Applied Mechanics and Materials *Online: 2012-10-26*
ISSN: 1662-7482, Vols. 201-202, pp 175-180
doi:10.4028/www.scientific.net/AMM.201-202.175

Analysis of Dynamic Characteristics of Key Parts of Centrifugal Atomizer in Waste to Energy

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Keywords: Atomizer; Uneven. Mass; Temperature; Coupling; Dynamic characteristics

Abstract. Uneven distribution of mass and temperature usually have important effects on the dynamic characteristics of rotary mechanism. Based on thermal-structural coupling theory, dynamic response analyses of the flexible rotor of centrifugal atomizer can be done by FEM. Considering eccentric quantity is caused by nozzle clogging and axial non-uniform distribution of temperature, the dynamic characteristics of the is studied; the response of mass unbalance and thermal bending or coupled the both is researched and compared. It is shown from the calculation that all of the three conditions have affected the dynamic characteristics of rotor system and the design can basically meet the stable operation requirement of the atomizer. The results can be applied to improve the design of the centrifugal atomizer. Developing more detail analysis to optimize the design in the future research.

Introduction

Waste to energy is an effective way to dispose municipal solid waste harmlessly. The critical process of flue gas is centrifugal atomizer, in which the liquid Ca (OH) 2 can effectively neutralize the hazardous composition from flue gas. Therefore, the normal working of atomizer is related to the flue gas disposal with satisfied standard. In order to meet the process requirements of Ca (OH) 2 droplet sprayed from nozzles, the flexible rotor should run fast and smoothly. However, in the real working condition, there are two situations which would affect the dynamic characteristics of flexible rotor. One is the nozzle clogging which leads to the eccentric quantity, and the other is the temperature difference between the upper and lower sections caused by the difference between flue gas temperatures entered the desulfurization tower and the environment temperature. As the critical part of atomizer, understanding the dynamic characteristics of flexible rotor plays a very important role in the atomizer design.

Flexible rotor is a complex mechanical system, and the analysis process of its dynamic characteristics is the solving process for characteristic value and response problem of the system. The flexible rotor system usually runs at a high speed, which leads to gyroscopic moment. In addition to bearing damping and other reasons, the vibration analysis equation "Eq.1" is the following form [1, 2, 3, 4]:

$$\boldsymbol{M}\{\ddot{z}\}+(\boldsymbol{C}+\boldsymbol{G})\{\dot{z}\}+(\boldsymbol{K}+\boldsymbol{S})\{z\}=\{\boldsymbol{F}\} \tag{1}$$

In the equation, M is mass matrix of system; C is the damping matrix which is an unsymmetrical matrix damping matrix; G is a gyroscopic and also anti-symmetric matrix; K is the symmetric part of stiffness matrix; S is the unsymmetrical part of the stiffness matrix; {z} is generalized coordinates vector of system;{F} is generalized external excitation applied to the system. The characteristic equation "Eq.2" is shown as follows:

$$(\boldsymbol{K}-\omega^2\boldsymbol{M})\{z\}=0 \tag{2}$$

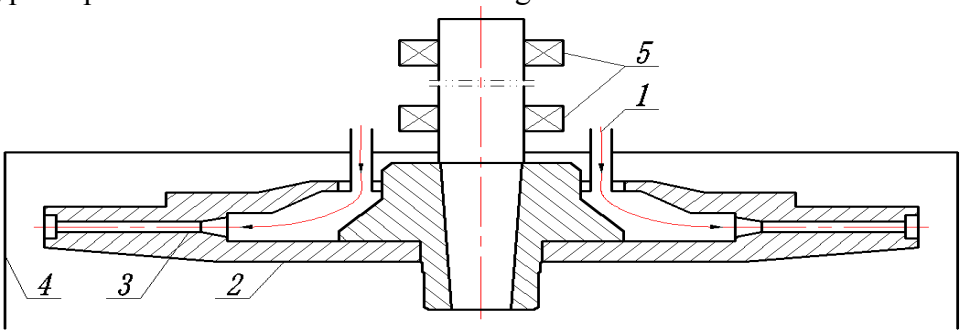
The solution is the i order natural angular frequency (i=1, 2, 3...n) [rad/s]; the corresponding i order nature frequencies is $f_{ni}=\omega_{ni}/2\pi$ [Hz]; the corresponding vector is {z} which is a vibration model, assuming that the structure vibrates with ωni.

At present, the common calculation methods of rotor dynamic characteristics are the transfer matrix method and the finite element method (FEM). The transfer matrix method takes less computer memory and calculates effectively, especially suitable for chain system like flexible rotor. However, this method cannot directly write into the equation of motion form, and it is necessary to simplify the rotor, bearing, bearing chock of actual rotor system into distributed parameter model composed of mass, damping and stiffness. In addition, in the computation of large rotor system, if the segmentation points are too many, the numerical instability phenomenon appears. In contrast, the FEM can build a complete and accurate calculation model of rotor system without the numerical instability phenomenon. Furthermore, FEM can directly write the independent motion equation for respective shaft segments and establish the mathematical model more easily [5, 6, 7, 8].

In this paper, the response of mass unbalance and thermal bending, or coupled the both are investigated with FEM in order to assure the working speed of atomizer within a safe range.

The Working Principle of the Atomizer

The high speed motor drives the main shaft, hence to drive the atomizer disk rotate at a high speed. The $Ca\ (OH)_2$ suspension liquid is sprayed through the special nozzle of the atomizer disk and atomized to micron scale droplets because of the strong centrifugal force. The surface area of the atomized droplets increases which is beneficial for the purification of hazardous substances in flue gas. The operation stability of atomizer is directly related to the quality of the flue gas purification. The working principle of the atomizer is shown as Fig. 1.



1. Entrance of lime slurry, 2.Atomizer disk, 3.Nozzles, 4. Desulfurization tower, 5.Bearing

Fig. 1 Working principle of atomizer

In the atomizer real working process, when reaching to some rotational speed, it will cause severe transverse bending vibration, even result in the rotor and bearing failure; and when the rotation speed is beyond the certain range, the rotor can run smoothly. This speed is called the critical speed of the rotor. The influence of gyroscopic effect must be considered when analyzing the critical speed of high rotational system as described before.

The Finite Element Model of Flexible Rotor

A lot of factors have impact on the dynamic characteristics of rotor systems as mentioned before. Therefore, in the modeling process, in addition to the design size, the simplification has also been carried out and the following principles has been established: establishment the complete models of rotating shaft and the atomizer disk, interference fit is adopted between the original connection, here it is considered as rigid connection ignoring the interference effects; the structure of some small parts such as screws are neglected, meanwhile the materials of those small parts are considered to be consistent with the main parts, some other small parts such as chamfer are also be neglected; bearing constraint is simplified as a elastic support in which the angular rigidity is neglected, only considering the radial stiffness, while the bearing load and speed on bearing stiffness effect are also ignored, the bearing stiffness is considered as a fixed value, realizing with the *MPC* plus spring unit constrained methods; The solid modeling will be employed. The quality unit will be applied to simulate the unbalance caused by clogging and the specific position is located in the central part of the nozzle.

According to the design size and the above simplified equations, the flexible rotor system model is established. The length of the shaft is 1.2[m] and the diameter of the shaft is from 0.05[m] to 0.08[m], the atomizer disk with 0.4[m] diameter is assembled on the shaft. The atomizer disk is hollow and 32 nozzles are distributed uniformly on the edge. The bearing positioned on the two ends is isotropic rolling bearings with shaft sleeves. The designed stable speed is 6000 [r/ min].

The hexahedral unit is applied for the calculation model in the HYPERMESH software. There are totally 43391units, the 50682 nodes, all of which are then loaded into the analysis calculation software. Fig. 1 shows the calculation model. The red component in this figure is the bearing location. The material parameters of the shaft are listed as follows: density ρ=7.82×103 [kg/m3]; material elastic modulus with temperature changes: 210[GPa] at 20[℃], 195[GPa] at 100[℃] and 185[GPa] at 300[℃]; thermal expansion coefficient α=1.25×10-5/℃, thermal conductivity k = 25.96[W/ (m·℃)]. The material parameters of atomizing disk are shown as follows: density ρ = 4.45×103 [kg/m3]; material elastic modulus with temperature changes: 113[GPa] at 20[℃] and 94[GPa] at 350[℃]; thermal expansion coefficient α = 0.93×10-5/℃, thermal conductivity coefficient k=10.47[W/(m·℃)]. The materials for bearings at both ends are the same: K=1.05×107[N/m]，C=2000 [N·s/m].

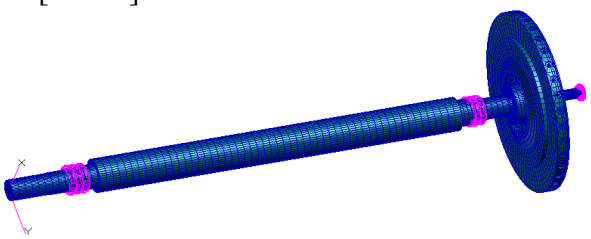


Fig. 2 Three-dimension graph of calculation model of rotor

Results

Mass Unbalance Response. Many factors can lead to the mass unbalance of atomizer rotor system, such as manufacturing, assembly and the error in the working process. In this paper, the previous factors are assumed “not exist”. The unbalance of mass distribution caused by the lime slurry nozzle clogging in the real working condition is considered here. The process requires high speed rotation so that the $Ca(OH)_2$ slurry is atomized and sprayed into desulfurization tower. However, the lime slurry itself is a kind of typical Non-Newton flow; atomizer is easy to be clogged in the long term use, which may induce the unbalance of rotor mass distribution. The vibration caused by the unbalance centrifugal force is directly related to speed. When frequency is same as speed, the dynamic characteristics of rotor will change.

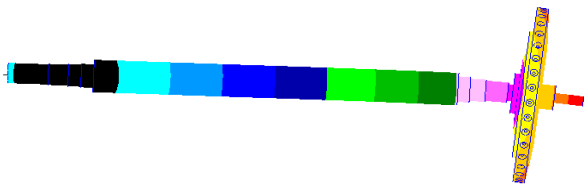


Fig. 3 The first order natural frequency is 84 *Hz* with mass unbalance considered

Fig. 4 The second order natural frequency is 135 *Hz* with mass unbalance considered

The vibration alarm device is designed for the atomizer, hence the special circumstance that only one of the nozzles is clogged is considered in this paper, namely a 0.0014[kg·m] unbalance mass is added to the atomizer disk. As shown from Fig. 3 to Fig. 5 are the first three order natural frequencies; if the initial start set as zero and the starting process takes 4 seconds, time domain response curve of node 392 on the disk edge near the eccentric quantity apply position on the starting process is shown as Fig. 6.

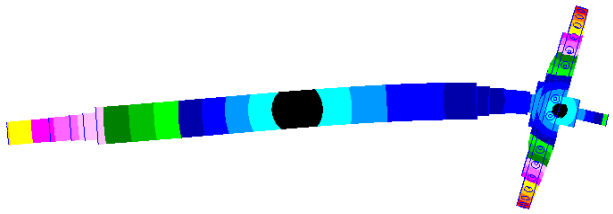


Fig. 5 The third order natural frequency is 310 *Hz* with mass unbalance considered

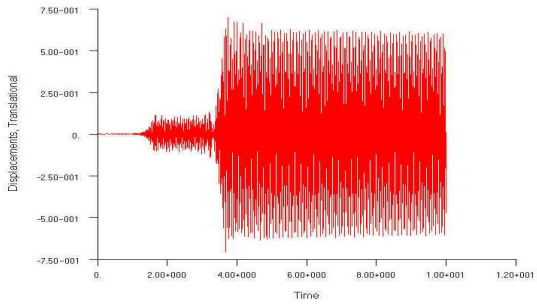


Fig. 6 Time domain response curve of starting process with mass unbalance considered

Non-uniform Distribution of Temperature Response Analysis. In a real working condition, the temperature of flue gas which entered desulfurization tower is different from the external environment temperature, which will result in heat stress, and produce a certain thermal deformation of the rotor system. The stiffness matrix of the rotor system is redistributed, affecting the vibration characteristics of the rotor. In addition, the material parameters (i.e., modulus of elasticity) of the rotor system change when temperature changes, which also affect to the vibration characteristics of the rotor. The limit condition of axial temperature difference 50[K] is considered in this paper. On the overall analysis, the first three order natural frequencies as shown from Fig. 7 to Fig. 9; the set is the same as previous, the time domain response curve of starting process for node 392 is shown in Fig. 10.

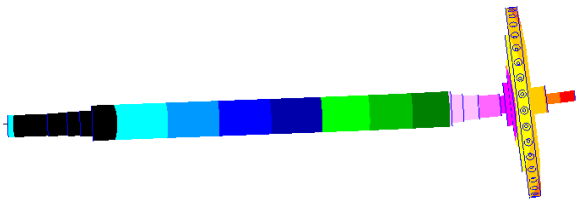


Fig.7 The first order natural frequency is 78 *Hz* with non-uniform distribution of temperature considered

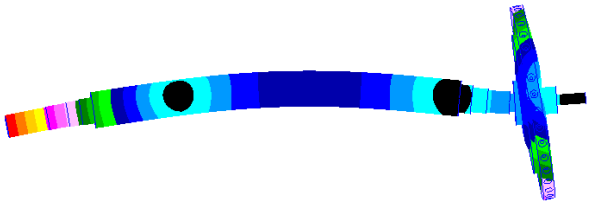


Fig.8 The second order natural frequency is 130*Hz* with non-uniform distribution of temperature considered

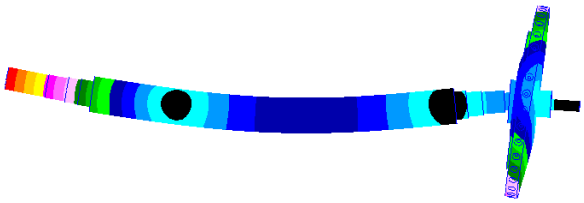


Fig. 9 The third order natural frequency is 310 Hz with non-uniform distribution of temperature considered

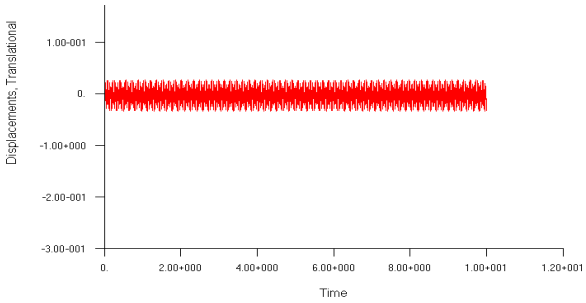


Fig. 10 Time domain response curve of starting process with non-uniform distribution of temperature considered

The Mass Unbalance and Temperature Coupled Response Analysis. In an actual working condition, both mass unbalance and axial temperature difference often exists at the same time, therefore, the critical speed of the coupled response, with temperature difference and mass unbalance considered simultaneously, is calculated. The first three natural frequencies are shown from Fig. 11 to Fig. 13; the set is the same as previous, the time domain response curve of starting process for node 392 is shown in Fig. 14.

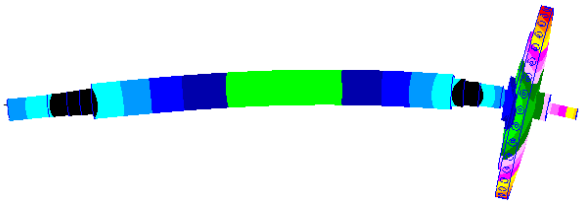


Fig. 11 The first natural order frequency is 67 *Hz* with coupling response considered

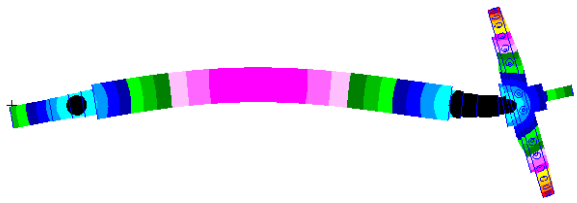


Fig. 12 The second order natural frequency is 121*Hz* with coupling response considered

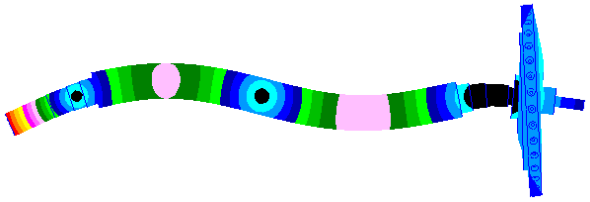


Fig. 13 The third order natural frequency is 295 *Hz* with coupling response considered

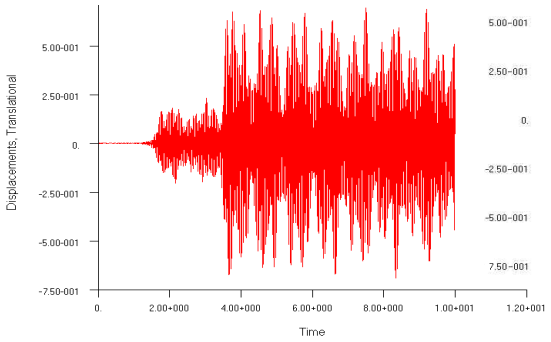


Fig. 14 Time domain response curve of starting process with coupling response considered

Conclusions

Three conditions of the natural frequency and the corresponding critical speed are illustrated in table 1 from the comparison results; it can be concluded as follows:

Table 1 Comparison of the results

Conditions	The first order		The second order		The three order		Starting process
	Natural frequency [*Hz*]	Critical speed [*r/ min*]	Natural frequency [*Hz*]	Critical speed [*r/ min*]	Natural frequency [*Hz*]	Critical speed [*r/ min*]	Maximum displacement [*mm*]
Mass unbalance	84	5040	135	8100	310	21000	0.68
Temperature difference	78	4680	130	7800	302	18120	0.2
Coupling response	67	4020	121	7260	295	17700	0.7

The critical speeds of the rotor under the three working conditions are different, showing that the first order critical speed of rotor system is more sensitive to the limiting conditions.

The first order critical speeds under the three working conditions are designed under the rotational speed 6000[*r/min*], and are lower than the second order critical speed, illustrating that it is flexible rotor. However, according to the general design requirement of rotation equipment, the working speed ω of flexible rotor is preferred between the range $1.3\omega_{n1} \leq \omega \leq 0.7\omega_{n2}$, which indicates that the design working rotation speed is somewhat lower when the eccentric mass is applied and is not conducive to the steady operation of rotor. This can be improved by increase the working speed, or replace the bearings to increase bearing stiffness, or replace the bearings to raise the bearing stiffness to increase the first order critical speed of the system.

The axial temperature difference exists, due to the thermal bending; the rotor system stiffness is redistributed, which results in the decrease of the 1st critical speed of the thermal vibration for the rotor bearing system, and has great impact on the vibration characteristics of the rotor. Moreover, when the two functions at the same time, due to the coupling effect, the first critical speed reduces again, which results in the change of vibration characteristics again.

Seen from the boot process, the displacement of nodes is larger when eccentric mass is applied, and that indicates the rotor is very sensitive to unbalance mass, and the nozzle clogging has great influence on the dynamic characteristics of structure; the temperature difference effect on the nodes displacement is relatively small. However, the vibration when temperature difference is coupled with mass unbalance is more complicated than that of mass unbalance response itself. Node displacement appears more irregular, indicating the coupling effect also has a great impact on the structural dynamic characteristics. The max value of the displacement appears at 3.6[s], showing the structure has passed the first critical speed at this time.

All of the three conditions have affected the dynamic characteristics of rotor system based on the above conclusions. It is shown from the calculation that the design can basically meet the stable operation requirement of the atomizer. In this paper, only one nozzle clogging is considered. However, there will be more extreme situation in the actual condition; therefore, more detail analysis can be conducted to optimize the design in the future research.

Acknowledgements:

1. This paper was supported by the Important Science and Technology Research Project of Chongqing—Project Numbers (CSTC2007AA7003).

2. This paper was supported by the Key Science and Technology Research Project of Chongqing —Project Numbers (CSTC2006AB301)

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