Assemblies for Parallel Kinematics

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Joints and struts are important mechanical assemblies in machine designs that are based on parallel structures. These assemblies are the critical components that determine the precision and performance capability of such machines.

1 Introduction
Two years ago, industrial sources claimed that parallel kinematics technology might well be the future of the machine tool. Very few people thought the new concept had a chance, but now it seems that no one wants to be left out. The future is already here!
A development like this has also been a challenge for manufacturers of machine components. The rolling bearing industry recognized this two years ago and drew a lot of attention when the first of the new components were introduced. It began with developments in mechanical joints and was followed by the design and production of telescopic strut assemblies. Currently such components are being tested by INA applications engineers along with engineering partners from universities and the industry. Besides practical tests, extensive testing is underway to validate performance data. On the basis of the positive results recorded so far, INA is presenting production-ready standardized products now. These products will surely have an effect on the development of parallel kinematics.
2 Heavy demands on joints
Joints are not new. They are standard components in both automobiles and agricultural machinery. However, the requirements in these areas are quite different from those in machine tool design and construction. In the context of parallel kinematics, the requirements that joints must meet are:

- high rigidity
- high static load-carrying capacity
- long service life
- pivot angle suitable for the design
- low mass
- high precision
- clearance-free bearing arrangements, preloaded systems
- smooth running, stick and slip free
- defined, measurable joint crosspoint
- low wear.

For tripods and hexapods, a parallel structure requires joints having both two and three degrees of freedom. Both versions were thus developed in each of the respective joint types. The specific requirements of different application areas led to further types. For example, a joint in a cutting machine must have maximum rigidity and precision. In handling technology and in cutting processes, large workspaces are traversed very quickly. This means large pivot angles and low joint mass.

Three different joint designs have been developed to meet these requirements. These will be discussed below.

2.1 Ball joints with three degrees of freedom
Ordinarily, ball-to-ball contact is a poor rolling-contact match in terms of surface loading. However, in the course of prototype development, this yielded highly favorable conditions in this application (Fig. 1).

Because a large number of small balls are used, the Hertzian pressure between the cup and the rolling element and between the rolling element and the inner ball remains low. However, favorable conditions can only be utilized if the geometric accuracy of the contacting ball surfaces is very high. Testing allowed manufacturing processes to be improved, and good results were achieved in terms of both shape and surface quality. Besides a high load rating, rigidity of joints is absolutely essential in a cutting machine. Despite point contact, high rigidity is achieved under preloading because of the favorable load distribution.
Of all joints, the ball joint offers the best ratio of load carrying capacity and rigidity in terms of design space. The surrounding seal carrier provided a complete sealing effect. With these characteristics and with pivot angles up to about 20°, this joint will mainly be used in cutting machines and heavy handling equipment.

2.2 Universal joints with two or three degrees of freedom

The universal joint (Fig. 2) is ideal for applications in the handling sector. Its low mass and large pivot angles allow structures to be designed that are subjected to high accelerations and speeds in large workspaces. In order to keep design size and rigidity within an appropriate range, the pivot angle was limited in the end positions (Fig. 3). The pivot angle diagram shows the permissible pivot positions of the axes to one another. Because of small support width and the use of angular-contact needle roller bearings, rigidity values are significantly below those of the ball joint, despite preloading. However, they are sufficient for the applications mentioned above. The advantage clearly lies in the low 2.7 kg mass and the large pivot angles.

2.3 Cardan joints

Cardan joints usually serve to transmit torques and to offset misalignment in shaft connections. In parallel structures, tensile and compressive forces must be transmitted and high rigidity must be ensured. For this reason the spider was optimized for tensile-compressive loading by a finite element program. In particular, the spider and the yoke was improved. The rolling bearings to be used are preloaded axial-radial needle roller bearings. These bearings offer the greatest possible rigidity relative to space, are completely sealed and represent the technical standard in the rolling bearing sector.

The cardan joint closes the gap between ball joint on the one hand and universal joint on the other. At very high rigidities, this joint permits large pivot angles. Also note the pivot angle diagram in Fig. 3. Its limitations in relation to the relative movement of axis I and axis II are also apply to the cardan joint. This makes the cardan joint ideal for applications in which large workspaces are traversed and rigidity is required (Fig. 4).
3 Telescopic struts

Two years ago INA presented the first telescopic struts as machine components at the EMO in Hanover. In contrast to the struts machine builders had developed for their own projects, these were the first struts to become commercially available. After further design enhancements, two diameter sizes are now available. Starting from the basic dimensions, the design is matched to customer requirements with regard to stroke path, rigidity and the reactive forces expected. Stroke length, the second suspension point and the type of joint are also variable and can be designed to meet customer requirements.

The functional structure of a telescopic strut, with outer tube and inner tube, is similar to that of a hydraulic cylinder. To ensure clearance-free guidance of the inner tube during extension, a preloaded linear guidance system is provided in the outer tube and patented, embedded INA KUVS raceway in the inner tube. Depending on rigidity requirements, feed drive is either a ball or roller screw drive. The drive spindle has a DKLFA series thrust angular-contact ball bearing.

At a maximum speed of 2000 min⁻¹ and spindle pitch of 20 mm, feed speeds of 0.8 m/s are possible (Fig. 5).

Stresses on the platform are converted to tensile-compressive loads by the joint kinematics. A transverse force on the linear guidance system occurs only because of the accelerated mass of the telescopic strut.

The decisive rigidity factor is therefore the rigidity of the axis of the telescopic strut. Rigidity is thus a function of stroke length when the diameter remains constant.
4 Determining rigidity

Joints, particularly ball joints, can be compared to rolling bearings by calculating characteristic values, but only to a limited extent. Calculation methods must first be developed in order to determine reference values for the design. These theories are then tested for plausibility and further developed.

To determine rigidity values, the joints were calculated using a finite-element program based on the 3-D CAD model. For cardan and universal joints, rigidity here appears as series/parallel switching of the individual components. The bearings behave like ordinary rolling bearings. Accordingly, deflection is clearly broken down into rolling element compression and the deformation of the spider.

Determining the rigidity in ball joints is considerably more difficult. Basically, a finite-element model is also generated here. The rolling elements are replaced by non-linear springs, and the geometrical accuracy is captured by means of various spring rigidities (Fig. 6). This is a justifiable procedure since the deviation from the ball shape nearly always occurs in the same area as a result of the process. This modeling makes it possible to include in the calculation the inaccuracy of the components with respect to geometrical accuracy.

In both the universal joint and the ball joint there was a good correlation between calculation theory and the test. Figure 7 gives a comparison of the rigidity values calculated for all joints.

5 Load ratings and rating life

The determination of load ratings in cardan joints and universal joints is based on classic rolling bearing theory. Finite-element calculations have demonstrated that the connecting components are not critical elements. For these types, a load rating can thus also be defined by the load rating of the weakest rolling bearing mounted. As for the determination of rating life, an analogy to oscillating bearing movements can be seen here, i.e., there is no fatigue in the conventional sense.

The best characteristic number for quantifying probability of failure is the static load safety factor. Because of the high bearing load ratings, in normal applications a static load safety factor of at least four can be ensured. Empirical values indicate that there is little tendency for the part to wear at these levels. These theoretical issues will be verified in the test.

Ball joints are a totally new design, and it is hard to compare them with rolling bearings. This type of joint is not a conventional rolling bearing type. INA is currently preparing calculation procedures for ball joints.

If a ball joint is pivoted, the roll-off movement of the rolling element occurs only at the pivot level. Outside this level, i.e., the case of about 95% of rolling elements, there is a movement about all three axes. Also, because of the full complement design, the effects of the balls on each other must also be taken into account.

In rolling bearing technology, a load that occurs in this joint, i.e., a pivoting movement with small pivot angles under load, cannot be calculated by conventional means. Long-term testing must be conducted to determine the fatigue behavior of these elements. Conclusions concerning possible calculation procedures can be drawn from the test results.
6 Lubrication
Swiveling motion or micro-oscillations in rolling elements place special demands on the lubricant. Greases designed especially for this application can reduce the wear mechanisms commonly found in these applications. More testing must be performed in order reach final conclusions on whether fretting corrosion can be eliminated if special greases are used. Using protective coatings on critical parts is another means of preventing fretting corrosion. Large-scale testing is also underway in this regard.

7 Conclusion
With the development of joints and telescopic struts for parallel kinematics machine components for this new generation of machine tools and handling equipment are now available to the designer. Components have been designed to meet the great number of demands placed on joints and telescopic struts. As for future developments, the results obtained from various research projects such as Dynamil II or from collaboration with customers will point the way to new directions.

Because of constant new developments, it is absolutely essential for design engineers to work closely with the manufacturer when using these new machine components.

Further information on implementation, design criteria and the state-of-the-art of development can be found in the detailed publications supplied by rolling bearing manufacturers or may be obtained by contacting their Application Engineering Service.

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