Vibration isolation system design and analysis for ultraprecision roll machining machine

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Abstract
An ultra-precision roll machining machine is developed in our laboratory and in this paper the design and analysis of a vibration isolation system for the machine are presented. A roll machining machine is used to machine micro-patterns on a surface of a roll mold which is used to fabricate patterned optical film. For the roll machining machine, with a roll weight capacity of 3 000 kg and a working distance of 2 m, in order to suppress the vibration noise which is transferred from the floor ground to the machine a vibration isolator is adopted. For the vibration isolator system two design types are considered and the first one is a system with the bed supporting cradle structure which is supported by the isolator parts and the other one is a system which adopted direct mounting (on a bed structure) method for isolators. Those two types are designed and analysed and the performance in several aspects is compared. The comparing aspects include thermal expansion and deformation, static structural deformation due to weight, modal frequencies and mode shapes and force-loop response in the frequency domain.

1. Roll machining machine description

Figure 1 shows the schematic of the machine. The machine consists of a headstock, a tailstock, cutting tool feeding axes and a bed structure, which is similar to the typical lathe machine tool. The machine can make, for example, triangle-shaped patterns with a pitch size of a few tens of micrometers on the roll surface. The patterned roll is used as a mold to manufacture patterned polymer film by various methods which includes a method using ultraviolet light curing polymer resin. The patterned film is used, for example, as a prism film layer of the back-light-unit(BLU) of the liquid-crystal-display(LCD). For the enhanced precision comparing with the conventional lathe(typically adopts rolling bearing for rotary and linear axis) all the machine's motional parts are supported by the oil-hydrostatic bearing which has advantages of low friction and high damping . The machine can handle roll workpiece with up to 2 m length and 3 000 kg weight. The weight of the machine is about 25 000 kg which should be supported by the vibration isolation system.

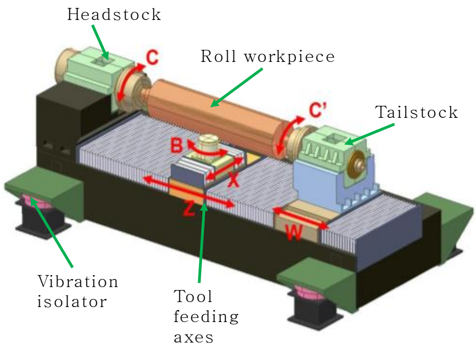


Figure 1: Roll machining machine schematic view

2. Vibration isolation system design

The floor vibration can be transmitted to the machine and if the transmitted energy excites the mode frequencies which are related with the distance between the tool and the roll the machined pattern shape will be distorted. In order to solve this problem in the design procedure of the machine the damping, stiffness and mode frequencies of machine components (such as bed , bearings, feeding table, spindle shaft) need to be considered. If there is a need for further enhancement of the external vibration insensitivity of the machine the vibration isolation system will be a good choice.

Two types of isolators are considered as candidates, which are a console type and a cradle type. As seen in the Figure 2 the console is a structure in order for the isolator to directly connect to the bed and the cradle is a bed supporting structure which has three contacting pads.

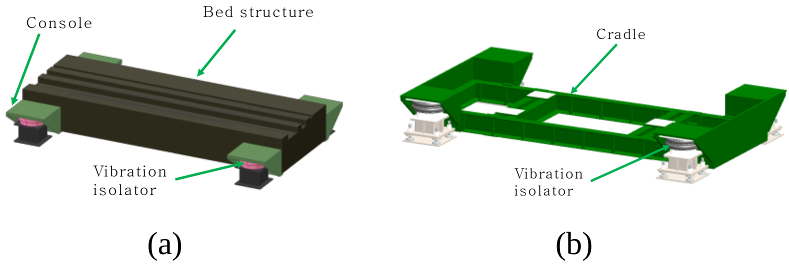


Figure 2: Two types of vibration isolators designed. (a) console type (b) cradle type

In order to support the machine's weight of 25 000 kg the stress analysis using FEM(Finite Element Method) is performed as shown in the Figure 3. For the console type the insertion-bolt in the granite bed was a problem and for the cradle type the long cross-rail structure was re-designed.

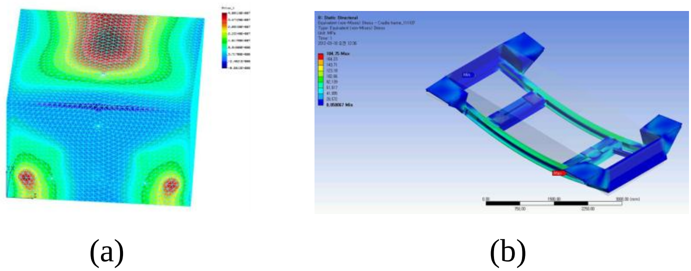
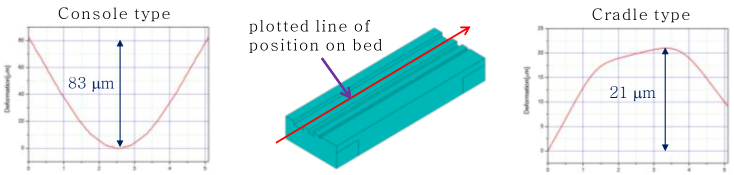


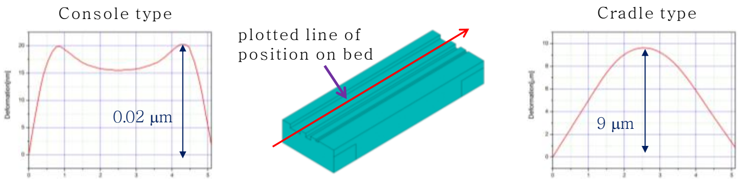
Figure 3: FEM stress analysis results for component design of two types (a) console (b) cradle

3. Analysis for performance comparison

The structural and thermal deformation analyses for two types are shown in the Figure 4. The granite material bed structure is used for deformation analysis due to its weight and the cradle type has lower deflection of the bed (measured at the position of hydrostatic guide rail). However, for the thermal deformation the amount is much lower in the case of console type which is due to the low spring constant value of the isolator. Figure 5 shows the modal analysis results. In case of console type the first modal shape is involving deformation of roll workpiece (46 Hz) and for the cradle type the frequency is 28 Hz whose modal shape involves the deformation of the cradle structure only. Therefore, the cradle type has mode frequencies below the roll-deforming modal frequency.

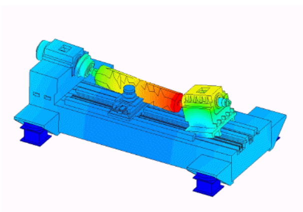


(a)

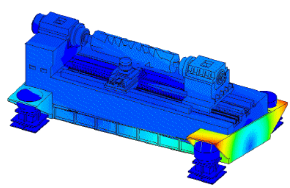


(b)

Figure 4: Structural and thermal characteristic analysis. (a) structural deformation (b) thermal deformation



(a)



(b)

Figure 5: Vibration mode analysis results (a) console type (b) cradle type

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