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Mode analysis of automatic placement machine *

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Abstract. The modal analysis of the automatic placement machine is made based on FEM technology. The mathematic model and the FEM model are built; the simulation is made on the factors that impact the natural frequency and the mode shapes; finally, the system model is modified and the mode analysis of the whole machine is made. The results indicate that the smaller the mass ratio of moving object to still object is, the smaller the natural frequency is; the higher the center of gravity is, the smaller the natural frequency is; and the first natural frequency of the modified model is 190.6HZ that exceeds the frequency of the control system and can advance the system dynamic performance.

Keywords: placement machine, FEM, dynamical model, modal analysis

1 Introduction

The automatic placement machine is the important electronic equipment of the PCB manufacturing enterprises, and its performance has an important impact on the electronic industrial development. The mode analysis technology is the basis of the modern mechanical design and the dynamic structural analysis and becomes the powerful tool of the dynamic analysis in recent years. The fully automatic placement machine is a high-speed and high-precision machine, and its structural dynamic characteristics directly affect the machine speed and accuracy. To enhance the dynamic performance of the placement machine, the factors that impact the natural frequency and the modal shapes are analyzed based on MSC. Patran and MSC. Nastran. Based on the results of the analysis, the optimized FEM model is established and its mode analysis is made.

2 FEM modeling of modal analysis

2.1 Mathematical basis of mode analysis

The analysis of the natural frequency and the mode shapes is the basis of the structural dynamical problems. Its importance lies in how to avoid structural resonance, and it is the analysis basis of the structural dynamical response and other dynamical characteristics. Corresponding author.

Normally, the format of the dynamics equation is:

$$[M]\{\ddot{a}(t)\} + [C]\{a(t)\} + [K]\{a(t)\} = \{F\} \tag{1}$$

The equation of Free-vibration is:

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$$[M]\{\ddot{a}(t)\} + [C]\{\dot{a}(t)\} + [K]\{a(t)\} = 0 \quad (2)$$

The practical experience has shown that the impact of damping on the structure frequency and shape is little, and the natural frequency and the mode shapes can be solved in the situation of the free undamped vibration.

The equation of the free vibration for an undamped system can be written as:

$$[M]\{\ddot{a}(t)\} + [K]\{a(t)\} = 0 \quad (3)$$

The free vibration of the elastomer can be resolved into a series of harmonic vibration. The equation of the harmonic vibration is introduced as:

$$\{a(t)\} = \{u\}sin\omega t \quad (4)$$

Here, $\{u\}$ is the intensity; ω is the frequency of the harmonic motion.

Eq.(4) is inserted into Eq.(3),and the solution is:

$$[K]\{u\} - \omega^2[M]\{u\} = 0 \quad (5)$$

The above equations are n-order algebraic equations for $\{u\}$. The necessary and sufficient condition that the sets of equations exist in the non-zero solution is its equation of coefficients is equal to 0, namely,

$$|K_{ij} - u^2 m_{ij}| = 0 \quad (6)$$

This equation is called as the system frequency equation. A n-order algebraic expression can be obtained From Eq.(6):

$$\omega^{2n} + a_1\omega^{2(n-1)} + a_2\omega^{2(n-2)} + \cdots + a_{n-1}\omega^2 + a_n = 0 \quad (7)$$

Supposes the quality matrix and the stiffness matrix of the system are the real symmetrical matrix. Under this condition, the n roots of the frequency equation may prove to be positive real roots in mathematics. These n roots correspond to the system n natural frequencies. It is here supposed that the equation does not have the multiple root, thus the roots may be ordered as follows from small to big: $\omega_1^2 < \omega_2^2 < \cdots < \omega_n^2$. $\omega^r(r = 12.n)$ that is solved from Eq.(7) is inserted into Eq.(3), and $\{u^r\}$ can be obtained which is the system modal vector.

2.2 Dynamical model of joints in placement machine

A CAD representation of the entire placement machine can be seen in Fig. 1. Four sub-structures form three join surfaces, namely, (1) the join surface between the upper rack and the down rack; (2) the join surface between the Y slide and the gantry; (3) the join surface between the X slide and the seat of the placement head. According to the above supposition, joints are simplified for the spring and damping and the dynamical model of the assembly is constituted. In the PATRAN software, the dynamical model of the spring element is shown in Fig. 2, and each sub-structure is connected with the corresponding spring elements. The dynamical model with the joins can be seen in Figure 3. The equivalent stiffness of the joins can be obtained by the literature data. The average equivalent stiffness of the joint node can be seen in Tab. 1.

Table 1. average equivalent stiffness of join nodes (N/m).

joins	X equivalent stiffness	Y equivalent stiffness	Z equivalent stiffness
1	2.4E7	1.2E8	1.2E7
2	2.4E7	1.2E7	1.2E8
3	2.4E7	1.2E7	1.2E8

1 join between X slide and seat of placement head2 join between Y slide and gantry3 join between upper and down rack

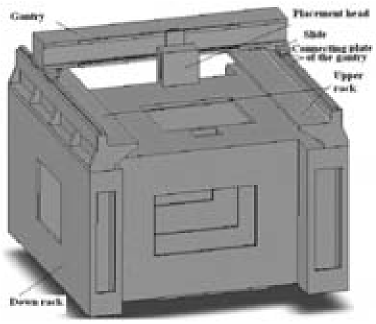

Fig. 1. CAD representation of entire placement machine.

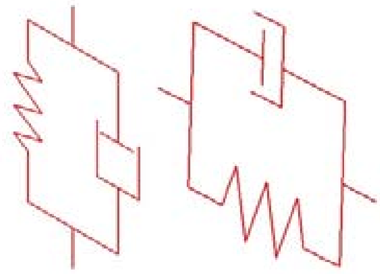

Fig. 2. Dynamic model of spring element.

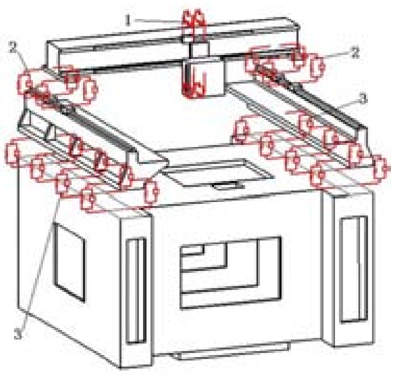

Fig. 3. Dynamic model of joints in placement machine.

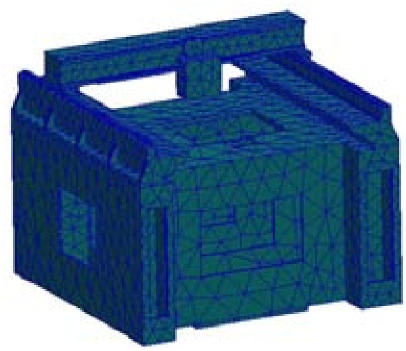

Fig. 4. FEM model of Modal analysis.

2.3 FEM model

The placement machine is mainly composed to the upper rack, the down rack, the gantry, X screw-nut assembly, Y screw-nut assembly and so on. Because the structure of the placement machine is quite complex, the simple model of the placement machine is established directly in the three dimensional software and is imported to PATRAN to carry on the meshing of the assembly. The finite element model is built in Fig. 4.

3 Modal analysis

3.1 Effect of moving-static ratio

Table 2. Effect of moving-static ratio.

Frequency moving-static ratio	1#	2#	3#	4#	5#
1/3m	190.7	237.5	262.1	348.4	400.9
2/3m	190.6	222.0	235.2	306.8	334.4
m	189.0	190.5	229.5	281.6	293.
4/3m	166.4	190.4	217.4	256.7	264.
5/3m	150.1	190.2	201.2	235.6	251.5

From the Tab. 2, as the quantity of the still body reduces, the natural frequency elevates quite slowly; when the quantity of the still body increases, the natural frequency drops quite quickly. Therefore, the original quality is taken as the quality of the optimized model.

3.2 Effect of mass position

The 402 kg load weight is separately added in the original rack, and the density of the rack is changed to remain unchanged for the mass of the rack (origin mass+402 kg). Thus under the same conditions, three analysis results can be compared.

(Downadd-added block under the rackUpadd-added block above the rackOrigin-origin position)

Table 3. Effect of mass position

Frequency Added block	1#	2#	3#	4#	5#
Upadd(402Kg)	157.0	190.4	202.6	250.9	285.6
Origin(402Kg)	167.3	190.4	218.1	257.5	266.1
Downadd(402Kg)	186.5	190.5	229.3	280.0	288.2

According to the table, the model with the block under the rack has the biggest natural frequency; therefore the optimized model uses this type.

3.3 Modal analysis of modified model

According to the above analysis result, the original model is modified, and the modal analysis of the modified model is carried on to obtain its first six order natural frequencies and the corresponding modal shapes.

(1) First six order natural frequencies

Table 4. Effect of mass position

1#	2#	3#	4#	5#	6#
190.6	216.5	234.9	303.0	316.3	346

It is seen from the table that 1# natural frequency of the complete machine is 190.6HZ, and has greatly surpassed the cut-off frequency 150HZ of the control system. Therefore, the system vibration-proof performance can be obviously enhanced.

(2) Modal shape

The first six order modal shapes are shown in Fig 5 $\sim$ 10, the 1st order shape is for the X direction translation; the second order shape is for the vertical direction translation; the third order shape is the Y direction translation; other three orders are for the torsion shapes.

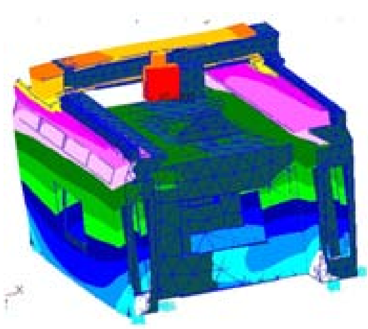

Fig. 5. 1# natural frequency (190.6HZ).

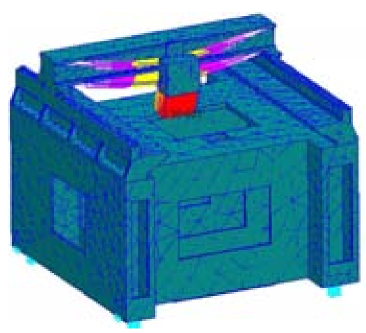

Fig. 6. 2# natural frequency (216.5HZ).

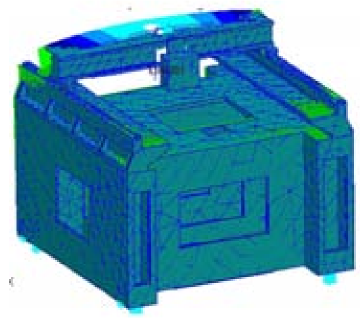

Fig. 7. 3# natural frequency (234.9HZ).

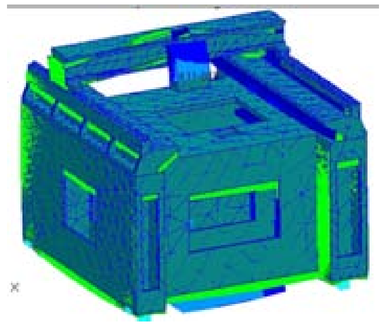

Fig. 8. 4# natural frequency (303.0HZ).

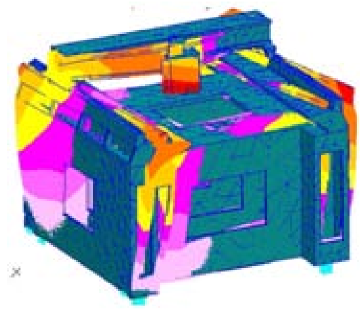

Fig. 9. 5# natural frequency (316.3HZ).

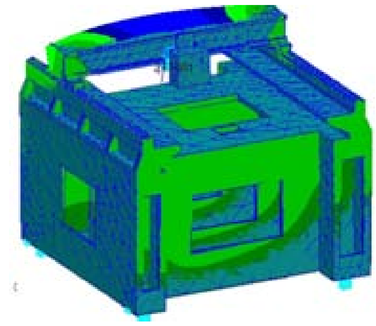

Fig. 10. 6# natural frequency (346HZ).

4 Conclusion

(1) The research that changes the quality of the rack shows that the bigger the static quality is, namely, the smaller the mass ratio of moving object to still object is, the smaller the natural frequency is;

(2) The three models are built including the model with the mass block under the rack, the model with the mass block above the rack and the model without the mass block. The modal analysis of these models is carried on, and the result shows that the higher the center of gravity is, the smaller the natural frequency is;

(3) Finally based on the optimized model, the modal analysis has been carried on, and the natural frequencies and the modal shapes of the optimized model are obtained. The result shows that the first order natural frequency is 190.6HZ that has avoided the excitation frequency of the control system and the vibration-proof performance of the system is effectively enhanced.

References

[1] M. Burdekin, N. Back, A. Cowley. Experimental study of normal and shear characteritics of machined surfaces in contact. *J .Mech.*

[2] R. Connoly, R. H. Thornley. Determining the normal stiffness of joint faces. *ASME*, 1967, **67**: 6213.

[3] C. Dekonink. Experimental investigation of the normal dynamic stiffness of metal joints. *Int. J . Mach. Tool. Des. Res.*, 1969, **9**: 279 – 292.

[4] S. Fumin, Z. Xiaoli, M. Ruzhen. Research to manufacture of smt2505 full vision and multi-functional pick&equipmen. *Equipment for Electronic Products Manufacturing*, 2004, **31**(4): 219–223.

[5] A. N. I. S. Grotzinger, D. Johnson. Component allocation and partitioning for a dual delivery placement machine. *Operations Research*, 1988, **36**(2): 176–191.