Introduction of four different drive systems used in LAMOST focal plane

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ABSTRACT

This paper describes four different drive systems adopted in LAMOST focal plane mechanism to achieve four movements: field derotation, focal plane attitude adjustment, focusing and move aside out of light path for optical checking. Different type drive systems, such as worm gear drive, spur gear drive, friction drive and direct drive, which were devised and used in telescopes in the past years, have their own inherent characteristics and their working conditions. According to feasibility, reliability, suitability and cost effective, friction drive, worm gear drive, ball screw drive and chain drive are selected to as the drive systems for the above four movements. The on-shop test results show that all the drive systems have met the design goals with the accuracy of image field derotation 0.45 arcsec, attitude adjustment 0.24 arcsec, focusing 2 microns and move aside 0.02mm.

Keywords: large telescope, drive system, accuracy, structure design

1. INTRODUCTION

The Large Sky Area Multi-object Fiber Spectroscopic Telescope (LAMOST), a national major scientific project in the process of construction in China, is an extraordinary horizontal meridian reflecting Schmidt telescope with 4 -meter aperture and 5- degree field of view. Due to its exceptional configuration, the reflecting Schmidt plate (M_A), focal plane mechanism and spherical primary mirror (M_B) are situated on the top of three base-connected concrete piers which are structurally isolated from the building. The light from the observing celestial objects is reflected by Schmidt plate, spherical mirror and imaged on focal surface, 1.75m diameter spherical cap formed by 4,000 fibers which are accommodated on the focal plate and linked to spectrographs simultaneously. The whole focal plane mechanism is shown in Figure 1.

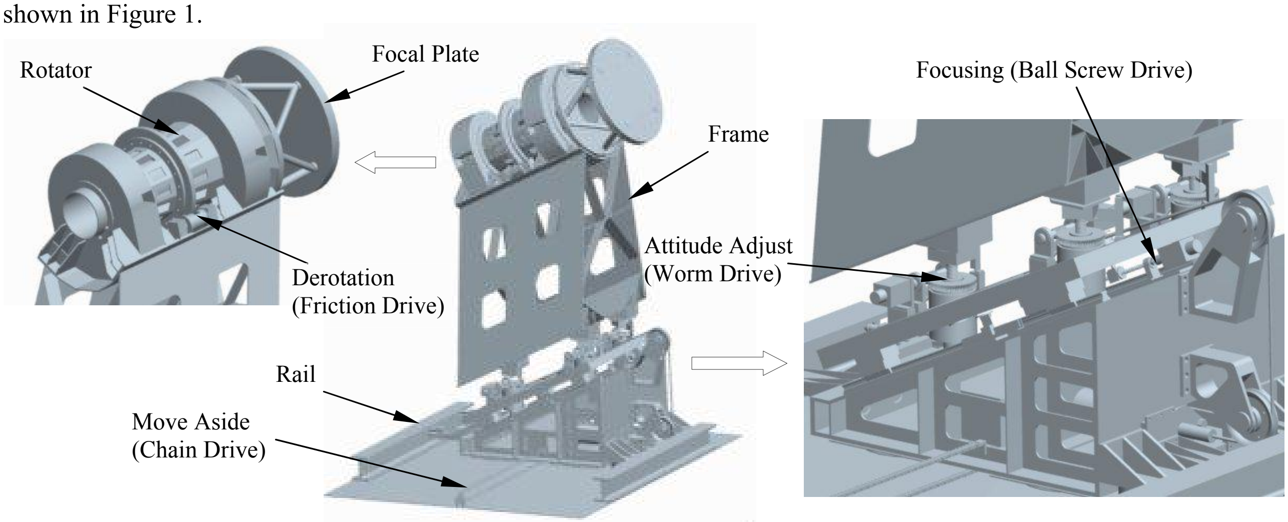

Figure 1 Focal plane mechanism overview

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According to the LAMOST operation configuration, the focal plate, in which 4,000 fibers were installed, has four motions to perform: derotating about light axis to compensate the image rotation in the field of view, tilting a certain angle about the focus in the meridian to get best image quality when observing different sky area[4], focusing along light axis to offset the thermal influence on the mechanical structure and moving aside from the light path, along the direction of perpendicular to light axis, for optical checking of M_A and M_B. Here the big problem is to choose the right drive type to accomplish the motion goals. Basing on the feasibility, reliability, cost effective and maintainability and according to the motions specifications, four different drive systems are adopted respectively: roll friction drive for image derotation, worm gear drive for attitude tilt adjustment, ball screw drive for focusing and chain drive for moving aside. The on-shop tentative assembly of these drive systems and corresponding control system test, cooperation with France CSP Inc., has been finished in Nanjing Institute of Astronomical Optics and Technology (NIAOT). The test data obtained so far suggest that the performance of these drive systems achieved the design targets. In the following sections, the structure design of the four drive systems are investigated respectively and detailed with corresponding calculations.

2. FRICTION DRIVE FOR IMAGE DEROTATION

2.1 Summary

The specifications of image derotation are given below:

- Tracking range: ±22.5°
- Tracking velocity: 0″-15″/s
- Tracking accuracy: 1″
- Maximum rotation velocity: 1°/s
- Maximum rotation acceleration: 0.3°/s^2
- Service temperature: -25℃- +35℃

Besides the specifications mentioned above, the following problems are also important which should be considered carefully at the period of structure design:

- Just as the Figure 2 shows, the whole image derotation structure of focal mechanism is in the light path. So, a good design is to reduce the buffet light area as small as possible.

- The all structure of focal plate, truss and rotator are supported by the frame welded using steel plates shown as Figure 1. The dead weight of these structures is about 6 metric tons. The support frame is also in the light path, as the Figure 2 shows. So, besides the enough stiffness to support the above structure, the cross section of frame will be as small as possible.

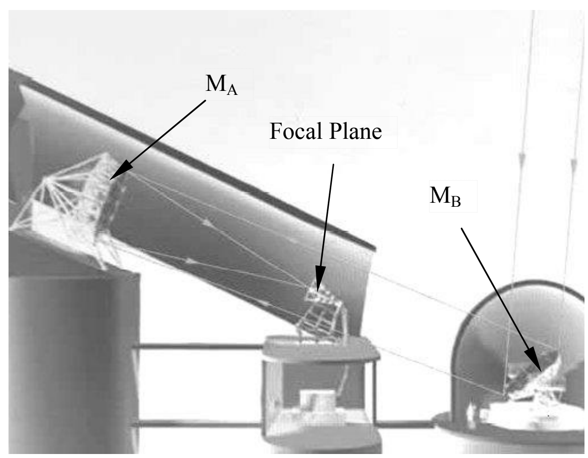

Figure 2 Optical configuration of LAMOST

- As mentioned in introduction section, there are about 4,000 fibers and 4000×11 wires mounted in the focal plate. So the diameter of inner hole of rotator will be large enough to let the fibers and wires to go through to the rear of focal mechanism and then to the spectrograph room and control room underneath pier. Meanwhile, the bearings to carry the rotator will have enough large inner holes, too. According to the dimension of fibers and wires, the ideal diameter of inner hole is about 540mm.

- A large number of fibers and electrical power and signal cables are required to pass from the stationary pier onto the rotating rotator (±22.5° of rotation) and then located in the focal plate. Due to pointing considerations, the torques necessary for these cable transfers must be as small and as linear as possible. But, it is a hard work to keep the torque constant for the 4,000 fibers and 44,044 electrical wires and signal cables. So, a motorized cable transfer device will be used on the focal mechanism.

➢ The whole structure of image derotation is supported from the focusing plate by three attitude adjustment pads bolted onto the bottom plate of frame as shown in Figure 1. High stiffness to weight ratio is very important for the structure.

The image derotation drive system is somewhat different from that of other telescope that an exotic structure and drive system will be required to meet the above demands and to guarantee high level performance at near zero speed. The detailed introduction will be given in the following subsections.

2.2 Drives

Friction drive was adopted to drive the focal plate. Comparing to other drive systems, such as gear drive system, no backlash, no high-frequency periodic errors and easy manufacture are the main advantages of friction drive. According to the experiment study results pointed out in the reference paper [6], the friction drive can get high drive accuracy in the state of slow rate, small acceleration, and little torque change, cleanness of contact surface and high precision of mounting alignment. By using relatively small drive rollers, the drive motors can directly drive the shaft, without the necessary of any gear reduction. The whole image field derotation mechanism is shown in Figure 3.

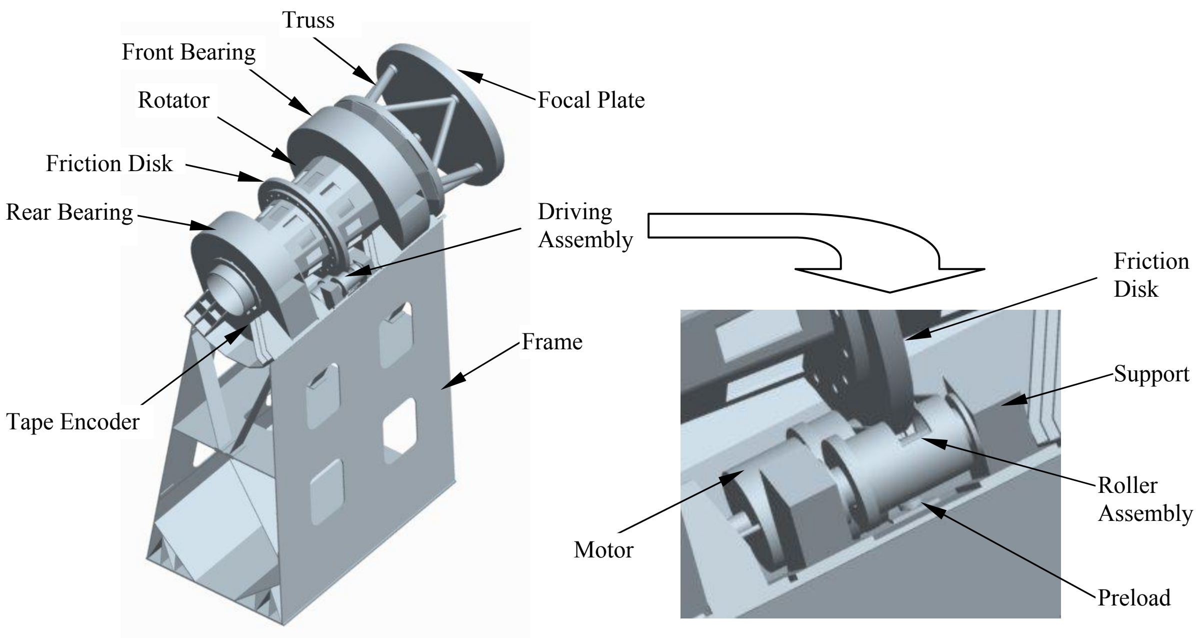

Figure 3 Structure of friction drive system

In order to lessen the trap light area, mounting plate of bearing housing, mounting plate of roller support pillars and top plate of support frame are in the quasi-circle shape whose diameter is little smaller than that of focal plate (1.75m), showing in Figure 4. With the help of FEM software ANSYS, the bearing housings and frame, which are weldments, were optimized to have a high stiffness to weight ratio according to the above design requirements. The axial stiffness of front bearing housing is 1.55×10^5 N/mm and rear bearing housing is 2.44×10^5 N/mm. Under the dead weight of above structure and itself weight, the deformation of support frame in vertical direction is less than 0.815mm. The stiffness of the frame will be improved greatly by the two bearing housings (front bearing and rear bearing

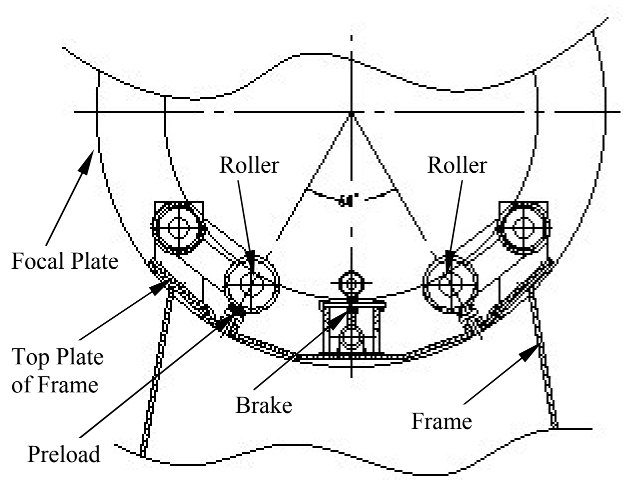

Figure 4 Configuration of friction drive

shown in Figure3) after they are bolted onto the frame. So, in the FEM calculation model, an equivalent stiffness bars (3-D spar element) were used to simulate the bearing basements. In this case, the top plate of the frame has little influence on the stiffness of the support frame and could be thinner to decrease the weight. After the FEM optimization design, the maximum deflection of focal plate center is 11.65 μ m and the maximum tilt angle of focal plate is 6.86″ . The lowest eigenfrequencies of the mechanism are：lateral mode 11.065H_Z, fore-aft mode 16.473H_Z and torsional mode 22.477H_Z.

The structure used to de-rotate the fibers and wires is shown in Figure 5. In the clamp, there are two kinds of slots, one for fibers (4mm in diameter) and another for cables (8mm in diameter). The clamp is embedded in the rotating plate with a ring gear mounted around its periphery. This ring gear will be meshed with two drive pinions each driven by a DC torque motor and gearbox unit, with an integral tachogenerator. The rotating plate is supported by two pivots fixed to mounting base. The position switch will stop the rotation if the rotating angle exceeds +/-22.5 degrees, providing protection primarily to the fibers and cables. However, the driving requirements for precision are not as high as for tracking in the image field derotation.

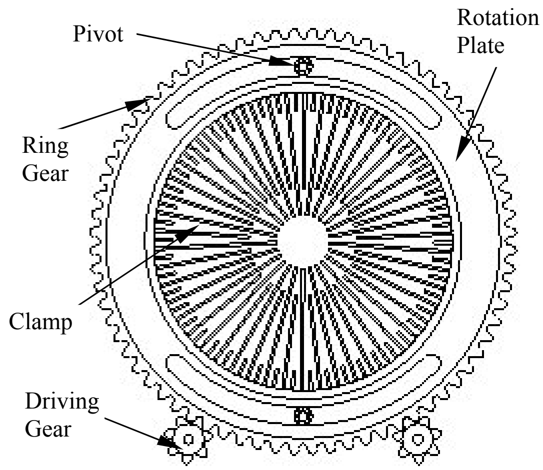

Figure 5 Structure of cable wrap

Since friction drives use contact friction to transmit the necessary drive power and torque between the driving and driven element, the cylindrical elements must be pressed together with sufficient force to maintain consistent drive between the roller and wheel with the lowest amount of slip under normal operation. On the other hand, the force must be low enough to maintain a safe level of contact (Hertzian) stress. The safe upper limit of preload can be got by calculating a suitable contact stress value, while the lower limit of preload is the critical load under which the slippage does not occur.

The drive consists of an 81.25mm diameter, 80mm long drive roller mounted at the top plate of support frame and pressed against the outside edge of the 1300mm diameter friction disk as shown in Figure 3. The drive reduction is 16. The big disk is made of bearing steel. Its surface roughness Ra was measured as 0.4 μ m, and hardness was measured as Rockwell C hardness of 58. The rollers are made of an alloy of steel, heat-treated to a Rockwell C hardness of 55, which has minimum yield strength of 539 Mpa. After the heat treatment, the rollers were finished with a grinder to remove the oxide layer so that the surface roughness Ra was measured as 0.4 μ m. The magnitude of the force between the rolling surfaces is set by the torque requirement and the coefficient of friction. The maximum torque requirement occurs in accelerating the telescope to slewing velocity and friction torque of bearing balls. For focal plane mechanism, the wind-induced torque is very small because it is placed in an immobile room. The maximum torque needed to drive moving parts is 615 N-m. The length of contacting cylinders is 50mm. Assuming a steel-to-steel coefficient of friction is 0.15, the contact force is 3154 N per sector, and the maximum contact stress is 213 Mpa, well below the yield point of the steel. The left space between the outside edge of big disk and top plate of frame for roller assembly mounting is limited as shown in figure 3 or figure 4, so the dishing springs are used to apply the normal force. This allows the roller assembly to be pressed to the disk surface to compensate for the surface runout of the big wheel. The torque motor unit is in line with the drive shaft by use of a high stiffness coupler. According the following formula, the compression distance of the springs f can be calculated depending on the require normal force F.

$$F = \frac{4E}{1-\mu^2}\frac{t^4}{k_1 D^2}k_4^2\frac{f}{t}\left[k_4^2\left(\frac{h_0}{t}-\frac{f}{t}\right)\left(\frac{h_0}{t}-\frac{f}{2t}\right)+1\right] \qquad (1)$$

Where: E is the Young's modulus for the dishing spring
μ is the Poisson's ratio for springs material
t is the thickness of single dishing spring
D is the outside diameter of spring; $h_0=H_0-t$, H_0 is the height of dishing spring

$$k_1 = \frac{1}{\pi}\frac{\left(\frac{c-1}{c}\right)^2}{\frac{c+1}{c-1}-\frac{2}{\ln c}} \qquad c = \frac{D}{d} \quad \text{d is the inner diameter}$$

$$k_4 = \sqrt{-\frac{c_1}{2}+\sqrt{\frac{c_2}{2}+c_2}} \qquad c_1 = \frac{\left(\frac{t'}{t}\right)^2}{\left(\frac{1}{4}\frac{H_0}{t}-\frac{t'}{t}+\frac{3}{4}\right)\left(\frac{5}{8}\frac{H_0}{t}-\frac{t'}{t}+\frac{3}{8}\right)}$$

$$t' = 0.94t \qquad c2 = \frac{c_1}{\left(\frac{t'}{t}\right)^3}\left[\frac{5}{32}\left(\frac{H_0}{t}-1\right)^2+1\right]$$

2.3 Bearings

The bearings used for the image field derotator should have the following characteristics:

- ✓ Large cylindrical bore to makes it possible for rotator to have enough inner holes to let the fibers and wires to go through to the end of the rotator.
- ✓ Self-aligning capability up to 1 degree to compensate for inaccuracies in co-axis of bearing housing mounting.
- ✓ Simultaneously bear radial and axial load and have high axial and radial stiffness.

An exotic bearing structure was proposed to satisfy the above demands, showing in Figure 6. We call it segment outer ring double-row spherical ball bearing (below abbreviated as bearing). It has two rows of balls and a common concave sphered raceway in the outer ring. The bearing is consequently self-aligning and insensitive to angular misalignments of the shaft relative to the housing. This angular misalignment can be accommodated without any negative effect on bearing performance. Additionally, this kind of self-aligning ball bearing has the lowest friction of all rolling bearings. In order to enhance the stiffness and increase running accuracy, the preload is applied to the outer ring. The outer ring is separated by a ring spacer whose thickness is determined by the demand preload.

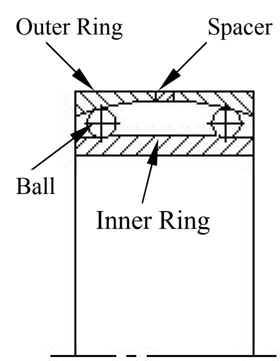

Figure 6 Sketch of bearing

According to the 3-D contact theory, the radial strain δ of ball and outer raceway under the radial force F_n is [7]:

$$\delta = \sqrt[3]{\frac{9F_n^2}{4E^2R}} \qquad \frac{1}{E}=\frac{1}{2}\left(\frac{1-\nu_1^2}{E_1}+\frac{1-\nu_2^2}{E_2}\right) \qquad \frac{1}{R}=\frac{1}{R_1}+\frac{1}{R_2} \tag{2}$$

Where: ν_1, ν_2 are the Poisson's ratio for contact materials
E_1, E_2 are the Young's modulus for contact materials
R_1 is the radius of ball; R_2 is the radius of outer raceway; F_n is the radial force

$$F_n = \frac{Q}{n\cdot\sin\theta} \tag{3}$$

Where: Q is the axial preload; n is the ball number per row; θ is the contact angle
So, under the axial preload Q, according to the above formulas, the machining of spacer thickness Δt is:

$$\Delta t = \frac{2\delta}{\sin\theta} = \frac{2}{\sin\theta}\cdot\sqrt[3]{\frac{9}{16}\left(\frac{R_2-R_1}{R_1R_2}\right)\left(\frac{1-\nu_1^2}{E_1}+\frac{1-\nu_2^2}{E_2}\right)^2\left(\frac{Q^2}{n^2\cdot\sin^2\theta}\right)} \qquad (4)$$

The weight of the rotator supported by the bearing, together with external force, such as fibers and cables, is 7.582 metric tons. The axial component force is 2.14 metric tons. The preload acting at the outer ring is set to 3 metric tons. According to the formula (4), the machining value of spacer thickness △t of the two bearings are given in Table 1. During the mounting, the axial preload is controlled by the torque wrench. The structure of this kind of bearing is shown in Figure 7.

Figure 7 Structure of bearing

Table 1 Machining value of bearing spacer

Parameters	Front Bearing	Rear Bearing
θ(°)	8	10
R_1 (mm)	20.68	23.82
R_2 (mm)	626.534	424.906
ν_1	0.3	0.3
ν_2	0.28	0.28
E_1 (N/mm^2)	2.1×10^{11}	2.1×10^{11}
E_2 (N/mm^2)	2×10^{11}	2×10^{11}
n	62	38
Q (N)	30000	30000
Δt (mm)	0.423	0.383

2.4 Encoder

The tracking accuracy of the image field derotation is 1 arc sec. The main problem for the encoder is linked to the large diameter of the rotator to be equipped, introducing a problem of both high- and low-frequency errors into the readings. A 572.63mm diameter grove was machined to the required tolerances, given by encoder producer Heidenhain Company, for installation of the tape on which the angles are read. The tape encode ERA 881C is chosen to detect the rotation angle and feed back the position to the control system. With the two readings and calibration curve, the accuracy of this tape encode is high enough to ensure the derotation precision. The mounting position precision of readings and tape meets the requirements given in the mounting manual by a special double screws adjustment mechanism as shown in Figure 8.

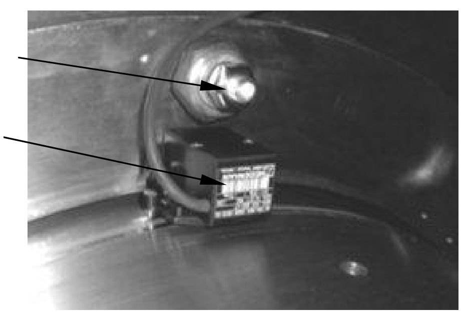

Figure 8 Encoder mounting

3. WORM GEAR DRIVE FOR ATTITUDE ADJUSTMENT

3.1 Summary

According to reference [4], the LAMOST focal surface will tilt a certain angle about the cross point of focal surface and light axis for observing different sky area in the plane formed by main light axis and focal surface light axis. During the observing sky coverage of -10°≤δ≤90°, the tilt angle changes from 0.44 arc minutes to 5.42 arc minutes. We call this motion as attitude adjustment of focal surface. The following parameters are the requirements of attitude adjustment:

- ✧ Attitude adjustment range: 0 ~ 10 arc minutes
- ✧ Attitude adjustment accuracy: +/- 2 arcsecs

3.2 Drives

Basing on the drive requirements and the actual conditions, the attitude adjustment drive system will have the following characteristics:

- The weight load driven by the drive system is about 12 metric tons, so the drive movement must be irreversible to ensure absolute safety.
- Just as Figure 1 shows, the total height of the focal mechanism is 6m given by the construction of pier and dome, so the space left for attitude adjustment mechanism is limited. The adjustment drive system should have a compact structure.
- Have large gear ratio possible in a single stage to minimize the motor size.

Worm gear drives are the traditional drive ways for most medium ground-based telescopes. Two of its advantages meet the above requirements: large gear ratio in a single stage and intrinsic mechanical irreversibility. A well made worm gear stage can be very accurate and smooth because there is significant averaging of the tooth to tooth machining errors. The worm and gear can be lapped together to reduce the machining and concentricity errors. As mentioned in requirements, the movement accuracy of attitude adjustment is 2 arc seconds, worm gear drive can achieve the performance in normal manufacture and carefully adjustment. So, worm gear drive is selected for the attitude adjustment.

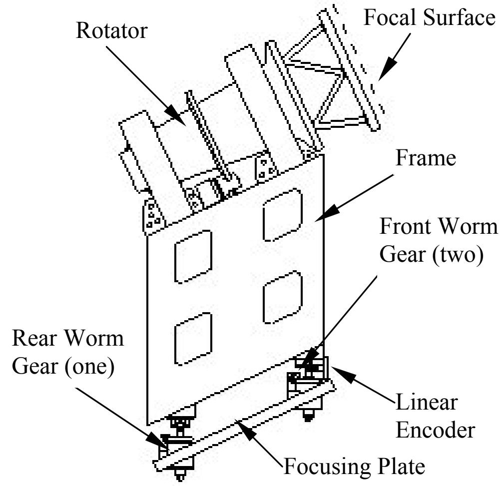

Figure 9 Schematic of attitude adjustment

According to the reference [4], the rotation axis of attitude adjustment is better to through the vertex of spherical focal surface and perpendicular to the plane formed by system light axis and focal surface light axis. But, considering of the configuration of LAMOST optical system, it is nearly impossible to arrange a rotation axis in the place of focal surface vertex. In order to minimize the blockage of light, the attitude adjustment structure is placed underneath the sloeplate of support frame, out of the light path, as shown in Figure 9. The attitude adjustment mechanism consists of three units of identical worm gear drive system, two units in front and one in the rear. The required tilting angle of focal surface can be achieved by the composition of high precision linear movements, guaranteed by 0.2 μ m precision linear encoder, of three drive units driven by motors with the coaxial encoders. After finishing the tilting movement, focusing along the system light axis is needed to position the focal surface in the required position. The detailed structure is shown in Figure 10.

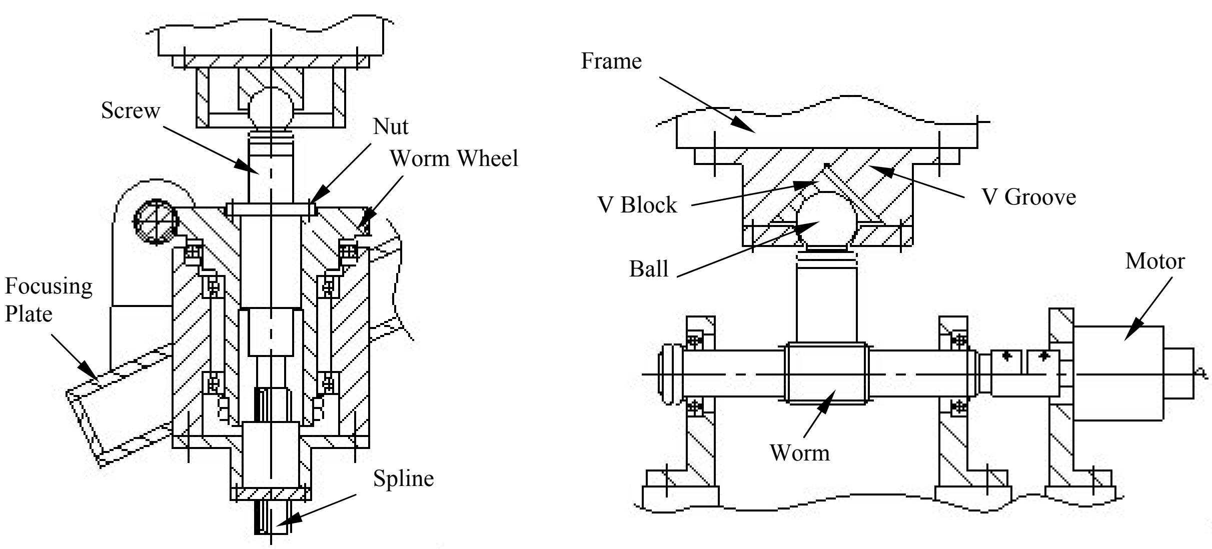

Figure 10 Drive assembly for attitude adjustment

Ball screws over the years have often emerged as the most efficient and cost-effective choice to convert rotary motion into smooth, accurate and reversible linear motion. When properly selected, a ball screw will minimize the possibility for positioning error. Depending on the application, either a rotating screw shaft or nut will then translate in a linear direction, resulting in minimal mechanical wear and ability for a ball screw to convert 80 to 90 percent of a motor's torque into thrust for optimized efficiency. Drive torque limits, lead accuracy, system stiffness and life are the main factors to consider for the attitude adjustment mechanism. Here, the backlash is not so much of an issue because the load constantly pushes down on the nut bolted onto the worm wheel, which supported by a thrust ball bearing and a pair of preloaded angular contact bearings. In order to increase the assembly's rigidity and stiffness, the preload is applied with an axial force applied to the split nut. The read head of linear encoder is attached to the screw shaft to detect the linear position and feed back to the control system.

In order to prevent back driving of ball screw, the end of screw shaft is machined as precision linear motion spline shaft according to the usage demands and usage conditions. The spline nut is bolted onto the bearing housing where the bearings and supporting worm wheel are mounted. Spline shaft is made of high-grade alloy steel quenching HRC 58, and spline nut is made of high-grade alloy cementite steel quenching HRC 58, so long service life and hard intensity can be obtained. Clearance of rotating direction is controlled to zero value.

During the movement, the position of three drive screws are fixed, that is to say, the mutual position dimensions are fixed. While the focal surface performing the tilting motion, for the rear drive unit shown in Figure 9, the V block will slide along the V groove besides the turn movement of ball on the ball socket required for three balls, as shown in Figure 10. For the two front drive units, the V blocks can not slide along the V grooves because their motion directions are at 120°. The velocity of these motions occured between balls and sockets, V block and V groove, are very slow. In order to prevent the low-speed stick-slip phenomena, the contact surfaces are submitted to a special treatment to enhance the surface hardness and decrease the friction coefficient.

4. BALL SCREW DRIVE FOR FOCUSING

4.1 Summary

Due to LAMOST exceptional structure configuration, the reflecting Schmidt plate (M_A), focal plane mechanism and spherical primary mirror (M_B) are situated on the top of three separate concrete piers which are structurally isolated from the building. It is necessary to do some adjustment along the light path after the focal mechanism has been mounted. On the other hand, due to the thermal influence and mechanical structure deformation, focusing motion is indispensable for the focal mechanism.

The drive specifications of focusing required by LAMOST project are as followings:

- ✧ focusing range: ±200mm
- ✧ focusing accuracy: ±0.01mm

Followings are the considerations to pay attention to during the drive structure design:

- ➢ Just as Figure 1 shows, focusing mechanism is mounted between the focusing plate and support base. The space is limited with the height only 178mm, so what kind of drive system to choice should be fitting to this space.
- ➢ Because the main optical axis of LAMOST lies in the plane of meridian and is inclined at an angle of 25° to the horizontal, the focusing motion must be performed along a slope plane. In such case, the load for the control system is not a constant in the up and down directions. A mechanism should be found to balance the extra load during focusing motion.
- ➢ High stiffness in needed during the long working stroke ±200mm.

4.2 Focusing drive system

In view of the above requirements, we choose the ball screw as the drive system of focusing. The ball screw drive structure and its accessories are shown in Figure 11.

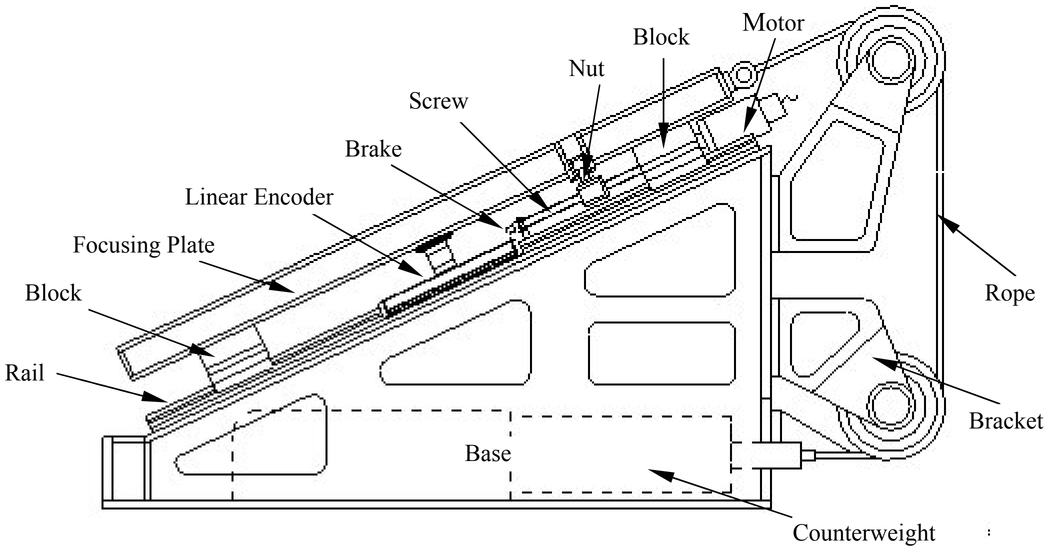

Figure 11 Focusing mechanism

For decades, ball screw is widely used for all kinds of industrial equipments and precision instrument because of excellent friction function and high positioning accuracy. Especially in recent years, as linear motion function machine parts, ball screw is widely used in machine tools. Ball screw assemblies can provide long stroke and efficient. Circuits of precision steel balls circulating in the groove between the screw shaft and nut results in minimal mechanical wear. But a good part of that promise hinges on the design and actual operating conditions.

The structure over focusing plate is supported from the base by two separate mounted linear motion guides composed of rail bolted to base and blocks bolted to the focusing plate. The guides are lying along the light path with 25° inclination to horizon. The weight of above structures is about 15.5 metric tons, with component force in guide axial direction 6.55 metric tons and normal to guide plane 14.04 tons. In order to get high performance in focusing, the imbalance force between up and down should be counterweighted. For most medium size telescopes, lead weight is a common way to treat the imbalance of the structure. But, here the way does not work. The followings are the main reasons:

- ✓ The volume of lead blocks to balance the 6.55 tons will be 0.58 m^3. There is no enough space to hold such big volume in the mounting site of focal mechanism according to the site construction.
- ✓ The focusing stroke is ±200mm, but the height of base is not enough for lead blocks to move even if using pulley block.
- ✓ The whole focal mechanism will move aside out of light path for optical checking as detailed in introduction section. The lead blocks will be swing during this motion, which is a big problem.

So, the constant force mechanism is a good way to deal with this problem. Pneumatic and hydraulic systems are the traditional methods to achieve a constant output force regardless of input displacement. But these methods require a power source and pollution will be introduced into the telescope dome. An innovative constant force mechanism, shown in Figure 12, is proposed to settle this issue. The constant output force is the composition of main compression spring force and auxiliary compression spring force. Additionally, the level of constant force supplied by the mechanism can be actively adjusted. Optimizations were performed to minimize the variation of the output force from constant-force by adjusting lever shape parameters.
A special mount tool was used to ensure two guides be parallel to each other and parallel to the main light axis. The latter is more important for focal

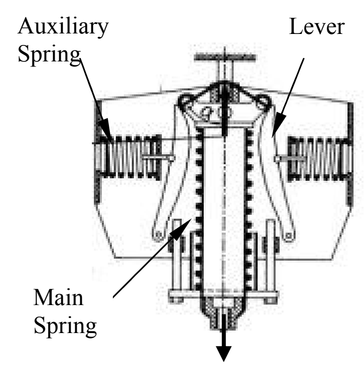

Figure 12 Constant force mechanism

mechanism because it has influence on the adjustment accuracy of focusing and attitude adjustment. The ball screw is left hand thread on the aft focal structure and is driven by a motor. After the structure was balanced well, the net torque to move the whole structure is only the torque required to overcome friction force between the guider contact surface, such as rail and block. Just as the ball screw used in attitude adjustment, in order to enhance drive stiffness, the preload is applied to the split nut and the screw is supported by a pair of angle contact bearings mounted back-to-back with preload applied to. Ball screw is only used to take the axial load. Radial force and torque let ball screw have a bad load such as surface contact force. It will damage the ball screw forever. So the axial line of ball screw must be parallel with axial line of linear motion guide and the three points of bearing seat of two ends and nut seat must be in same line. With high precision gauge and devices, all these requirements have been met well.

An electromagnetic brake is mounted at the end of screw to prevent reverse transmission as a result of dead weight after the motion stops or motor is out of electricity. A Heidenhain linear encoder is mounted along the guide to detect the position signal and feed back to the control system with the same mounting requirements as ball screw.

5. CHAIN DRIVE FOR MOVE ASIDE

5.1 Summary

According to the LAMOST configuration, the whole focal mechanism will be moved aside out of light path for optical checking. After checking, the mechanism will be moved back and the re-position accuracy is +/-0.02mm. There is no accuracy demand in-between start point and end point. The drive system and bearing system are under the base as shown in Figure 1. The dead weight of mechanism is about 22 metric tons with dimension 2.18m×2.6m×7m (L×W×H) and the center of gravity is 3m above ground, so safety is a very important factor to be considered during the design work.

5.2 Drive system

Considering the special requirements and using conditions, the subsystems of the drive are considered as followings:

✧ Bearing system

Air bearing, supplied by Hovair Systems, Inc., is chosen to support the structure in order to minimize the stick-slip introduced by difference between static and dynamic friction coefficients. The whole structure is lifted up by four square aluminum load modules and then driven by a small motor through a chain. The modules are permanently housed beneath the focal mechanism base via a mounting plate. The levelness and flatness of the top surface of pier where the air bearing will work are better than 1mm. In order to improve the working performance of air bearing, the top surface of pier is covered with two stainless steel plates, 2mm thickness and flatness is better than 0.5mm.

✧ Guide

The guide's role is to ensure the whole mechanism to move along a certain direction. It consists of two rails bolted to the pier like railways, guide rollers bolted onto the mechanism base. Another important role of the guide is to ensure the mechanism moving safely in case of emergency.

✧ Chain drive

Chain drive is rarely used anymore on observatory class instruments now. But, here it is a very cost-effective drive system because it is very easy to implement since precise alignment between components is not critical and it is very affordable compared to other types of drive systems. Even though the dead weight of the whole mechanism reaches 22 metric tons, a small motor can drive the moving parts easily because of the low friction force of air bearing. The motor does not work until the four sensors mounted near the four modules show that the mechanism is lifted by the air bearing. There are some position switches mounted in the start point, stop points and end point to ensure the mechanism stop in the right positions. After finishing the optical checking, the whole structure will back to the original position. The requirement of position accuracy is 0.02mm, which is achieved with the help of three sets of locating device.

6. TESTS AND CONCLUSIONS

The tentative assembly of the whole focal mechanism, including the four different drive mechanisms, has been finished in the workshop of Nanjing Institute of Astronomical Optics and Technology (NIAOT). The purpose of this on-shop assembly is to minimize the assembling errors, to test the drive performance and to review the technics processes of final assembling on the site. The test data obtained so far suggest that the performance of these drive systems achieved the design targets.

- ✧ For image field derotation, the tracking accuracy is 0.45 arcsec（RMS）at the range of +/-22.5 degree at the different velocity from 0 to 15 arcsec per second. The accuracy is 0.69 arcsec at the speed of 1 degree per second at the +/-10 degree rotation range, and the accuracy is 0.35 arcsec at the speed of 10 arcsec per second during the 10min tracking time.
- ✧ The position accuracy of attitude adjustment is 0.24 arcsec in the adjustment range of 6 arc minutes. This value is calculated according to the mechanical dimension and position accuracy detected in the attitude adjustment drive units. The real position accuracy value of the focal plate will be little larger than 0.24 arcsec because of the elastic deformation of mechanical structures.
- ✧ The accuracy of focusing reaches 2 microns in the focusing range of +/-50mm.
- ✧ With the help of three high accuracy V grooves to position the support V blocks, the position accuracy of the moving aside motion is better than 0.02mm. The main factors to influence the re-position accuracy are the gravity deformation and manufacture errors of position units.

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REFERENCES

1. Shou-guan Wang, Ding-qiang Su, Yao-quan Chu, Xiang-qun Cui and Ya-nan Wang, “Special configuration of a Vary Large Schmidt Telescope for Extensive Astronomical Spectroscopic Observation”, *Applied Optics*, pp. 5155-5161, Sept. 1996.
2. Ding-qiang Su, Xiang-qun Cui, Ya-nan Wang and Zheng-qiu Yao, “Large Sky Area Multi-object Fiber Spectroscopic Telescope (LAMOST) and its Key Technology”, Advanced Technology Optical/IR Telescope VI, *Proc. of SPIE*, Vol. 3352, pp. 76-90, Jun. 1998.
3. Ding-qiang Su, Ya-nan Wang, “Tracking movement of the Large Sky Area Multi-Object Fiber Spectroscopic Telescope (LAMOST)”, *Acta Astrophysica Sinica*, Vol. 17 (3), 315, 1997.
4. Ya-nan Wang, “Dynamic Simulation of the Actual Observing Process of LAMOST”, *Acta Astrophysica Sinica*, Vol. 20: pp.9-15, 2000.
5. Xiang-qun Cui, “Preliminary requirements on focal plane of LAMOST”, LAMOST Report, Jan. 1996.
6. Guomin Wang, Lisheng Ma, Zhengqiu Yao, Guoping Li, “Experiment Study on Friction Drive”, *Proceeding of SPIE* ，2004，Vol. 5495: p419-428
7. Jian-feng Zhang, Zhi-fang Zhou, “Friction, Wear and Anti-wear Technology ”, Tianjin Science and Technology Press, May 1993
8. Yao Zheng-qiu, Li Guo-ping, “ The Tracking System of LAMOST Telescope”, *SPIE* Vol.3352, P560-571
9. Wang Guo-min, Li Guo-ping, Cui Xiang-qun, Yao Zheng-qiu, “Finite Element Analysis and Optimization of LAMOST Focal Mechanism”, *MIE of CHINA*, 2004, Vol. 33(7): P91-93,96