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Experimental Modal Test and Analysis for Auxiliary Beam of Sand Blender

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ABSTRACT

The auxiliary beam of sand blender had a frame structure which was welded by profile steel, the structural feature of which determined that the natural frequencies along the vertical direction were fairly low. During the operation of sand blender, the possible equality between working frequencies of on-board devices and natural frequencies of the auxiliary beam could cause a strong vibration, not only influencing the normal fracturing, but also leading to structural damage of the auxiliary beam, jeopardizing the safety of the sand blender. In this study, the experimental modal test of the auxiliary beam was conducted by using the method of single point pulsing and multiple points receiving. The main goal of the study was to determine basic dynamic characteristics of the auxiliary beam. The working frequencies of on-board devices were calculated and the relationship between which and natural frequencies of the auxiliary beam was analyzed. The study results would provide reliable basis for modifying the structure of the auxiliary beam and adjusting the operating parameters of on-board devices aimed to reduce the vibration.

Key words: Auxiliary beam, experimental modal test, natural frequency, working frequency, coherency function

INTRODUCTION

Acting as one of the core devices of fracturing equipment group, the function of sand blender was to mix, stir and deliver the high pressure fracturing fluid. The structure character of the sand blender was that the on-board devices including mixing tank, reciprocating pump and water tank were installed on the auxiliary beam while the auxiliary beam was fixed on the main beam of a heavy truck. During the operation of the sand blender, due to the periodic motion of the motors, reciprocating pump and mixing propeller, the auxiliary beam might develop a serious periodical vibration which had an adverse effect on normal operation of sand blender, moreover, the periodical vibration could result in microcracks in the auxiliary beam, putting great hidden troubles to fracturing work and endangering the safety of on-board devices. To avoid this situation, it is necessary to master the dynamic characteristics of the auxiliary beam for its possibly needed structure modification and for the proper setting of working parameters of sand blender.

At present, the study on the dynamic characteristics of the auxiliary beam is seldom, but modal analysis provides us an effective way to study mechanical dynamics (Zhu *et al.*, 2009). There are two methods of modal analysis: Computational modal analysis and experimental modal analysis. The former one mainly adopts CAE software and finite element method to calculate the structure modal parameters (Wang *et al.*, 2008; Eritenel and Parker, 2009; Kiracofe and Parker, 2007) while the later one calculates the structure modal parameters by indentifying input and output signals obtained by test devices (Guan *et al.*, 2005; Takatsu, 1991; Liu *et al.*, 2009). Comparing to computational modal analysis, experimental modal analysis has higher accuracy, it has already widely used in the fields of car, bridge and petroleum equipments, etc. In this study, based on related modal test theory, the experimental modal test for the auxiliary beam of sand blender was carried out by using uTekl dynamic signal analysis system, the testing data were analyzed, the former six order natural frequencies and vibration modes were obtained, then comparision between natural frequencies of the auxiliary beam and working frequencies of on-board devices were made, the vibration mechanism of the auxiliary was founded which could work as the basis of structure modification of the auxiliary beam and proper setting of working parameters of the sand blender.

EXPERIMENTAL MODAL TEST OF AUXILIARY BEAM

Theorem of experimental modal test: The vibration of auxiliary beam can be assumed as the movement of a linear elastic physical system with n degree of freedom whose vibration differential equation is as follows (Fu and Hua, 2000):

$$[M]\{\ddot{x}(t)\}+[C]\{\dot{x}(t)\}+[K]\{x(t)\}=\{f(t)\} \tag{1}$$

where, [M] is mass matrix, [C] is damping matrix, [K] is Rigidity matrix, [M], [C], [K] are real square matrixes with n factorials; {x(t)} is displacement vector $\{\dot{x}(t)\}$ is velocity vector, $\{\ddot{x}(t)\}$ is acceleration vector {x(t), $\{\dot{x}(t)\}$ and $\{\ddot{x}(t)\}$ are n dimensional vectors and {f(t)} is exciting force vector with n dimensions.

By applying Fourier transform to Eq. 1, we get following equation:

$$(-\omega^2[M]+i\omega[C]+[K])\{X(\omega)\}=\{F(\omega)\} \tag{2}$$

where, ω is natural frequencies of vibration system, i is unit of imaginary component; by setting $H[\omega] = [-\omega^2[M]+i\omega[C]+[K]]^{-1}$ and placing it into Eq. 2, it is obtained that:

$$\{X(\omega)\}=H[\omega]\{F(\omega)\} \tag{3}$$

where, H[ω] is frequency response matrix.

If we excite the vibration system at point P and pick up the response signal at point L, then the transfer function between point P and L can be expressed as (Chen *et al.*, 2007):

$$H_{LP}(\omega)=\sum_{i=1}^{n}\frac{\Phi_{Li}\Phi_{Pi}}{-\omega^2M_i+j\omega C_i+K_i} \tag{4}$$

where, M_i is modal mass of order i, C_i is modal damping of order i, K_i is modal rigidity of order i, Φ_{Li}, Φ_{Pi} are vibration modes of order i at point L and P.

According to dynamic Betti-Rayleigh Theorem (Achenbach, 1984), the whole frequency response matrix can be obtained by testing just one column or one row of the transfer function. In real test, the input force signal as well as its response signal can be tested and identified by testing system. Then the auto power spectrum of both force pulse signal and response signal and the cross-power spectrum between the two signals can be calculated.

Thus we have the frequency response matrix as following:

$$H(\omega) = \frac{G_{yy}(\omega)}{G_{xx}(\omega)} \tag{5}$$

where, G_{yy} is auto power spectrum of response signal, G_{xx} is auto power spectrum of force pulse signal.

Coherency function $\gamma(\omega)$ is used to judge the confidence of the test whose expression is:

$$\gamma(\omega) = \frac{\left|G_{xy}(\omega)\right|^2}{G_{xx}(\omega) \cdot G_{yy}(\omega)} \tag{6}$$

where, Gxy is cross-power spectrum between input force signal and response signal. The closer the value of $\gamma(\omega)$ is to 1, the greater correlation between input signal and response signal will have which means the frequency response is more believable, so coherency function can be used to judge whether a certain exciting is reliable.

The method of averaging is often employed in order to reduce the influence caused by undesirable signal and increase the signal-to-noise ratio. In this test practice, repeating exciting at intervals was used. To avoid the interference between two pre and post signals, the time interval between two excitings should be large enough.

Model establishing for modal test of auxiliary beam: Figure 1 was the 3-D model of the auxiliary beam established in SolidWorks software. As can be seen in Fig. 1, overall, the auxiliary beam was welded by profile steel, forming a frame structure whose length, width and

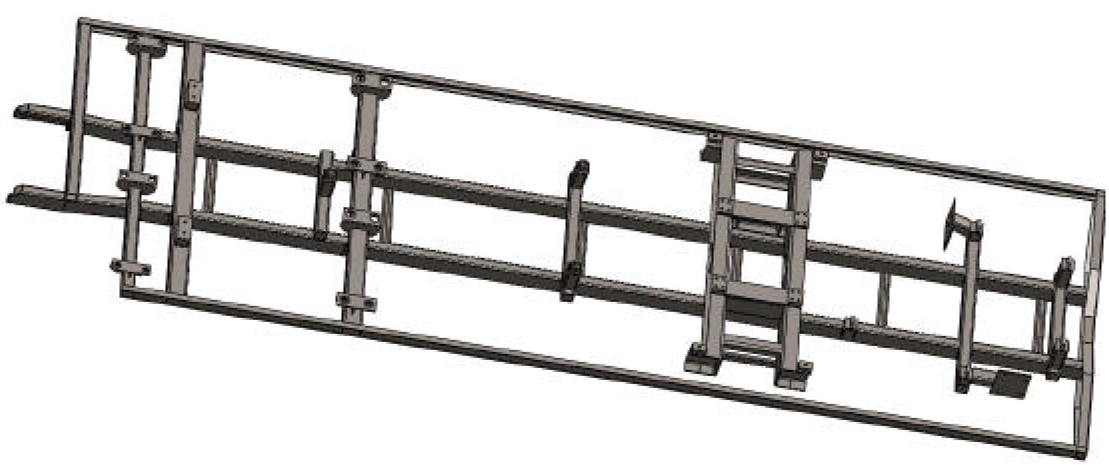

Fig. 1: 3-D model of auxiliary beam

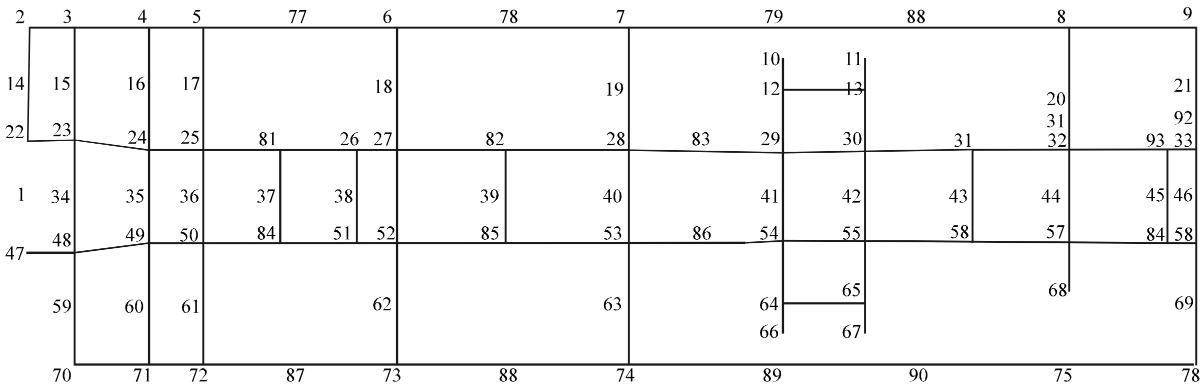

Fig. 2: Node-line model of the auxiliary beam

height were respectively 8.77, 2.48 and 0.58 m. The structure along the length was constructed by four sections with 2 rectangular steel tubes on the center and 1 channel steels on each side. The upper structure along the width consisted of 9 channel steels and rectangular steel tubes welded on the structure along the length which was functioned as the platform for on-board devices. The lower structure of along the width was formed by welding 7 steel plates on the structure along the length which was used to connect the auxiliary beam and the main beam of the truck. Also there were dozens of mounting bases and lugs welded on the auxiliary beam.

The auxiliary beam was a continuous elastomer. When it vibrates under the external stimulation, displacement will occur between any two points. Thus as in finite element analysis, in experimental modal test, the continuous elastomer should be simplified as a structure in which some neighbouring nodes have geometric relationship. By connecting all neighbouring nodes with lines, the continuous elastomer was turned into a node-line model.

Because harmful vibration was mainly along the height while the height was much less than the length and width, when establishing the node-line model, the height size was neglected, the auxiliary was therefore simplified as a 2-D model by changing the crossed length-width structure into the intersected one. Also some small structures such as mounting bases and lugs were neglected. The completed node-line model of the auxiliary beam was shown as Fig. 2 which consisted of 94 nodes. Here node 1 was not located on the entity of auxiliary beam, it was just a reference point used to define the coordinates.

Modal test method: As shown in Fig. 3, the modal testing system consisted of 1 signal acquisition unit, 1 multi-channel charge amplifier, 1 impact hammer, 1 piezoelectric force sensor, 15 ICP piezoelectric acceleration sensors, data lines, notebook computer and modal analysis software.

Since the purpose of modal test was to study the resonant frequency of the auxiliary beam, a pre-installed auxiliary beam was used to carry out the modal test, on which all the on-board devices were mounted and the auxiliary beam was fixed on the main beam of a trailer.

The method of "single point pulsing and multiple points receiving" was used to conduct the modal test. All test points on the auxiliary beam were marked with numbers which are the same as those on the node-line model. Based on the observation of the structure of the auxiliary beam, the exciting point was selected at the middle position on the rear of the auxiliary beam which was located at the opposition of point 46. When collecting the data, the exciting point was hammered vertically downward. Totally there were 16 channels on the signal acquisition unit, one of which

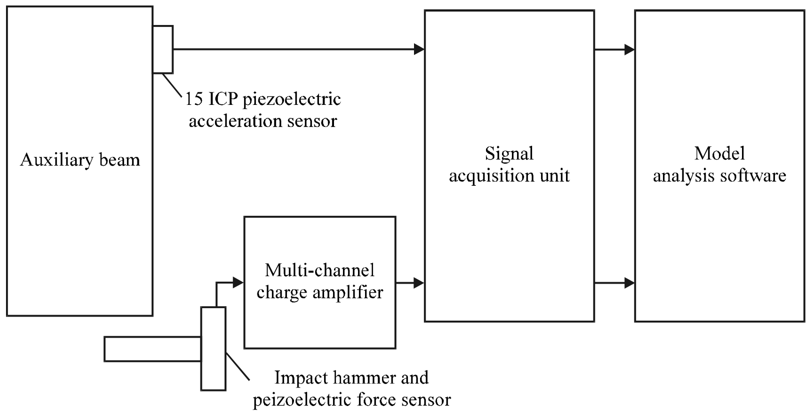

Fig. 3: Structure diagram of modal test system

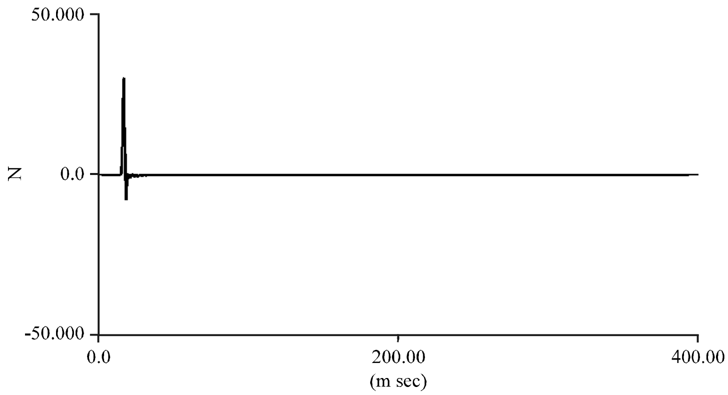

Fig. 4: Time history of exciting force signal of test point 76

was connected to the force hammer, thus at a time it was impossible to test all 93 points, at most 15 points could be tested. For this reason, batch test was employed to conduct the test, that was, the 93 test point were divided into several groups, after one group was tested, moving acceleration sensors to next group of test points, continue exciting and testing until response signals were picked up from all test points. Among all 93 test points, acceleration sensors could not be mounted on 14 points because of the block of on-board devices, so they could not be tested directly with acceleration sensors, constraint equations were used to obtain the vibration parameters of these points. The rest 79 test points were tested in 7 batches, each exciting point was hammered 4 times. To reduce the interference from external environment, the force window was used to process the force signal; to improve the signal-to-noise ratio, speed up the attenuation of vibration signal and avoid the leak of frequency response function, response signals were processed by rectangle window.

Data collection and analysis: By using the method introduced above, the input force signals and response signals of all test points were collected. For example, Fig. 4-7 were the curves of time history of exciting force signal, time history of response signal, auto power spectrum of response signal and coherency function between exciting force signal and response signal of test point 76.

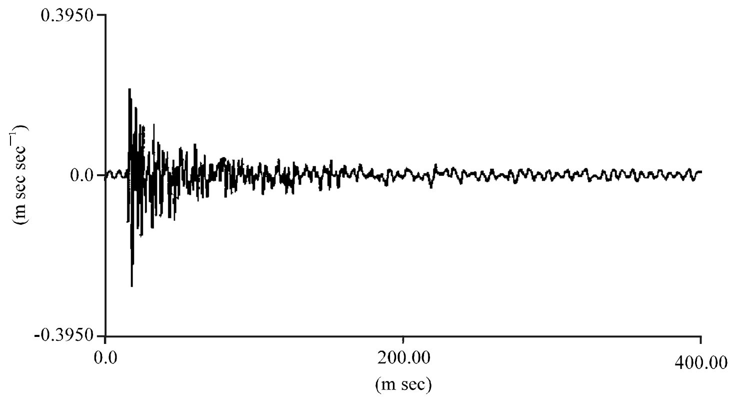

Fig. 5: Time history of response signal of test point 76

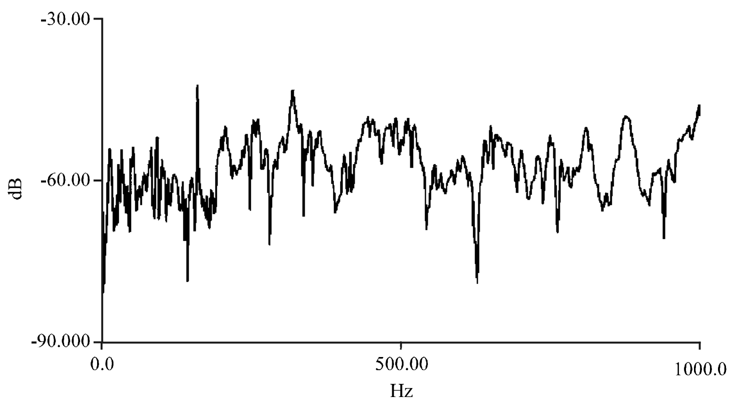

Fig. 6: Auto power spectrum of response signal of test point 76

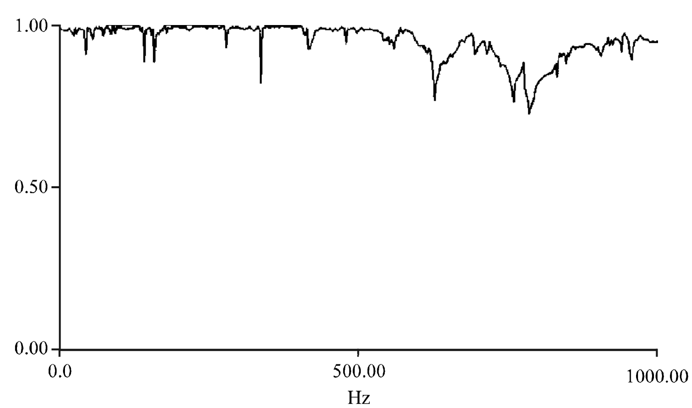

Fig. 7: Coherency function between exciting force signal and response signal of test point 76

As can be seen in Fig. 7, in the range of 0-1000 Hz, the coherency function values of most points were larger than 0.9 while in the range of 0-500 Hz, larger than 0.95 which strongly demonstrated that the test data was creditable.

Modal analysis was conducted after all points were tested. In this study, modal calculation was on the basis of real modal theory, modal type was selected as normal density, data fitting method

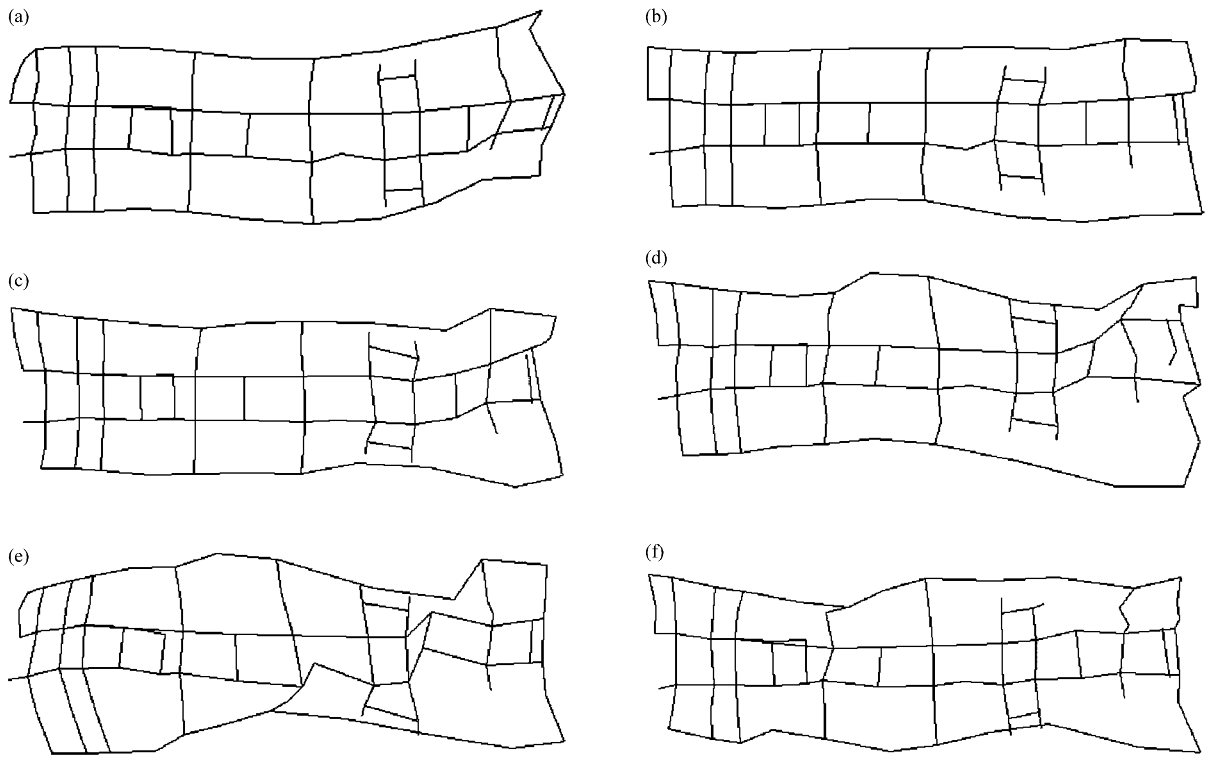

Fig. 8(a-f): (a) 1st order modal shape of the auxiliary beam, (b) 2nd order modal shape of the auxiliary beam, (c) 3rd order modal shape of the auxiliary beam, (d) 4th order modal shape of the auxiliary beam, (e) 5th order modal shape of the auxiliary beam and (f) 6th order modal shape of the auxiliary beam

Table 1: Condition of the experiment

Order	Frequency (Hz)	Damping ratio (%)
1	12.45	3.38
2	27.51	2.65
3	34.22	1.76
4	48.60	2.23
5	52.58	0.97
6	72.33	1.34

was integral fitting. According to the peak values of ensemble average curve of frequency response function, the initial estimated values of modal frequencies were selected and fitted, then the completed modal parameters were obtained by integrated processing of vibration modes, including processing of measuring direction, processing of constraint equations and unitary processing of modal shape by maximum freedom degrees. The first six order modal shapes of the auxiliary beam were shown in Fig. 8. The related modal frequency and damping ratio were listed in Table 1.

By analyzing the first 6 order vibration modal shapes of auxiliary beam shown in Fig. 8 and Table 1, it could be found that the first order modal shape was mainly bending vibration with natural frequency was 12.45 Hz while the second to six order modal shapes were mainly torsional vibration accompanied by local vibration with natural frequencies from 27.51 to 72.33 Hz. The

Table 2: Working frequencies of truck mounted equipments

Device name	Rotation speed (rpm)	Working frequency (Hz)
Chassis-mounted engine	1500	25.00
Onboard engine	2100	35.00
Clean water centrifugal pump	2900	48.33
Circulating centrifugal pump	2100	35.00
Charge centrifugal pump	2100	35.00

bending vibration frequency was smaller than the torsional vibration frequency, that was because the length of the auxiliary beam was much larger than the width, the rigidity along the length was smaller than that along the width.

From the damping ratio listed in Table 1, it was found that the damping ratios of all first 6 order vibration modals were between 0.97~3.38% which were small values and near to that of passenger car with similar frame structure (Cai *et al*., 2007), thus the result of experimental modal test was proved to be reliable.

The working frequencies of on-board devices were listed in Table 2. By contrasting the related data of Table 1 and 2, we found the rotation speed of onboard engine, circulating centrifugal pump and charge centrifugal pump was 2100 rpm, the corresponding working frequency was 35 Hz which was near to the third modal frequency (34.22 Hz) of auxiliary beam, also, the rotation speed of clean water centrifugal pump was 2900 rpm with working frequency was 48.33 Hz which was near to the fourth modal frequency (48.60) of auxiliary beam. Thus, during the operation of sand blender, the third and fourth modals of the auxiliary beam would be excited and serious vibration would be generated, therefore proper measures such as structural dynamic modification of the auxiliary beam and adjustment of the working condition of the sand blender and so on should be taken to avoid the vibration.

CONCLUSION

The structure of auxiliary beam of sand blender was analyzed in this study, the principles and method of modal experiment was explained, the experimental modal test for auxiliary beam was carried out, obtaining the first six order modal shapes and vibration frequencies, precisely demonstrating the dynamic characteristic of auxiliary beam.

The results of modal experiment showed that for the current structure of auxiliary beam and operating condition of sand blender, because the third and fourth modal frequencies were near to the working frequencies of onboard engine and clean water centrifugal pump, during the operation of the sand blender, the auxiliary beam would develop a serious vibration.

To avoid the serious structural vibration and possible fatigue failure, proper measures of structure optimization and operation parameters adjustment of truck mounted equipments should be taken. This study provided scientific references for such measures. Further work will focus on studying the method of structural dynamic modification and adjustment of working condition of the sand blender.

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