 Fatigue-cracked section of the vibrating feeder

type of failure. The fatigue crack propagated after about 2 days of operation at a feeder frequency of 15 Hz with a utilization factor of approximately 40%. Thus, the total number of cycles to failure was about 1.04×10^6.

The robustness of vibratory equipment structures is of paramount importance as they undergo significant dynamic forces. Improper installation, misuse, as well as inadequate design can lead to premature failures. The occurrence of metal fatigue, resulting from continuous dynamic loading and eventually leading to cracking and complete failure under high stresses, necessitates a thorough understanding of the vibratory system and the loads it transfers to its supporting structure. Specifically, in the case of a cantilevered vibratory conveyor, the forces exerted pose a significant challenge, demanding an exceptionally sturdy support structure capable of withstanding dynamic loads that surpass the static load [16].

The specifications of the feeder are given in Table 1.

The predominant methodology in designing a vibrating feeder is to model the feeder using FEM. By conducting feeder modeling within a FEM environment, it becomes possible to manipulate parameters without necessitating the production of a new feeder for each design iteration. Creating a FEM model represents an economical and straightforward substitute for adjusting parameters on an actual feeder. Developing a FEM model for the feeder enhances material processing efficiency and simulates faults in vibrating feeders. Static analysis, modal analysis, harmonic analysis, transient dynamic analysis, spectrum analysis, buckling analysis, and explicit dynamic analysis are some of the several FEA analysis types that are now employed [13, 17, 18].

Table 1 Technical characteristics of vibrating feeder

Specification	Value
Feeder total weight (kN)	79.56
Feeder length (m)	6.078
Feeder width (m)	1.949
Feeder height (m)	1.67
Feeder capacity (kN/h)	10
Feeder material density (kg/m^3)	7850
Feeder material yield strength (MPa)	250
Feeder material ultimate strength (MPa)	460
Feeder material poisson's ratio	0.3
Feeder material elastic modulus (GPa)	210

Numerical Analysis on the Original Feeder

In order to reduce the number of mesh and nodes, simplifications based on the elimination of welds, bolt holes, nuts, washers, fillets, radii, and some other details are removed for geometry cleaning, minimizing the factors that may create some problems during the meshing process as they have a minor effect on the analysis.

The FEM analysis was conducted to obtain the vibrating feeder's vibration modes, natural frequencies, and stress characteristics under harmonically forced vibrations. Based on this analysis, the stress and fatigue performance of the vibrating feeder can be judged, and if necessary, design improvements can be made.

For the sake of simplicity, for the FEM analysis, all the materials for the vibrating feeder structure were considered to be structural steel. All the analyses of the study were considered as elastic analysis. Each helical spring used in the real physical feeder has a 171 mm outer diameter, 32 mm wire diameter, and 378 N/mm stiffness (spring constant). All the feeder springs were considered cylinders with the spring constants, as shown in Fig. 3. This is mainly due to the fact that, as pointed out elsewhere [19], springs can be regarded as hollow cylinders if the structural significance of the spring is lost. Furthermore, the complex structure of the springs is one of the problematic factors affecting the meshing process. Another simplification introduced in the analysis is the electrical motors. Thus, the two motors were replaced as two masses in the analysis. The feeder was fixed at its bottom legs. All the contacts of about 750 between the elements were checked one by one, and a suitable contact type for each contact was defined. Solid body contact was used in all the contacts and no stabilization was used. Forces of 1000 N of equally distributed load were applied on each grill in front of the feeder grids. A force of 71.43 N loads was also applied uniformly on each grid. After that, the meshing of the feeder was done. Altogether, 114,528 nodes with 128,459 elements were used in the FEM analysis, as shown in Fig. 3. Solid-type elements with the linear meshing methodology used in the analysis and the number of meshes noted were calculated by the program accordingly. Additionally, radiuses surrounding apertures required attention to ensure a robust meshing process. While these radiuses were successfully addressed, deliberate omission of closing the apertures was necessary due to their functional significance in feeder operations. Closure of these apertures would induce linearity in the meshing process. Consequently, meshing in regions devoid of nearby apertures proceeded linearly. Conversely, meshing in regions proximal to the apertures did not adhere to linearity. Despite the visual semblance of tetrahedron in the resulting mesh from a distance, it predominantly comprised linear elements. Local deviations from linearity prompted the emergence of tetrahedral elements in certain areas.

Various methods are employed to increase the efficiency of vibrating feeders, such as reducing energy consumption, improving productivity, minimizing downtime, and streamlining maintenance. In a study conducted by Yue-Min and his co-workers. [20], the reliability of vibrating feeders was enhanced by optimizing the dimensions of the stiffeners on the side plate while considering multiple frequency constraints. This optimization process led to a reduction in the weight of the side plates and an increase in natural frequencies, effectively moving them further outside of the operating frequency. By reducing the feeder's

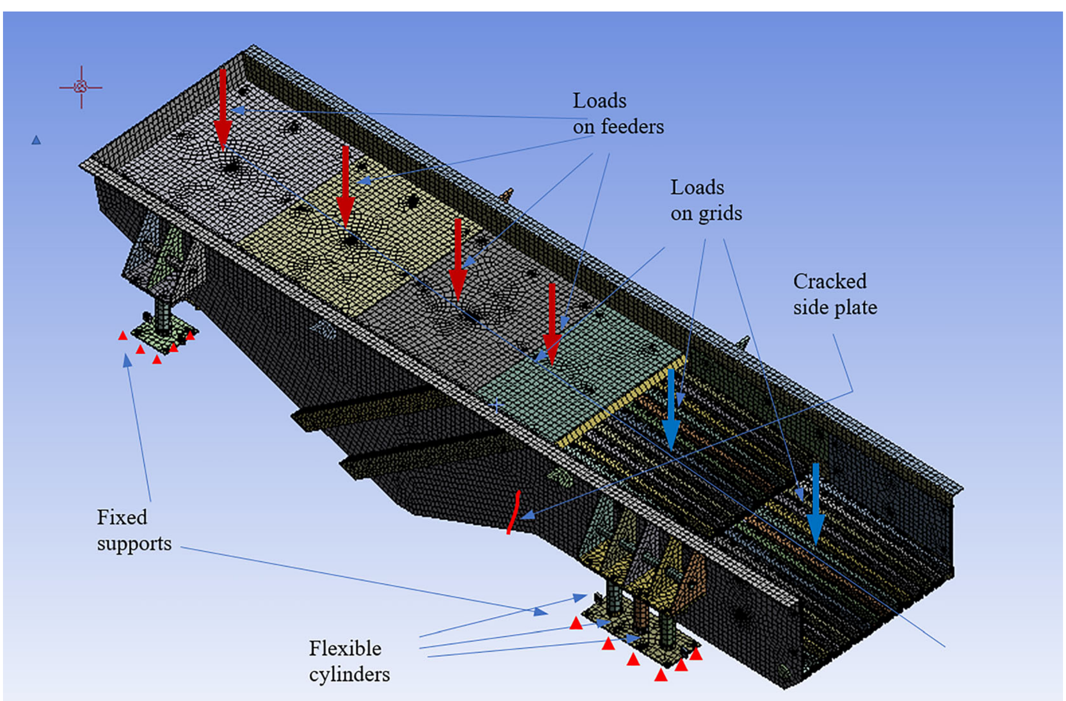

Fig. 3 CAD and finite element models used in the analysis

weight, manufacturing costs were reduced. Moreover, relocating the natural frequency outside the operating frequency improved resonance avoidance, mitigating potential feeder damage.

Total deformation, equivalent elastic strain, and equivalent stress distribution throughout the system were the three analyses performed using ANSYS.

Static Stress Analyses

In order to get an overall view of the stress distribution and weaknesses in the vibration feeder under static loads, the FEM analysis was conducted. As the main concern is the magnitude of stresses at the fatigue cracked region, the maximum stressed points at other locations of the feeder were ignored. Figure 4 shows the equivalent (von-Mises) stress distribution. The analysis reveals that the highest equivalent (von-Mises) stress occurs at the region where the fatigue crack was initiated with a value of is 37.88 MPa, much less than the material's yield stress. As the material's yield stress is 250 MPa, the vibrating feeder is safe for the statically applied loads with a factor of safety (FOS) of 250/37.88 = 6.6. There is a structure working under such dynamic conditions and repeated loading, such as vibrating feeders; this FOS still needs to meet the long-term operating design requirements.

Vibration and Modal Analysis

After creating the FEM model of the feeder, a modal analysis of the structure is carried out to determine the natural frequency and vibration mode, as the vibrating feeder structure will generate a certain amount of vibration under normal steady-state operating conditions. Operating the feeder outside of these natural frequencies will help the external excitation to avoid its natural frequency, prevent the resonance phenomenon, and ensure stable equipment operation. The structure's critical frequencies and mode shapes are obtained and compared through this analysis, and the structure's best operating frequency ranges are evaluated.

As the vibrating feeders work under dynamic loading conditions at certain frequencies, studying the feeder's response under repeated loadings would be essential at the design stage. As the resonance occurs, the stresses may exceed the material yield stress, causing catastrophic failure. Thus, the modal analysis is used to detect failures in the structural components under the steady-state response of a structure to overcome resonance and fatigue-induced failures.

The results are presented in mode shapes with corresponding natural frequencies, as shown in Fig. 5.

The results of both vibration and mode shape results were used to validate the feeder design to ensure the safe

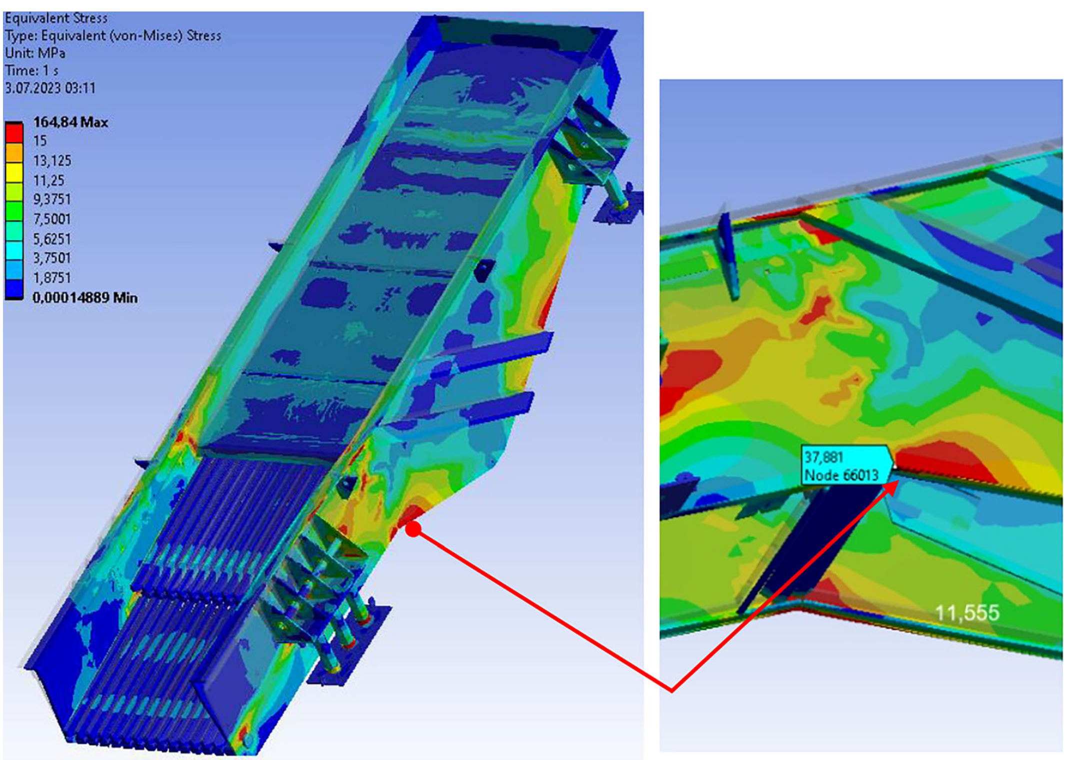

Fig. 4 von-Misses stress distribution in the feeder

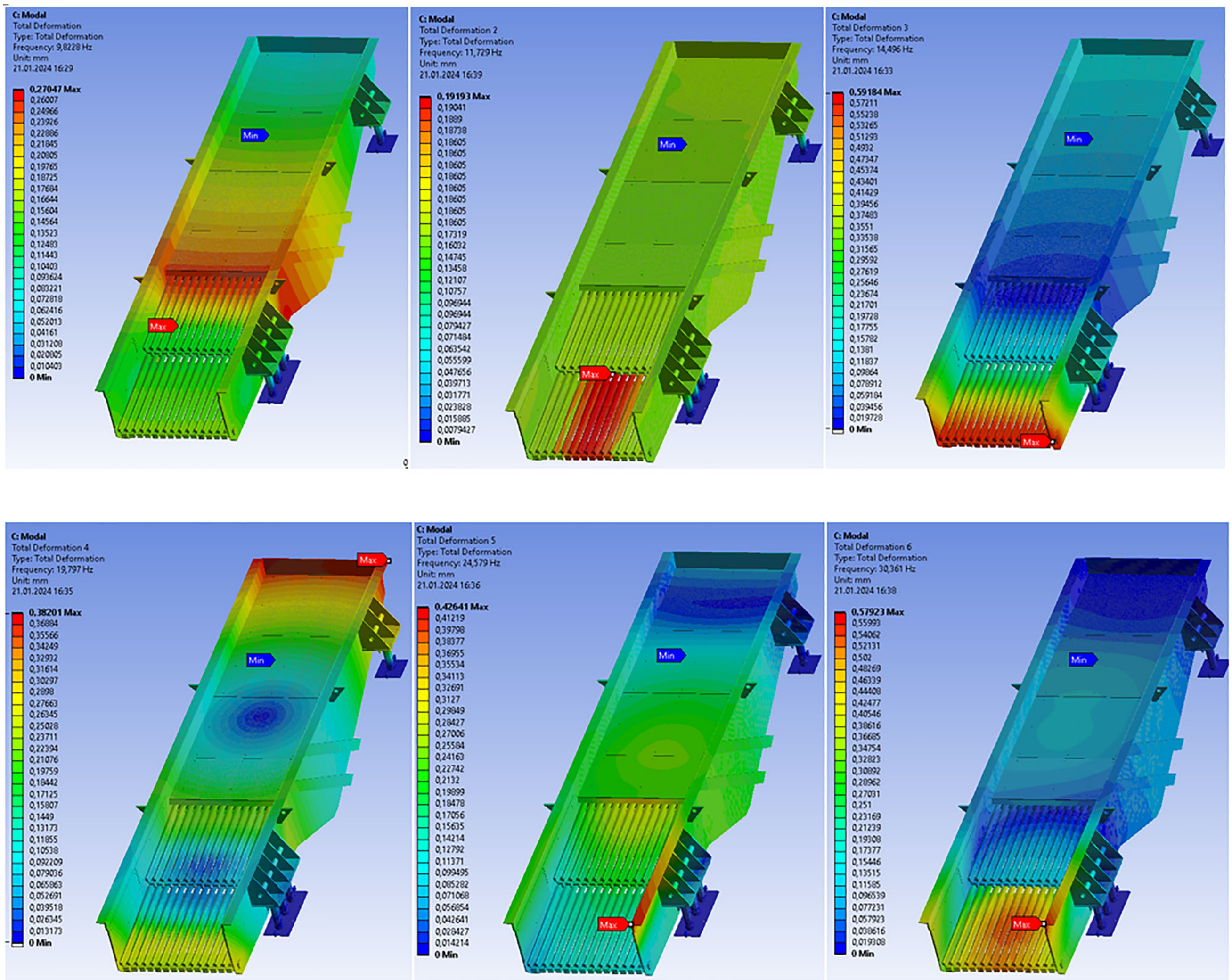

Fig. 5 First 6 mode shapes of the original feeder

and reliable operation of the equipment. Furthermore, these vibration and modal analysis results could be used to predict the static and dynamic structural behavior either at the design or rehabilitation stage.

Following the analysis, the critical frequency and associated deformations for the original feeder were investigated. As the feeder's normal operation frequency is 14 Hz, the frequency analyses were conducted to cover frequencies ranges between 1 and 50 Hz. In the frequency analyses, the same boundary conditions as in static analyses were used. The first 10 natural frequencies of the original feeder are tabulated in Table 2, and the natural frequencies vary between 9.82 and 43.97 Hz.

According to the analyses, the first 6 critical frequency values where the total deformations stand out the most are shown in Fig. 5.

The total deformation values for every mode shape differ from each other, and the maximum deformation for

Table 2 First 10 critical frequencies of the original and modified feeders

	Critical frequencies (Hz)	
Mode shapes	First feeder	Second feeder
1	9.82	10.31
2	11.73	11.06
3	14.50	14.43
4	19.79	21.42
5	24.58	26.12
6	30.36	32.84
7	36.14	40.16
8	37.60	41.90
9	40.56	46.63
10	43.97	47.66

every mode shape varies between 0.192 mm (mode shape 2) and 0.657 mm (mode shape 10).

Fatigue Analysis

The aim of fatigue analysis is to characterize the ability of a material to withstand the many cycles that a structure may experience over its lifetime. The calculations and results of this analysis allow engineers to evaluate their designs to avoid failures under real-world conditions.

The fatigue analysis is performed based on linear static analysis results, and the total life of the feeders and damage caused due to cyclic loads are determined. Material properties considered in fatigue analyses were the same as given in Table 1. The fatigue data at zero mean stress come from 1998 ASME BPV Cade, section 8, Div. 2, Table 5-110.1. The fatigue analysis was calculated based on the Goodman approach. The fatigue strength factor considered in the analysis is set to 1.

This section presents the results of the life, damage, and safety factor analyses. The result of the life analysis of the original feeder is shown in Fig. 6, showing that the most critical region of the life of the feeder is 1 x 10^6 cycles, pointing to the location where the fatigue crack was initiated in real-life working conditions.

As Fig. 6 shows, the minimum calculated fatigue life overlaps with the location of the origin of physical fatigue crack origin. The result of the Factor of Safety (FOS) analysis damage for the whole feeder is given in Fig. 7, showing that at the cracked region, the FOS drops to a value of 3.83.

Design Improvements and Reanalysis Results

Considering the physical nature of the failure and the FEM analysis results in the existing vibrating feeder, the following design improvements and modifications were identified and implemented on the existing feeder. The side plate thicknesses were increased from 12 to 15 mm, the number of side plate stiffening profiles was increased from 2 to 4, and the stiffening profile thickness was increased from 8 to 10 mm. Furthermore, the inner support pipe diameter under the screen was increased from 89 to 219 mm. The radius of the side plate where the fatigue crack originated was increased from 150 to 1000 mm to reduce the stress concentration.

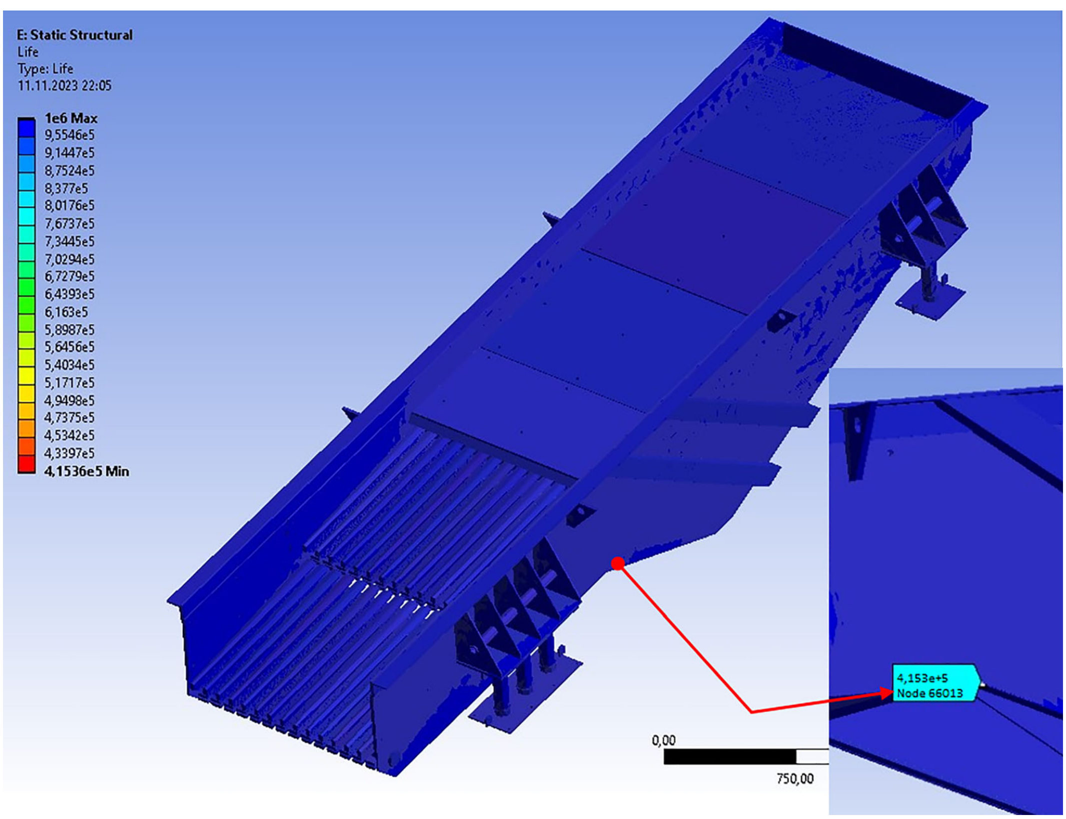

Fig. 6 Fatigue life prediction of the original feeder

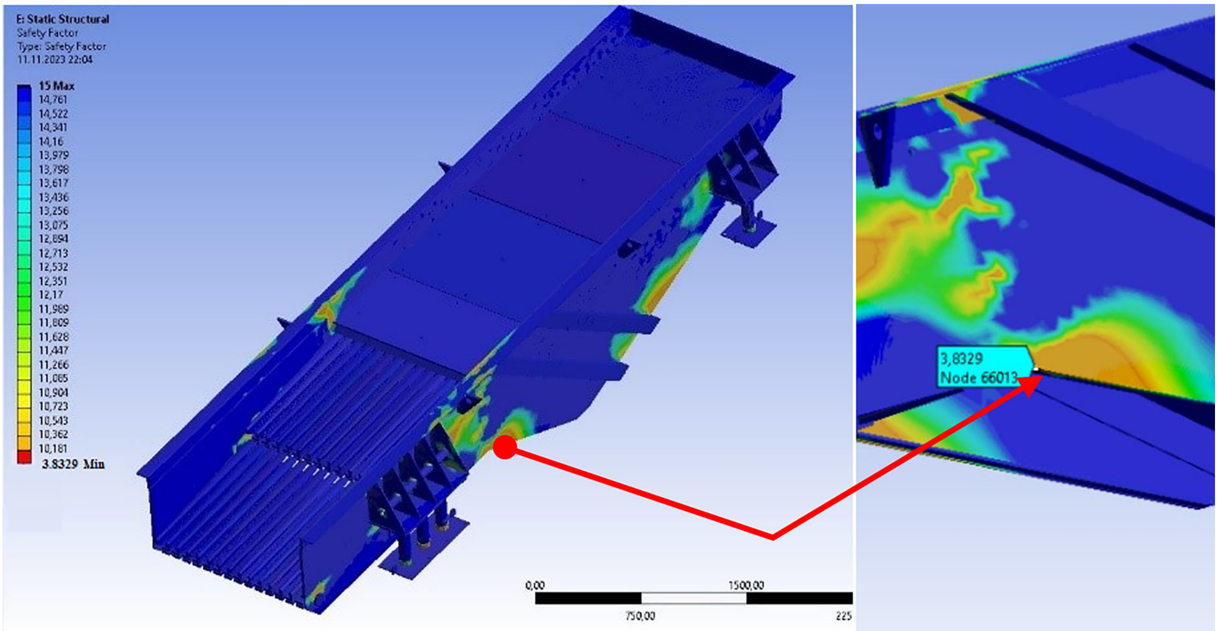

Fig. 7 Factor of safety distribution on the whole original feeder

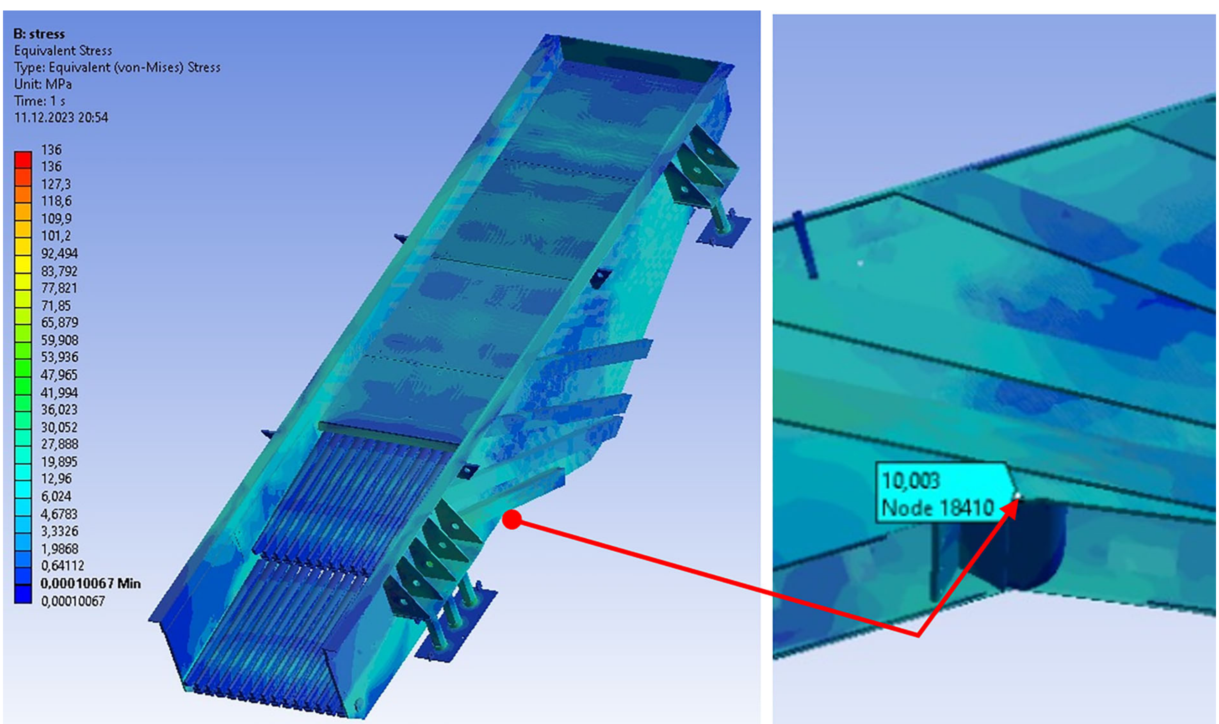

Fig. 8 von-Misses stress distribution in the modified feeder

After a period of use, the initial feeder experienced breakage of its side plates, rendering it inoperable. As a result, a modified feeder design was developed, with the primary distinguishing feature being the supports in their side views.

Static Stress Analyses for Modified Feeder

Figure 8 shows the equivalent (von-Mises) stress analysis result. As a result of the analysis, the equivalent (von-Mises) stress at the cracked region is about 10 MPa. The

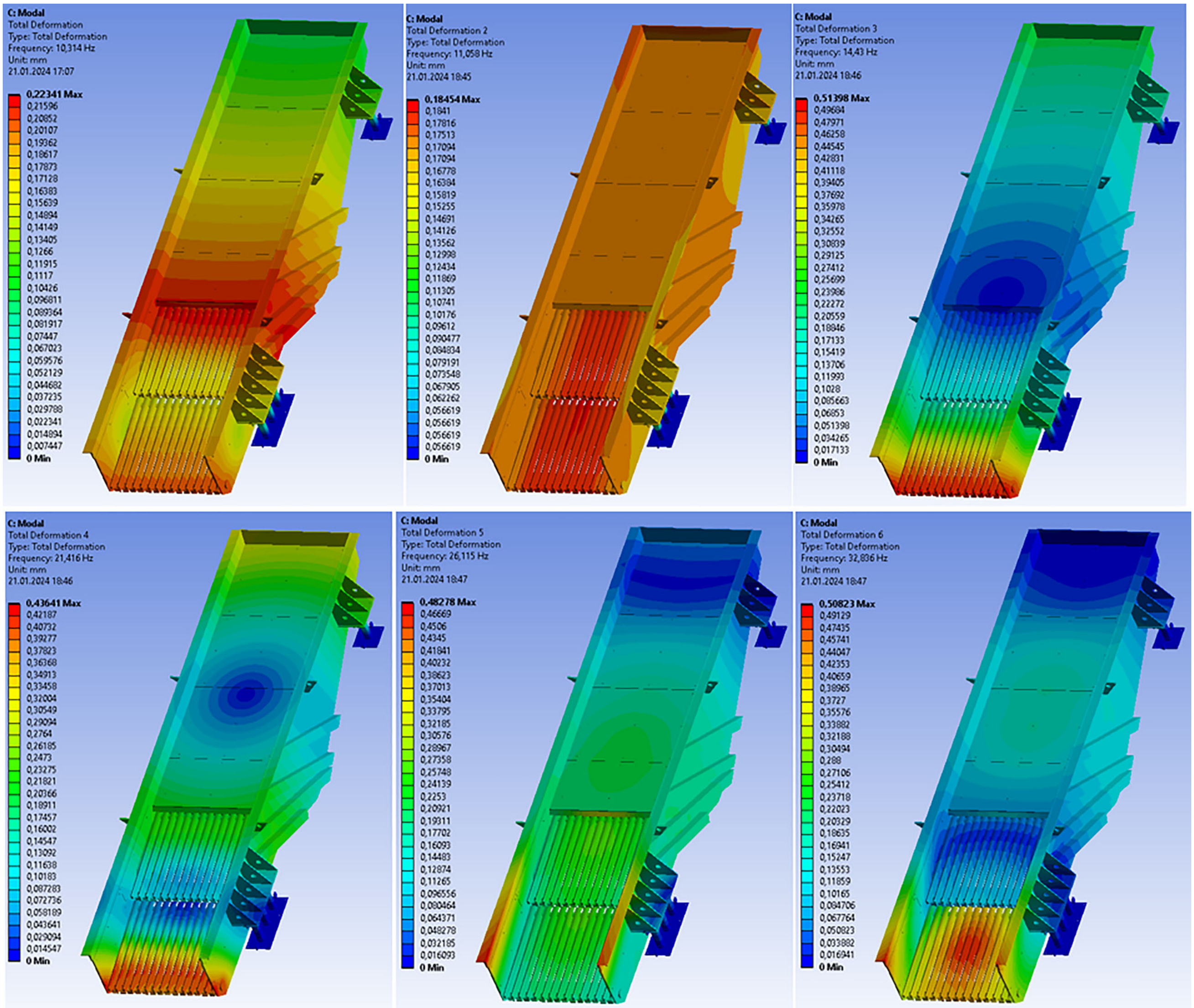

Fig. 9 First 6 mode shapes of the modified feeder

higher stresses on the color scale are the values at the inner locations between the spring cylinders and the base plates (not shown in the figure).

The static stress analysis on the modified feeder reveals that the highest equivalent (von-Mises) stress decreased from 37.88 MPa to 10.0 MPa, which is significantly less than the yield stress of the material. As the material's yield stress is 250 MPa, the vibrating feeder is safe for the statically applied loads with a factor of safety (FOS) of 250/10.0 = 25.0. The analysis shows that the design improvements are beneficial and have reduced the equivalent stress by almost 3.8 times, from 37.88 MPa to 10.0 MPa.

Vibration and Modal Analysis for Modified Feeder

The first 10 natural frequencies of the original and modified feeders are tabulated in Table 2, showing that the natural frequencies of the modified feeder are only about 6% higher than those of the original feeder.

According to the analyses, the first 6 critical frequency values where the total deformations stand out the most are shown in Fig. 9.

The total deformation values for each mode shape of the modified feeder differ from each other, and the maximum deformation for each mode shape varies between 0.185 mm (mode shape 2) and 0.609 mm (mode shape 8). Furthermore, a comparison of Table 2 and Figs. 5 and 9 shows that, despite certain similarities, each design of these two feeders has unique intrinsic vibration responses.

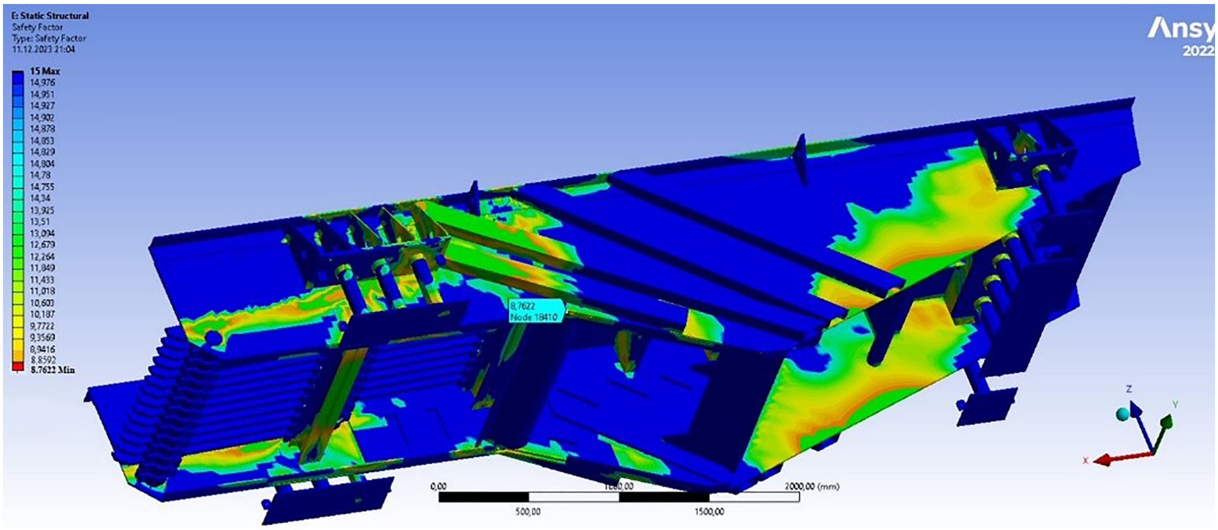

Fig. 10 Factor of safety distribution on the whole modified feeder

Fatigue Analysis of Modified Feeder

The result of the fatigue life analysis of the modified feeder is calculated to be 1.04. $\times$ 10^9 cycles, pointing to the location where the fatigue crack was initiated in real-life working conditions.

The result of the Factor of Safety (FOS) analysis for the entire feeder is shown in Fig. 10, showing that the FOS drops to its minimum value of 8.76 at the cracked area.

The maximum von-Misses decreased by approximately 3.8 times as a result of the design improvements in the modified feeder, while the FOS at the cracked region increased from 3.83 to 8.76, indicating the effectiveness of the design.

Discussion of the Results

In this study, three important analyses were conducted on the vibrating feeder: static stress, vibration, and fatigue analyses. These analyses must be highly accurate and error-free to ensure their practicality and applicability in the field.

Regarding the static equivalent (von-Mises) stress analysis of both feeders, the original feeder had a maximum stress value of 37.88 MPa. This value was reduced to 10.0 MPa for the modified feeder, a reduction of approximately 3.8 times. Evaluation of the equivalent stress analysis of the original feeder revealed significantly larger equivalent stress regions than those of the modified feeder. Based on the failure analysis and FEM analysis, the implemented design modifications to the feeder effectively reduced the equivalent stress in the modified feeder.

Following the static stress analysis, the vibration analysis was carried out, which produced three similar results to the static stress analysis.

During the vibration analysis, it was observed that the original feeder had a maximum deformation of 0.657 mm. Conversely, the modified feeder had a maximum deformation of 0.607 mm. Despite the prevalence of vibration in both feeders, certain regions and areas showed no differentiation, as shown in Figs. 5 and 9.

As pointed out elsewhere [21], the accuracy of the calculation could be improved by coupling the vibration of the feeder to the materials being conveyed and their individual geometries. Furthermore, all the analyses are based on linear elastic analysis, and thus, all possible nonlinearities resulting from contacts and material responses were ignored. Moreover, it is assumed that the loads acting on the feeder and grill are uniformly distributed, which is not valid in real-life operating conditions.

Conclusions

The results of the present study confirm that the static and dynamic behavior of a heavy-duty industrial vibrating feeder can be investigated using FEM. The stress distribution of the vibration feeder was investigated, and the critical fatigue location was predicted. The FEM results indicate that the stress levels and fatigue life predictions were correct, as the original feeder failed at the side plate where the numerical simulation showed the highest equivalent stress level. The analysis results can be used at the design stage to diagnose potential failures and their locations. Taking into account the FEM results, several design improvements were made to prevent the recurrence of the failure in the original feeder. Both numerical analysis and the full-field working model of the revised vibrating feeder verified the adequacy and soundness of the design changes implemented. The improved design has much higher structural strength with improved static and dynamic behavior. The methodology presented here can be used to analyze and prevent failures in engineering structures.

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Conflict of interest The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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