A TELESCOPIC JIB FOR CONTINUOUS ADJUSTMENT

By C. Ch. Etzler

Dornier System GmbH

ABSTRACT

For special space applications, e.g. for experiments distant from any orbital platforms or manipulators a new kind of jibs with extreme extension capacity has to be designed. Considering the requirements the telescopic principle is found to be the most promising. For the choice of the stiff structure, design criteria are evaluated. Special effort deals with the drive system. An electromechanical system can satisfy the requirements. First results of the development of such a drive are presented. The most significant features are: A telescopic assembly of tubes which can be mutually moved by a short spindle in the centre of the package. An elastically suspended screw is located at the bottom of each tube. For the jib extension these screws will be linked with the spindle. The control of their sequence and the adjustment of tubes in mutual end positions are performed by latches. A functional model proved the basic idea.

INTRODUCTION

There are fields of activities in space, where the application of extendable jibs is necessary. One field is signified by the requirement to perform experiments in a certain distance of the space shuttle and other orbital platforms. The physical reason is to be free of all kinds of disturbances caused by the platform. Distances between 30 and 50 meters are in discussion as being satisfactory. In other cases the establishment of special geometric positioning of experimental objects is desired. These targets have to be attained without introduction of intolerable inherent jib-dependent new disturbances, and within narrow tolerances for deviations. The other field refers to the construction of large structures in space. In this case the jib shall have the capacity of being the integral part of a manipulator which partly fulfills the task comparable to the function of a human arm amplified in range and force. The jib shall be easy to handle manually or automatically.

Up to now the extendable jibs are not constructed for general application and do not satisfy the whole requirements above circumferenced. This paper presents the characteristic items of a development we are persuaded to content an answer to the challenge set by the aims of space technology. The paper refers to an extendable jib which incorporates the advantage of tailored booms to be optimal in weight and alignment accuracy with the flexibility required for use at manipulators.

DESIGN REQUIREMENTS

For application within the development of the extendable jib the generally formulated requirements given in the introduction have to be formulated precisely to get their rationally provable demands. Signified by head lines these requirements are for the experimental application case:

- large extension range (30 through 50 m),
- large positioning flexibility (continuous extension) with narrow-hold mode tolerances ($< |\pm 2.5$ cm$|$),
- guaranty of alignment within narrow tolerances ($< |\pm 1^{o}|$),
- tolerable dynamic response properties (vibration levels of launch, attitude control loads by vernier and primary thrusters of the supporting platform, preliminarily the space shuttle) regarding the jib strength capacity, the extension functions and control efficiency,
- small storage volume (mainly aspired to be smaller than one space shuttle pallet length) and weight (less than existing — and under restrictions applicable — competitive systems),
- clean environment (no gas, debris or contaminating fluids which could falsify experimental results or hamper their functions,
- easy adaptation to payload interfaces (e.g. development goal 200 kg payload).

The design requirements for manipulators are partly equal, partly not yet animated by final figures. The differences influence sizes or special parameters, e.g. in the case of experiments it is possible to wait up to the elastic vibrations are damped down;for manipulators one desires a behaviour which is nearly equal to the aperiodic boundary case. Nevertheless these differences do not make questionable the main design principle for jib extension.

CHOICE OF JIB STRUCTURE

For unfolding mechanisms several structural principles have proved their capability: the two hinges and the multi hinges booms, the boom built up by stiffly erected continuous material cords and the telescopic booms. Already for reason of the volume demands the first principle has to be excluded. By means of a design based on the second and third principle it is difficult to get the desired stiffness and alignment independently from the extension status. The only principle promising to cover all requirements is the telescopic structure.

The european GEOS satellite was equipped with two pairs of telescopic booms. Structural layout ideas could be expanded to general application. But for their drive system a new mechanism had to be developped to become able to fulfill the requirements. Several reasons (simple power supply, tolerable disturbance of the environment, short and direct methods for control operations) urge to use electric motors for energy feed in.

The conversion of the motor revolution into mutually telescopic tube linear motion is carried out by means of a spindle drive. The advantage and reason for choice is the short way for drive torques in motion and hold phases, e.g. if compared with rope drives. Weight saving reasons favour reception of the drive system in the tube assembly core. Some guiding and latching mechanisms have to work together with the drive.

In the following the whole system will be described. Fortunately this can be done separately for the main functions:

- procurement of stiff structure,
- drive,
- guiding, and latching for hold phases,
- control,
- power and signal transfer.

The two latter functions are only mentioned for completeness and not discussed in detail.

THE STIFF STRUCTURE

The stiff structure consists of an assembly of tubes with different diamter (Fig. 1), which are mounted into one another. They put together nearly fully in the stowed condition and only partly up to a final overlap length in unfolded conditions.

The tubes do not have necessarily a circular cross-section . The triangle too is an interestic shape with some advantages for guiding. The circular shape presents advantages for manufacture and is chosen as a basis for design estimations.

For constructional reasons, namely to mount the drive mechanism, the smallest tube should have a diameter not smaller than 60 mm. Advantages for manufacture can be deduced if the increase of diameter is performed by constant steps, thus getting constant gap widths between the tubes to place identical latching mechanisms within. The ratio of the smallest to the largest diameter (i) can be optimized with regard to the desired response behaviour, assuming a constant ratio between inner and outer diameter (α) for each tube. The ratio i is bigger than tolerable from the manufactural view point. Therefore a constant (α) policy for all tubes is not practicable. But this influence is not so important that for further estimations this aspect will be expanded here.

The design goal is to get a sufficiently stiff construction with minimum of weight. Representative for stiffness is the deflection (δ) under static loads. To describe the dependencies Fig. 2 has been drawn. The abscissa presents the acceleration environment at the space shuttle spread over the range from vernier thruster to primary thruster pulses. The vernier thruster pulses may induce a boom oscillation with amplified amplitudes within the first half of the described range. The graphs are valid for a construction with one kind of material, constant wall thickness ratio (α) and constant resulting deflection (δ). Additional deflection caused by backlash is not considered. The graphs describe weight expense, readable at the ordinate of the figure. The additional weight by overlapping of the adjacent tubes and by the drive mechanism is considered. Competitive materials are only aluminum alloy and CFP.

From the Fig. 2 it can be deduced furthermore the tube wall thickness has a tremendous influence on the total weight. It should be as small as possible and as far as the pertinent increase of diameter allows. Further it can be deduced that the increase of stiffness by the factor of ten means increase of weight by the factor of three. In this situation the aluminum alloy boom can be replaced by a CFP construction gaining half of the weight.

In Fig. 3 the design goal stiffness is checked against strength, resp. manufactural and constructional view points. The diagrams demonstrates that there exist lower and upper limits for the jib diameter. The lower limit is bounded to a wall thickness smaller than it would be practicable for a plain CFP shell. This suggests a network construction seems to be adequate for the material.

Concerning the upper boundary more detailed investigations are necessary.

An important viewpoint is the thermal stability. At CFP designs by its special properties this is reachable with special variation of the directions of the fibers. In case of aluminum constructions it may be necessary to perforate the boom, mainly at tubes with small diameter because the temperature dependent deflection is inversely proportional to the tube diameter. But this technology is bounded to weight expense.

Not included in the description of basic design aspects and parametric relations but necessary to investigate are local reinforcements at latching points and guiding tracks.

THE DRIVE MECHANISM

Two elements of the drive mechanism have been already mentioned: the motor and the spindle. The spindle has a threaded and an unthreaded section. In Fig. 1 this section is located between the motor and the upper spindle part. Further elements required for the drive are screws elastically suspended at the bottom of each tube and finally keyway rails at the shell surface of each tube to prevent mutual rotation of the tubes.

In the stowage mode, when the tubes are nearly completely fitted into one another, the screws are stringed up over the unthreaded store – section of the spindle. Only the uppermost screw is thrown into gear (Fig. 4).

The deployment mode starts with shift of the upper screw along the spindle, when it starts to turn. To prevent the other screws and therewith the other tubes to follow these tubes are latched with one another by pins. This will be described in detail later.

Shortly before the moving screw leaves the spindle a skewed ring of the moving tube comes into contact with a counter ring fixed at the upper end of the neighbouring outer tube (see Fig. 5). Now this tube is urged to leave the store. After its bottom screw has been thrown into gear the bottom screw of the preceding tube leaves the spindle not before the coupling between both tubes is performed.

The coupled tubes are now together the moving part. The described sequence from the start of deployment will be repeated up to fully boom extension. At any longitudinal extension is a stop of the motion (hold phase) and also a reversal of motion possible.

In the totally extended condition the holding phase is signified by the fact that all screws are outside the spindle without one. The reachable position of the last screw depends on the length of the container tube. For direct coupling of the last moving tube with the container tube the screw should reach the upper end of the spindle to release the coupling mechanism as in the cases before. Then the most outer (container) tube has to have nearly the block length (L_B) of the telescopic boom, whilst all other tubes have the length $L_B - (n-1) \cdot L_S$. L_S is the thickness of the bottom sections with the suspended screws and assumed to be equal for all tubes and n is the number of moving tubes. The described construction has the advantage that in the most unfolded condition the spindle is eased from loads. These have to take the direct path through the tube walls.

In Fig. 6 our functional model is sketched, all tubes having the same length. The desired end positioning of the last screw is maintained by loss of overlapping between the last two tubes, which is not a preferred design. Loss of overlapping means loss of stiffness and alignment accuracy.

The choice of the motor for the drive system depends on the awaited motion performances. Those are not finally fenced in, therefore this problem will not be discussed here. The choice of the coupling between the motor and the spindle and the supporting of the spindle which has to be suited and therefore different for operation and launch, has been excluded too.

THE LATCHING AND GUIDING DEVICES

The control of the sequence of motion and the blocking of the tubes in mutual end positions are performed by latches and pins. In the stowage mode each tube is blocked with the neighbouring outer tube by at least three pins (see Fig. 4). With exception of the pins of the most upper tube all other pins are blocked by additional pins which dive from the bottom of each tube into the bottom section of the adjacent tube.

During the deployment mode the lifting bottom pulls its blocking pin out of the next resting bottom. There the fixation of the coupling pins is solved and the resting tube can be set into motion as soon as a force along side the resting tube is high enough to overcome the lateral forces at the coupling pins from the skewed resting hole borders. The force required will be set free, by the drive system and introduced into the resting tube by the above mentioned contact rings (Fig. 5).

The amount of the force has to be significantly larger than friction between the moving tube and the resting tube, accelleration forces gathered directly from the single resting tube.

For the last (bottom section) coupling pins a suitable number of resting positions (n—1) in equal distances along side the container tube has to be provided. This is necessary to warrant the blocking and unique positioning of the tube packages.

The coupling of moving tubes is performed by latches, three for each bottom section as visible in Fig. 7. A skewed plug of each latch is pushed into appertaining holes of the neighbouring outer tube by a compression spring. This latching can occur as soon as latch pins give the latch free to rotate. These pins on the other hand are blocked by the spindel. Only when the bottom section with the pins is outside the spindle length the pins release the latch.

For the reversal,that is the opening of latching,the spindle head cone has to penetrate the pin section and to push the pins outside. Then the pins will turn the latches back with drawing their plugs out of the resting holes.

The latching pins are mounted in the same bottom section as the coupling pins. The latching pins serve only for one task. In the case of booms with a sufficiently stiff structure the coupling pins can be used in addition for realization of a jib free of backlash, bridging the necessary gap to prevent jamming between two tubes by spring elasticity. But it may be required to solve the task by means of different elements. And this has to be handled together with the bridging of the gap between the stop rings. The problem has to be handled very cautiously because all failures will be augmented by the number of tubes.

A final remark has to be made referring to the keyway rails which prevent the mutual torsion of the tubes. Because only in maximum two rails are in relative motion simultaneously their tolerances and gaps between their keyways induce no problems for alignment. The locked tubes can be hold in unique positions thus creating no additional geometric deviations.

INTERFERENCE OF REQUIREMENTS AT THE SPINDLE

The topics presented up to now were functional components built up by sets of structural elements. These elements have to be chosen appropriately. The spindle is an example.

If the length of the fully extended boom is specified then the length of the spindle depends on several parameters. At first there is the number of tubes and this number (n) depends on alignment requirements ($\Delta\delta$), the desired stowed boom block length (L_B), on the tubes overlapping length (L_U) and the tube bottom section thickness (L_S). The last parameter interferes with the thread pitch. The elastically suspended screw should have the possibility of performing the elastic motion equal to two spindle revolutions in both axial boom directions. This drive at a low pitch number. The necessity for self locking if the motor gear does not perform this work in intermediate boom positions also claims to a low pitch number.

The requirement to minimize weight asks for a small number of tubes. Finally dynamic response estimations as well as aspects of strength have to be considered. In Fig. 8 the main influencing parameters are gathered. The abscisse of the diagram refers to the number of tube elements. The ordinate refers to ratio between the induced alignment error and the diameter tolerances. The latter are the reason for alignment errors. The lines in the diagram are signified by constant overlapping length, constant block length, or constant spindle length.

It is visible that the overlapping length has a very significant influence on the alignment. Better alignment calls for more overlapping. This in addition has as result a larger block length. Only with a special number of tubes a minimum of alignment errors is reachable.

A realistic compromise for greater 0.8 m diameter booms will be 3.5 m block length, 17 tubes and 2.5 m spindle length (threaded part). For reason of the oscillation behaviour this spindle should have about 25 mm diameter. Then the amplification of boom dependent inputs for the spindle will be in any case smaller 6 dB.

CONCLUSION

A summary of important design details of the telescopic jib has been presented. The estimation of the interference of all details is not closed. By means of a functional model it has been proved that the drive and latching mechanism works satisfactorily. Further efforts therefore will be to develop a jib based on the initial success and experiences. Thus mating by the presented estimations the initial design idea with the required properties. The main steps will be:

- guaranty of latching system reliability,
- increase of the spindle/screw efficiency,
- development of a stiff tube structure.

The first two tasks will be signified by a lot of test activities.

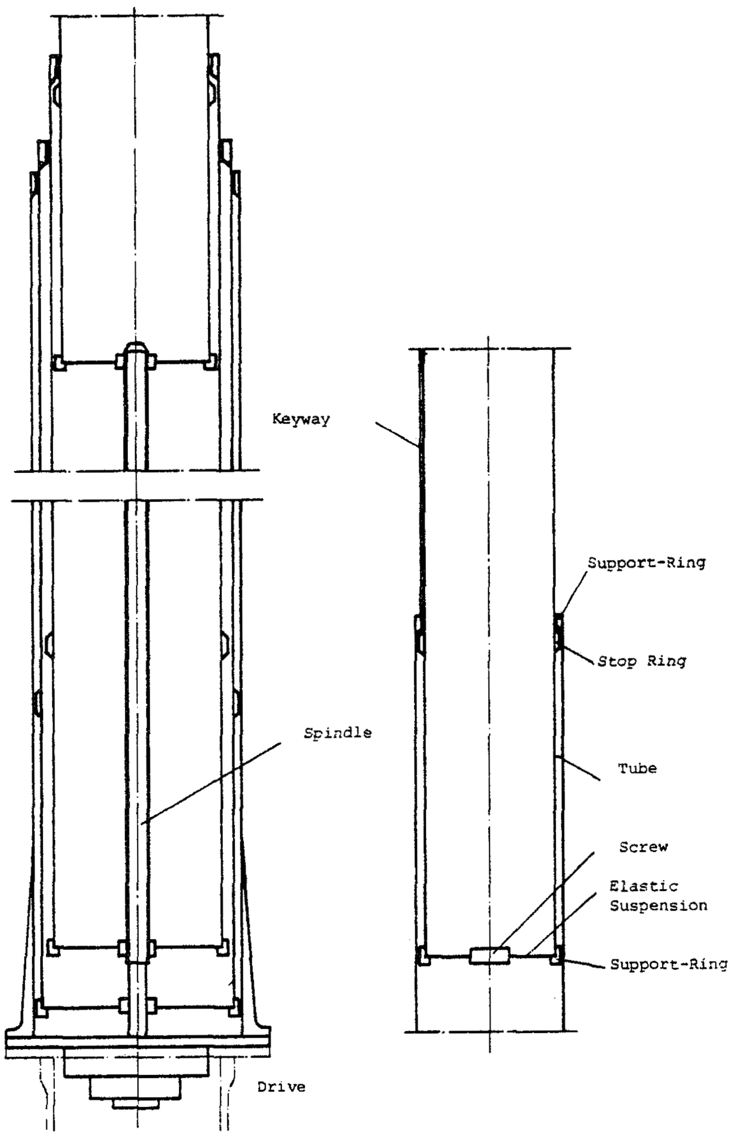


Fig. 1 : The Main Elements of the Telescopic Jib

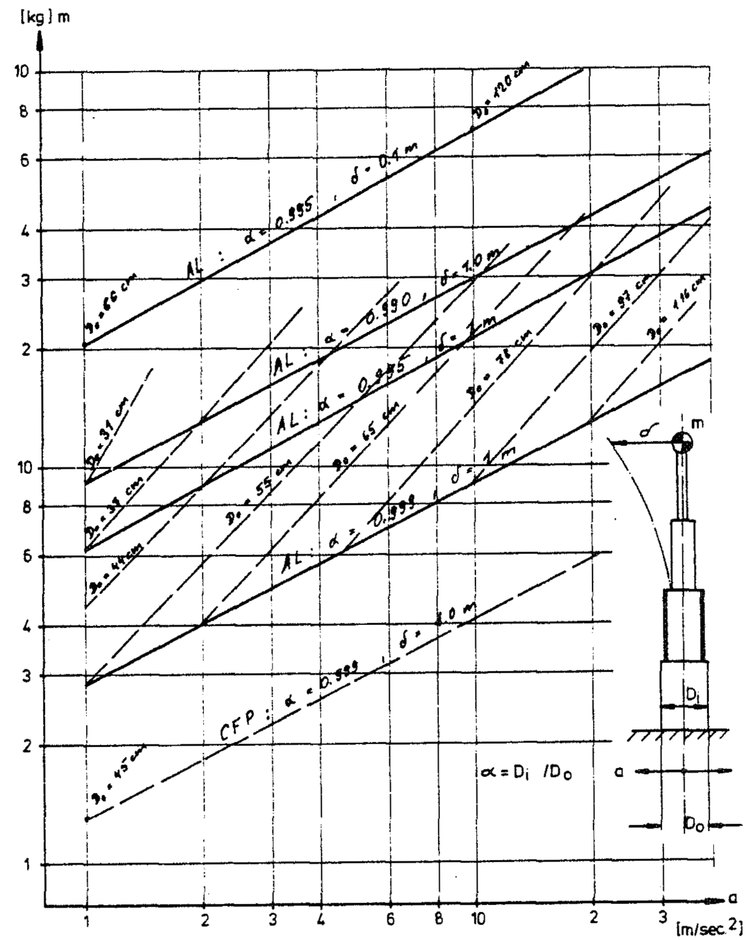


Fig. 2 : Telescopic Jib Dimensions and Weight Depending on Stiffness and Lateral Load

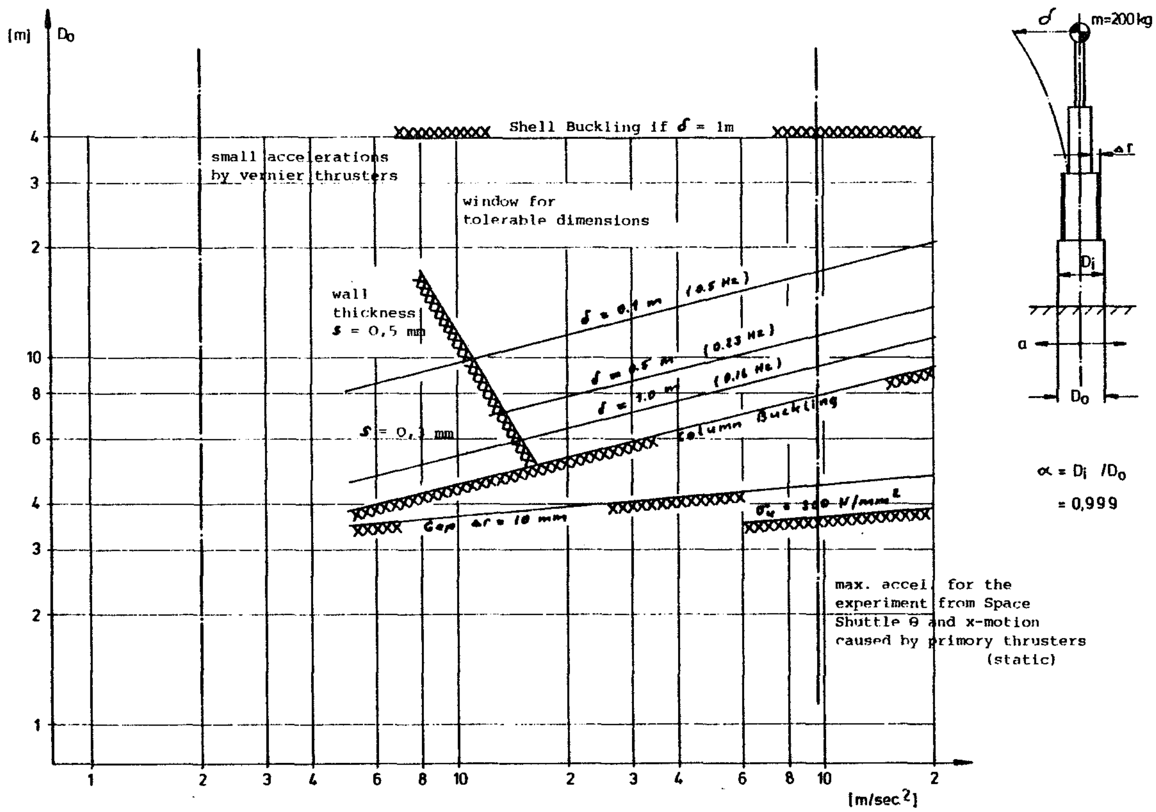


Fig. 3 : **Limits for the Tube Diameter Choice of a 30 m Jib**

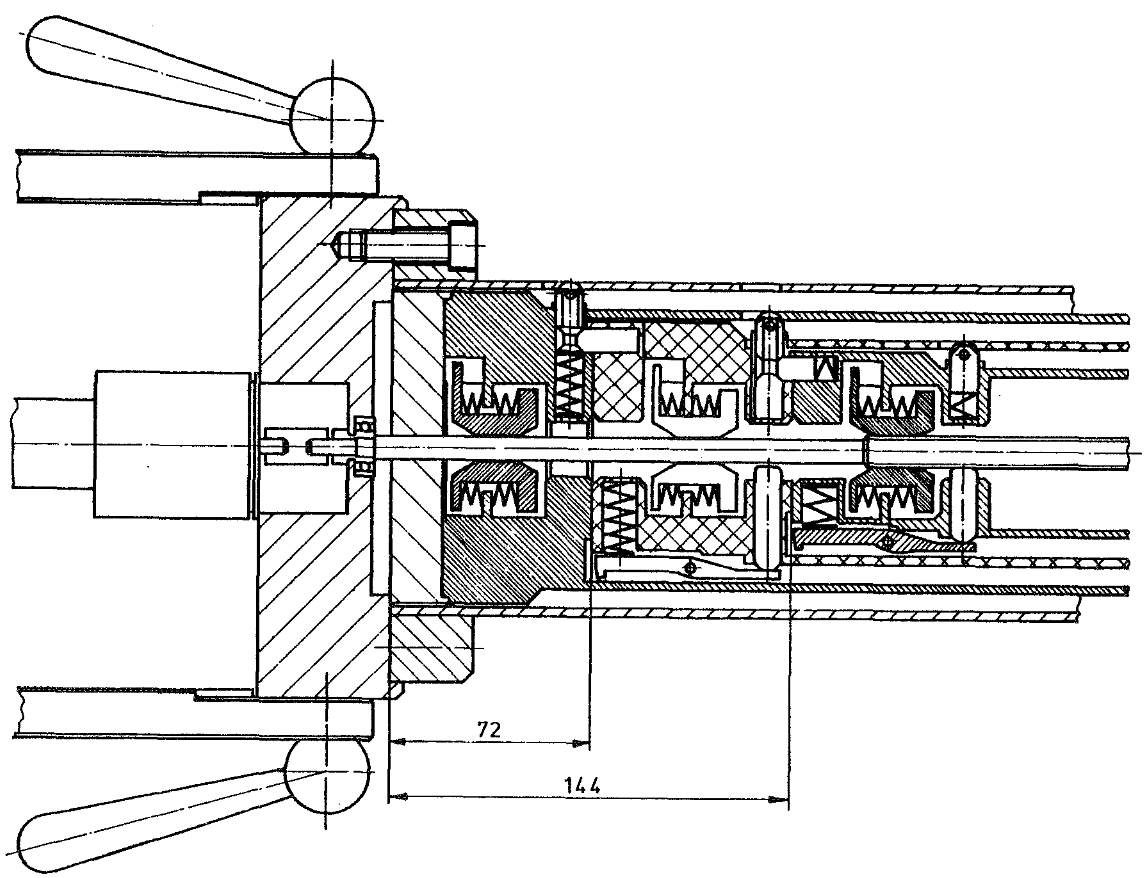
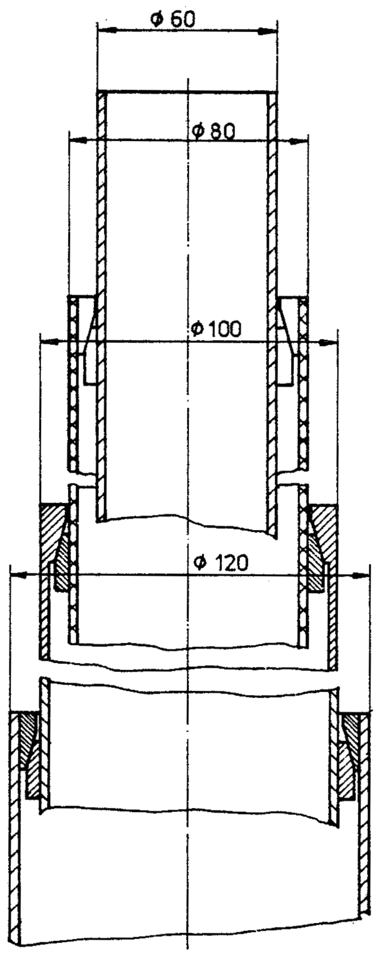


Fig. 4 : **Storage and Release Mechanisms**



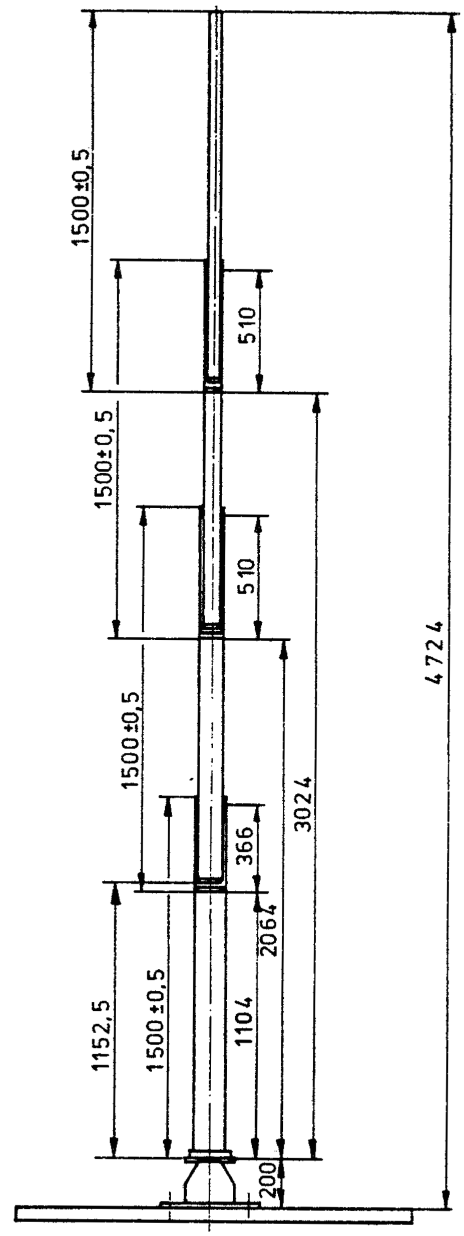


Fig. 5 : **Limit of Relative Motion by Stop Rings**

Fig. 6 : **Functional Model Full Deployed**

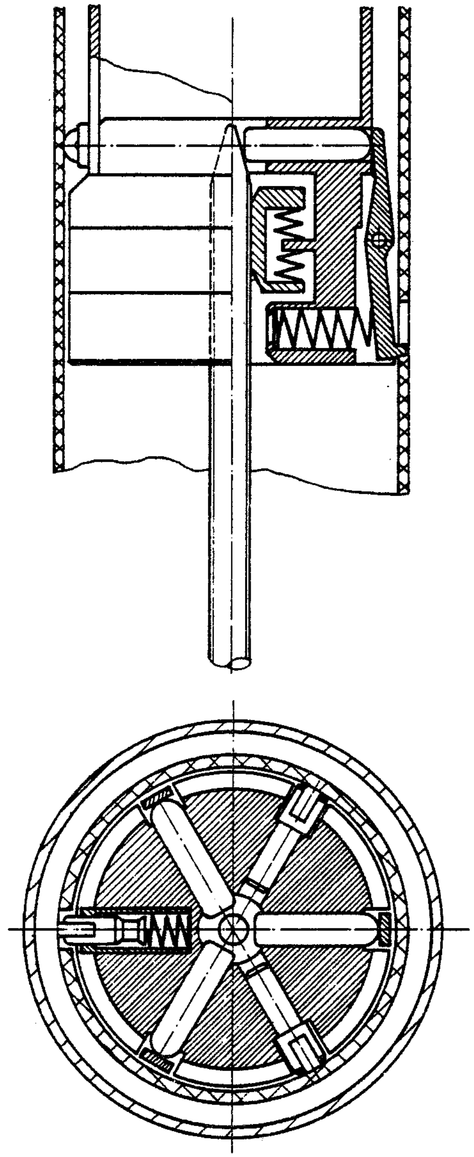


Fig. 7 : Clutching of Boom Tubes

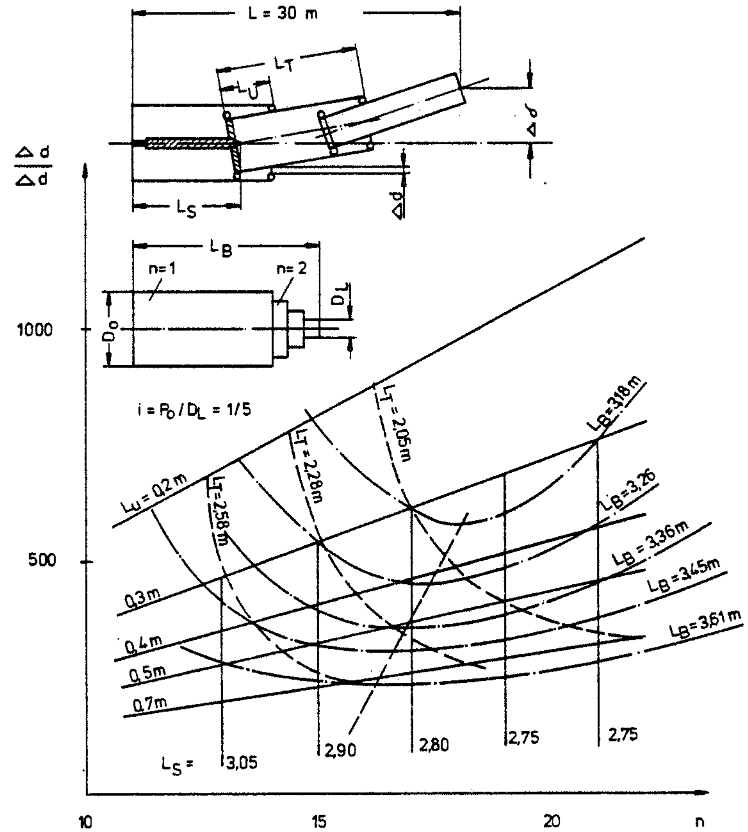


Fig. 8 : Fingerprintdiagram for Choice of Spindle Length