SUPPRESSION CONTROL OF POSITIONING NOISE FOR DUAL STAGE WITH COUPLING DAMPER

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Long stroke and nanometer precision positioning technology based on dual stage has important applications in the fields of ultra precision machinery and equipment, which consists of a coarse stage for long stroke and a fine one for high precision. In order to further suppress the positioning noise of dual stage, we design an abdominal and passive damper for the fine stage. However, it becomes a coupling damper owing to its location. Therefore, we explore that whether it is still effective in suppressing the positioning noise of the system and how we should choose it reasonably. Then, model of dual stage with coupling damper is established based on parallel dual servo control, and used to study the effect law of coupling damper and reveal its effect on noise transmission and dynamic characteristics of each stage. In addition, a series of system tests are followed, and it is verified that coupling damper can be used to suppress the positioning noise of dual stage, and reduce the relative distance between coarse and fine stage.

Keywords: Dual stage, Positioning noise, Coupling damper, Stroke saturation

1. Introduction

Dual stage is the effective way to realize the positioning platform with long stroke and nanometer level of positioning accuracy, and is widely applied in many fields such as ultra precision machining [1-4], which consists of one coarse stage to realize the long travel requirements, and one fine stage to achieve nanometer level of positioning requirements, making full use of their respective advantages. In this kind of system, usually, coarse stage is driven by linear motor or DC motor combined with screw, and fine stage is driven by voice coil motor or piezoelectric ceramics combined with flexure hinges.

We have constructed a dual stage that the coarse stage is driven by permanent magnet synchronous linear motor to realize no contact, high speed and long stroke, and fine stage is driven by voice coil motor to achieve no contact, no force ripple and high precision [5]. Furthermore, air bearings are adopted for both coarse stage and fine stage to obtain the nanometer level resolution, because they have the advantages such as extremely low friction coefficient, no creeping under

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the low speed condition, no abrasion and no frictional heating. However, these also induce transmission damping of air bearing very small, and the rail gas film stiffness and damping very low, which make the linear air bearing vulnerable to disturbances. And the disturbances are mainly from two aspects, that is, the change of gas film clearance, and the joggling of air guide pipe and motor cable. These can affect the positioning precision of the stage, and especially, the traditional throttle-type air lubricated guide is more vulnerable to disturbance [6]. But considering the cost, we still choose this type of guide for the dual stage, and take appropriate measures to mitigate the effect of the disturbances on it.

Aimed at solving the shortcoming of linear air bearing vulnerable to disturbance, some organizations and scholars proposed some new and improved methods in some links of servo control. For example, on software, the disturbance observer [7] is designed to observe the equivalent disturbance, and the corresponding equivalent compensation is introduced to the loop control to restrain the disturbance with the help of high speed controller. On hardware, for one thing, new type of aerostatic guide [8] is designed, but its structure is very complex, which sets a very high request to the processing technology. For another thing, mechanical damping is designed and added to the aerostatic guide, such as electro rheological fluid damper [9,10], which can realize the comprehensive damping characteristics of small damping coefficient for acceleration process and big one for positioning process, however, it makes the control complex and may bring electromagnetic interference. In addition, by analyzing the transfer function of disturbance response [11], the system gain is improved to achieve high servo stiffness or the plant mass is increased to make its gain low to suppress disturbance. But high gain easily leads to system unstable, and it is difficult to obtain both high gain and good stability, and big inertia of the plant can decrease the dynamic characteristic of the system.

In order to further suppress positioning noise of the dual stage, we design an abdominal and passive damper for fine stage, because fine stage locates at the end of the dual stage and decides positioning precision of the system, and the damper can increase the transmission damping of air bearing of fine stage and increase the ability of noise attenuation. However, the damper locates at the junction of coarse and fine stage, and becomes the coupling damper of the dual stage system, which may cause some changes in the characteristics of the dual stage, yet this point still has not attracted attention and been analyzed. Therefore, model of dual stage with coupling damper is established based on parallel dual servo control and used to study the effect law of coupling damper, which reveals the coupling characteristics between coarse and fine stage. Then, the performance of dual stage with coupling damper is evaluated by experiments.

2. Dual stage design and kinematic model

2.1. Dual stage configuration

Mechanical structure of dual stage is shown in Fig.1. The mover of voice coil motor (VCM) is fixedly connected with the terminal platform, and the stator of VCM is fixed to the stage driven by linear motor, and the cooperation between the above two parts achieves rapid precise positioning. The stator of linear motor is fixed to the vibration isolation platform, by interaction with which the mover of linear motor realizes the long stroke positioning. Both the coarse stage and terminal platform are fixed with corner cube reflectors, the position signal feedback device of the dual stage, whose corresponding laser units and detector heads are all fixed in the vibration isolation platform. In addition, the terminal load mass is 3.9kg, and the stroke of dual stage is 300mm.

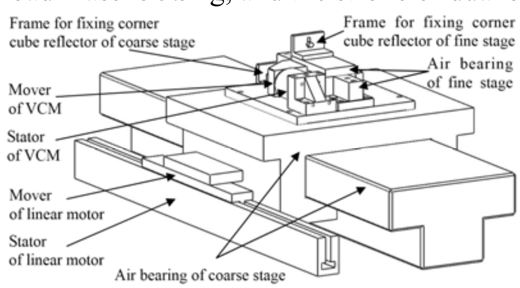

Fig.1 The mechanical structure of dual stage

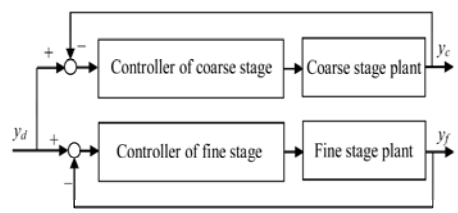

Fig.2 Control strategy of dual stage

2.2. Dual stage control system

As shown in Fig.2, the control strategy, parallel dual servo algorithm [12], is adopted in the dual stage system. That is, the controller of each stage is designed according to the single input single output system (SISO), neglecting the coupling effects between them. Namely, the closed loop feedback control of each stage is formed by the respective position sensor with the same reference position. Thus, the coupling effect between coarse stage and fine stage is regarded as the disturbance to each other, which can be collected by the respective position sensor and be weakened by the respective position controller. Fig.3 shows the concrete control model established in Simulink, and the top part is coarse stage control model, and the lower part is fine stage control model. The control method, position loop plus current loop, is adopted for each stage, and differential forward PID control method is used in position loop to avoid system oscillation and improve the dynamic characteristics for frequent change of given value (command). Coarse stage and fine stage follow the same trajectory command, and each stage has its own independent position sensor. In addition, the coupling effect of two stages is also considered in the control model, that is, counter-electromotive force of VCM is influenced by the relative velocity disturbance from coarse stage, while reaction force from fine stage is applied to coarse stage. And in Fig.3, the parameters and variables are defined as follows: R is the position command, K_{cD} and K_{fD} are the position loop differential gains, K_{cI} and K_{fI} are the position loop integral gains, K_{cP} and K_{fP} are the position loop proportional gains, K_{cc} and K_{fc} are the preamplifier gains, T is the time constant of command filter, T_s is small time constant of current loop, K_s is magnification of pulse width modulator and PWM transform device, K_{ciI} and K_{fiI} are the current loop integral gains, K_{ciP} and K_{fiP} are the current loop proportional gains, R_c and R_f are the motor resistances, L_c and L_f are the motor inductances, K_{ce} and K_{fe} are the counter-electromotive force constants of motors, K_{cf} and K_{ff} are the force constants of motors, m_c and m_f are the load mass, F_{VCM} is reaction force on coarse stage applied by fine stage, Y_c is the position of coarse stage, Y_f is the position of fine stage, Y_{cf} is the relative distance between the two stages.

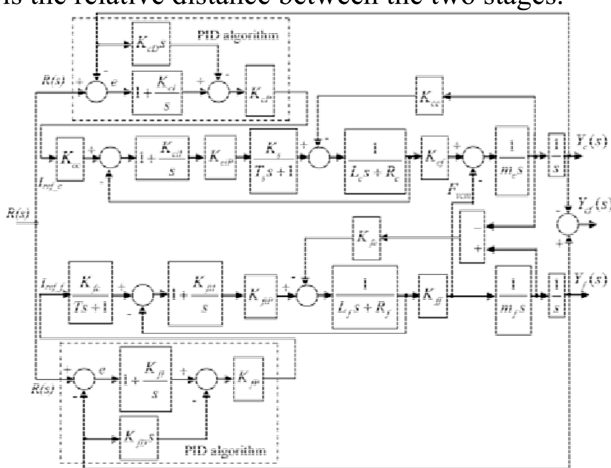

Fig.3 Simulink control mode of dual stage

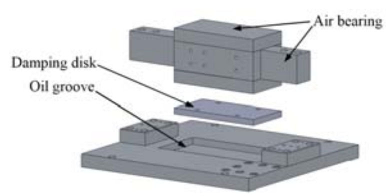

Fig.4 Damper structure of fine stage

3. Damper design and kinematic model of fine stage

3.1. The layout design of the damper

The common damper structure [9, 10] is that, the drive motor is located between two guide ways with the shape of letter π , and a pair of dampers are located on both sides of the air bearing stage, and the center of driving axis, the center of both guide ways and the center of table mass are placed in the same line to minimize the linear motion error such as straightness, yaw, pitch and roll. However, this requires higher mechanical design and assembly technology, especially, like the stage under the drive of one-side mode as shown in Fig.1, if the dampers are placed outside the stage, the connecting arm for fixing the dampers on the stage is too long to keep light mass and enough structure stiffness, which is not good to dynamic characteristics of the system. Therefore, as shown in Fig.4, we design a new structure damper to increase system transmission damping, which consists of damping disk fixed in the bottom of the aerostatic bearing sleeve and an oil groove directly under the disk. And the size of friction surface of damping disk is the same as that of the bottom surface of the aerostatic bearing sleeve, and for the oil groove, its transverse dimension is slightly bigger than that of damping disk, and its longitudinal dimension is a little bigger than that of stroke range of fine stage, and it is deep enough to make the damping oil over the damping friction surface of damping disk. It is obvious that this type of damper has the advantages such as almost no expansion in space, no increase in load mass and no change in mechanical structure stiffness.

3.2. The control model of fine stage with damper

The Simulink control model of fine stage with damper is shown in Fig.5, and F_{fN} is the external disturbance to aerostatic bearing sleeve of fine stage, c_f is the damping coefficient of the damper, and the remaining parameters meanings are the same as that in Fig.3. Based on the above model, numerical simulation of fine stage is made, under the condition of keeping the proper control parameters unchanged, as shown in Fig.6 and Fig.7, where c is the damping coefficient given in units of Newton second per meter. Fig.6 shows the Bode plot from noise disturbance F_{fN} to position Y_f of fine stage, which denotes that noise suppression degree becomes much bigger with increasing damping coefficient. Fig.7 shows the closed-loop Bode plot of fine stage with damper, which indicates that, with increasing damping coefficient, bandwidth of fine stage system decreases and its dynamic characteristic gets worse. Therefore, for the sake of keeping quick response of fine stage, the damping coefficient of the damper can't be chosen too big.

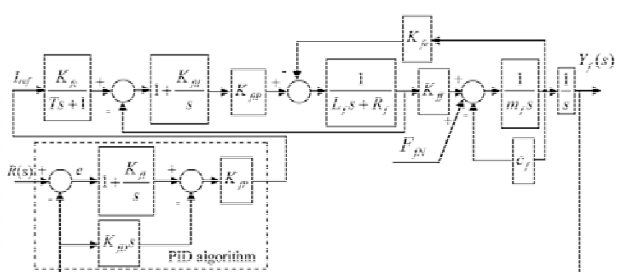

Fig.5 Simulink control mode of fine stage with damper

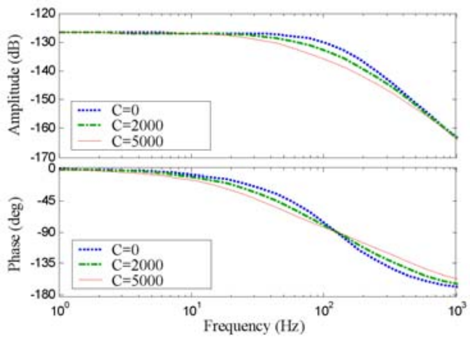

Fig. 6 Simulation Bode plot from noise disturbance to position of fine stage

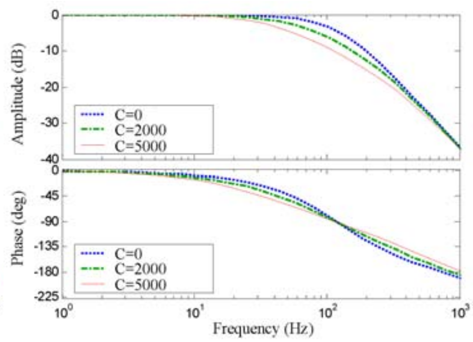

Fig. 7 Simulation closed-loop Bode plot with damper

4. Effect of coupling damping on the dual stage

Because the damper locates at the junction of coarse and fine stage, and becomes the coupling damper of the dual stage system, which may cause some changes in the characteristics of the dual stage, we further study the effect law of coupling damper and make sure whether it can continue to benefit the positioning noise suppression of dual stage. Based on the control diagram in Fig.3, the new dual stage control model added with the coupling damper is established in Simulink as shown in Fig.8, where F_{cN} and F_{fN} are external disturbances, c_f is the coupling damping coefficient, and the meanings of remaining parameters are the same as that in Fig.3. By numerical simulation, we find that, under the condition of keeping the proper control parameters unchanged, with increase in the coupling damping coefficient, the characteristics of dual stage are exhibited as the following three points. Firstly, expression of fine stage in dual stage system is almost in the same situation as it works alone, that is, the dynamic characteristics of fine stage decrease like in Fig.6, and the ability of suppressing the positioning noise becomes much bigger like in Fig.7. In the meantime, the characteristics of coarse stage are almost not affected by the damping. Secondly, the ability to influence the position Y_f of fine stage by noise disturbance F_{cN} of coarse stage is increasing as shown in Fig.9, where c is the damping coefficient given in units of Newton second per meter. Finally, the maximum relative distance between coarse stage and fine stage becomes smaller as shown in Fig.10, where c is the damping coefficient given in units of Newton second per meter. We further confirm these conclusions by the following experimental research. At the same time, some conclusions above are not difficult to understand, just imagine, when the viscous force of the coupling damper increases big enough, the mover of fine stage is almost integrated with that of coarse stage, then the positioning noise of coarse stage almost all reflects in the fine stage, and the dynamic characteristics of fine stage are almost lost and its function of high precision compensation disappears, which also makes the dual stage lose its meaning. Therefore, coupling damping can not be too large, and we should make a compromise between the positioning noise suppression and the dynamic characteristics of fine stage.

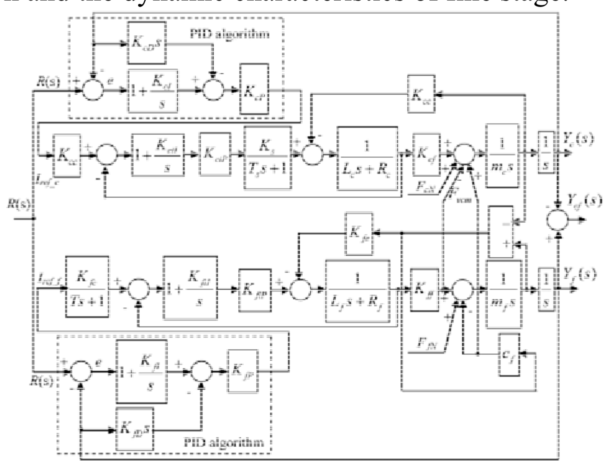

Fig.8 Simulink control mode of dual stage with coupling damper

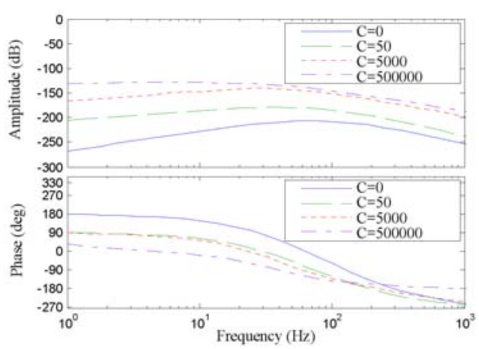

Fig. 9 Simulation Bode plot from noise disturbance of coarse stage to position of fine stage

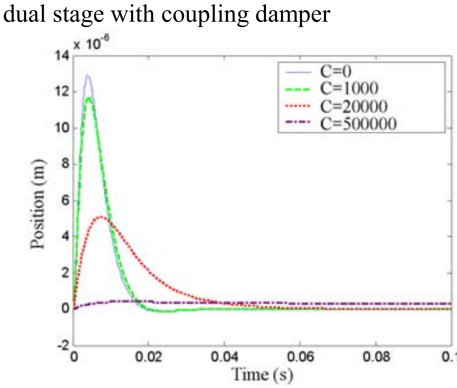

Fig. 10 Relative position between coarse and fine stage with 20μm step response

5. Experimental results

The experimental system of dual stage consists of the following parts: host computer, servo motion control card, coarse stage and fine stage, vibration isolation platform, biaxial laser interferometer, and damper of fine stage. And servo motion control card takes +/-10V command, 16 bits of D/A converters, 110μs servo period. The biaxial laser interferometer provides the resolution of 20nm and maximum response speed of 400mm/s.

The host computer and servo motion control card constitute the control system of dual stage, and the computer as a human-computer interface, can be used to debug and set motion parameters, monitor the system operation status and render the state curve, and the servo motion control card as the core of the control system, is used to receive computer instructions, acquire the position pulses of laser interferometer, complete the operation of servo control, implement the real time closed position loop control, and upload the system motion information to the host computer through PCI bus. In addition, the vibration isolation platform is used to fix the dual stage and reduce the effect of the vibration and noise from external environment.

The control system structure diagram of dual stage is shown in Fig.11, and two servo axes of the motion control card are used to receive the position command and complete the closed loop control of coarse stage and fine stage respectively.

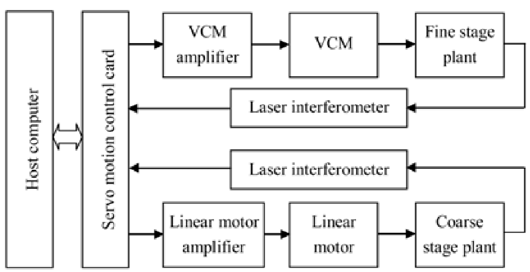

Fig.11 Structure diagram of motion control system

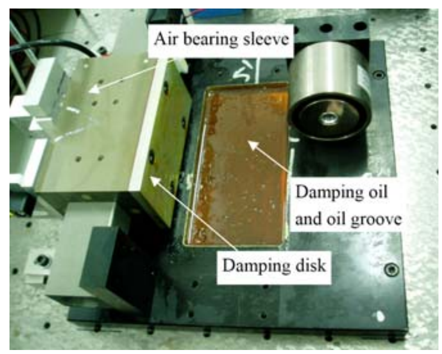

Fig.12 The damper of fine stage

5.1. Positioning noise suppression of fine stage

According to the structure design method described in the section 3.1, the damping oil damper for fine stage is designed and manufactured as shown in Fig.12, and the parameters of damper are given in Table 1. The damping oil is required to be excellent oxidation stability and anti rust, and hydraulic slide way oil with good stick-slip characteristic, and be able to change its viscosity by adding different amount of thickening agent.

Table 1

Parameters of damping oil damper

Parameters	Dimensions
Damping disk	140mm×72mm×10mm
Oil groove	176mm×88mm×10.5mm

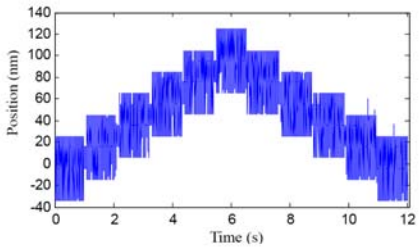

Fig.13 Positioning resolution of fine stage without damper

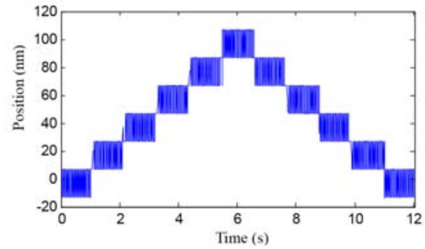

Fig.14 Positioning resolution of fine stage with damper

Under the condition of keeping the proper control parameters and load mass unchanged, the performance of fine stage is tested based on the new damper designed and servo system as shown in Fig.11, while only the fine stage is fixed on the vibration isolation platform for the closed-loop control. By gradually increasing the damping oil viscosity, we get relatively ideal positioning noise and dynamic characteristics of fine stage, and in this case the damping coefficient is measured, 293.78N/ (m/s). As shown in Fig.13 and Fig.14, positioning resolution of fine stage is the same 20nm with and without damper, positioning noise of fine stage is 60nm (pk-pk) without damper and 20nm (pk-pk) with damper, respectively, and positioning stability of fine stage is greatly improved. Since the control parameters of fine stage remain unchanged, it must be the viscous force of damping oil that improves the mechanical properties of air bearing and suppresses the positioning noise, which verifies that the theoretical derivation and the designed damping oil damper are effective. In addition, the air bearing with such damping structure is more beneficial to keep the working environment clean than hydrostatic guide.

5.2. Performance evaluation of dual stage

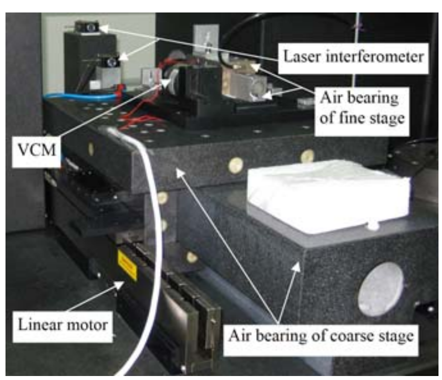

Fig.15 Experiment photo of dual stage

Experiment devices of dual stage are shown in Fig.15, and the performances of dual stage with and without the designed damper are tested.

Fig.16 shows the dual stage response for 20nm stepwise positioning, and without the damper, the positioning noise of fine stage is 80nm (pk-pk) as shown in Fig.16(a), that of coarse stage is 200nm (pk-pk) as shown in Fig.16(b), and with the damper, the positioning noise of fine stage is 60nm (pk-pk) as shown in Fig.16(c), that of coarse stage is 200nm (pk-pk) as shown in Fig.16(d). That is, the positioning noise of dual stage with damper is smaller than that without damper. Thus, appropriate coupling damping is good to positioning noise suppression of dual stage, and the performance of coarse stage is almost not affected.

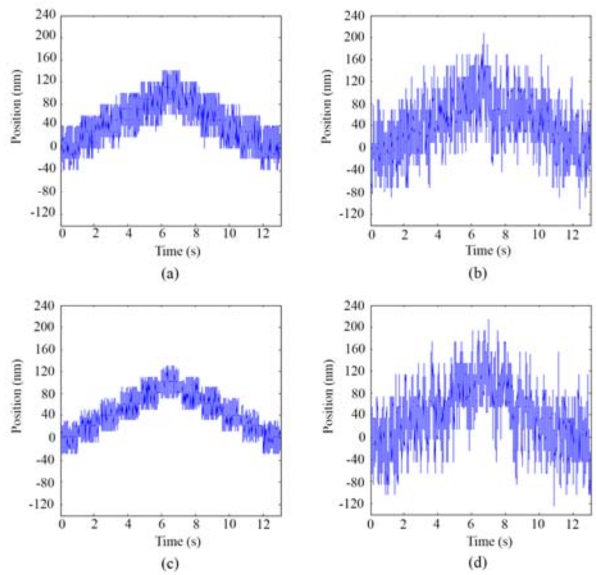

Fig.16 Stepwise positioning resolution test of dual stage with and without damper. (a) Position of fine stage without damper, (b) Position of coarse stage without damper, (c) Position of fine stage with damper, (d) Position of coarse stage with damper

However, terminal positioning noise of dual stage with damper is 20nm (pk-pk) smaller than that of dual stage without damper, but 40nm (pk-pk) bigger than that of fine stage working solely with damper. The reasons of such results are mainly the followings such as the disturbance from pipe and damping oil of fine stage excited by coarse stage, electrical interference of coarse stage crosstalk to the servo system of fine stage. And all these reduce the inhibitory effect of coupling damper on positioning noise of fine stage.

Fig.17 shows the planning trajectory and motion parameters of the dual stage without damper, and the movement distance is 300mm with velocity of 100mm/s. Fig.17(c) shows the following errors of coarse stage and fine stage comparing with the planning trajectory, and Fig.17 (d) shows the relative distance curve between fine stage and coarse stage, and the maximum distance is 40.9μm. Then, by using the damper, we make the same test on dual stage as shown in Fig.18, and the maximum relative distance between the two stages is reduced to 34.6μm. This is another exciting harvest, because it is beneficial to avoid the stroke saturation of fine stage.

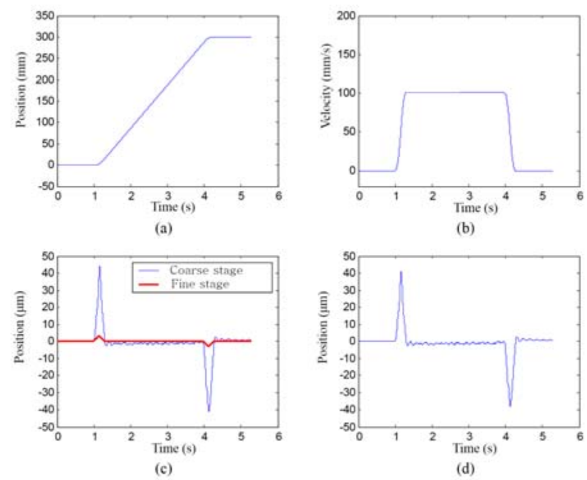

Fig.17 Performance test results of dual stage without damper. (a) Planning trajectory, (b) Measured velocity curve of fine stage, (c) Following errors of dual stage, (d) Relative position between two stages

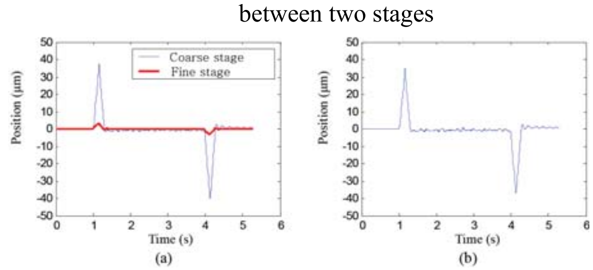

Fig.18 Performance test results of dual stage with damper. (a) Following errors of dual stage, (b)Relative position between two stages

6. Conclusions

With the rapid development of integrated circuit and digital control technology, ultra precision positioning stage, as the key component of precision instrument and equipment, is urgently required. In order to realize long stroke and nanometer level of positioning requirements, the dual stage is designed and developed. And to further suppress its positioning noise, the abdominal and passive damper is designed for fine stage that determines the precision of dual stage. Experiments demonstrate that the viscous force of damping oil can effectively suppress the positioning noise of fine stage. However, the damper for fine stage becomes the coupling damper in dual stage system. Thus, the model of dual stage is established based on parallel dual servo control, and simulation and experimental analysis are performed, which indicate that appropriate coupling damping is good to positioning noise suppression of dual stage, and the performance of coarse stage is almost not affected, at the same time, the maximum relative distance between fine stage and coarse stage is reduced and this is beneficial to avoid the stroke saturation of fine stage, but coupling damping can not be too large, and a compromise should be made between the positioning noise suppression and the dynamic characteristics of fine stage.

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