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THE DESIGN CONCEPT OF THE 6-DEGREE-OF-FREEDOM HYDRAULIC SHAKER AT ESTEC *

P.W. Brinkmann
ESA/ESTEC, Noordwijk, The Netherlands

D. Kretz
Mannesmann-Rexroth, Lohr a/Main, Germany

ABSTRACT

The European Space Agency (ESA) has decided to extend its test facilities at the European Space and Technology Centre (ESTEC) at Noordwijk, The Netherlands, by implementing a 6-degree-of-freedom hydraulic shaker.

This shaker will permit vibration testing of large payloads in the frequency range from 0.1 Hz to 100 Hz. Conventional single axis sine and random vibration modes can be applied without the need for a configuration change of the test set-up for vertical and lateral excitations. Transients occurring during launch and/or landing of space vehicles can be accurately simulated in 6-degrees-of-freedom. The paper outlines the performance requirements of the shaker and provides the results of the various trade-offs, which are investigated during the initial phase of the design and engineering programme. Finally the paper presents the resulting baseline concept and the anticipated implementation plan of the new test facility.

INTRODUCTION

The European Space Agency (ESA) has developed and maintains major environmental test facilities, at its Technology Centre (ESTEC) at Noordwijk, the Netherlands. The facilities are at the disposal of industry, scientific institutes and projects to support space programmes, in particular those of ESA and its member states.

Performance characteristics of the test facilities are regularly reviewed and adapted to the needs of future Agency programmes. In this context the Agency is executing a design and engineering phase of a 6-degree-of-freedom hydraulic shaker ("HYDRA") and of the associated infrastructure. The addition of these facilities will make the ESTEC Test Centre fully compliant with potential payloads of Ariane IV and Ariane V. An illustration of the extended Test Centre is shown in Figure 1.

The main facilities co-located in one building complex are:

- Large Space Simulator (LSS)
- Electrodynamic Vibration Systems
- Compact Payload Test Range (CPTR)
- Acoustic Facility (LEAF)
- 6-DOF Hydraulic Shaker (HYDRA)
- Support Facilities for Integration, EGSE, etc.

* Work supported by subcontracts from AGE-ELEKTRONIK gmbH (D), SEREME (F) and ASAP (D).

C-2

The co-location of facilities with transportation links within one building is a prerequisite for short system level test campaigns.

REASON FOR A HYDRAULIC SHAKER

Considerable effort has been spent by ESA during the last decade in order to study the possibilities for dynamic structure qualification and system acceptance of Ariane IV and Ariane V payloads. Mass and size of these payloads demand an extension of shaker forces and shaker table sizes, as well as an extension of the lower frequency range below 5 Hz. Also the introduction of a test method, which reflects a more realistic representation of the actual flight environment, has been a major objective of these studies (Ref. 1). It has been concluded that the simulation of the transients in multi-direction at the interface of launcher and spacecraft produces the most realistic structural responses, unlike traditional sine- or random tests, which lead to unrealistic responses and therefore bear the risk of over- or under-testing. Figure 2 illustrates a transient at the payload interface during an Ariane IV launch. Based upon experiments with multi-degree-of-freedom hydraulic shakers designed for earthquake simulation and taking into consideration the vast improvements of control systems in recent years, it has now become feasible to apply this technology for spacecraft testing. Multi-degree-of-freedom vibrators at DLR and IABG (Germany) have successfully demonstrated the reproduction of transients representative of the Ariane and Space Shuttle environment on structure models of a telecommunication satellite and of the Spacelab Pallet (Ref. 2 and 3). The encouraging results of the studies and demonstrations have lead to the decision to build a 6-DOF Hydraulic Shaker ("HYDRA") at ESTEC, which shall be operational in 1996. The shaker will allow conventional sine and random testing as well as multi-axes transient testing. Sine and random tests can be performed without any re-configuration of the test set-up for tests in the longitudinal and lateral axes. This will contribute to shortening of test campaigns and it will reduce the risks of damage to the test article because of reduced handling operations. The existing facilities in Europe were designed primarily for earthquake simulation and do not meet with stringent safety and cleanliness requirements which, together with the technical requirements, are necessary for the test of spacecraft. Therefore, the design and engineering phase for the HYDRA Facility has started in February 1992. The baseline concept was confirmed in a preliminary design review; it is based upon the following requirements.

MAIN PERFORMANCE REQUIREMENTS

- Span of test table	5.5 m
- Eigenfrequency in loaded condition	> 100 Hz
- Frequency range	0.1 Hz - 100 Hz
- Displacement in x, y, z direction	
a) baseline	± 25 mm
b) growth potential	± 50 mm
- Rotation around x, y, z	
a) baseline	± 0.5 degrees
b) growth potential	± 1 degrees

- Dynamic performance — see Figures 3 and 4

- Control accuracy

o	translation	acceleration	± 0.05 m/s^2 (rms)
		displacement	± 0.05 mm
o	rotation	acceleration	± 0.02 rad/s^2 (rms)
		displacement	± 0.001° degrees

or ± 2,5 % of the input signal, whichever is the larger.

- Test modes
  a) sine / sine sweep
  b) random
  c) transient

- Test Article Mass — 7000 kg *)

- Centre of gravity above table surface — 5.0 m *)

- Offset of payload centre-line from table centre-line — 0.5 m *)

- Moments of inertia related to centre of gravity — $I^*_{xx} = I^*_{yy} \leq 40.000$ kg/m² ; $I^*_{zz} \leq 15.000$ kg/m²

- Moments of inertia related to test table surface — $I^*_{xx} = I^*_{yy} \leq 215.000$ kg/m²

- The design must provide for on-line checks to protect the spacecraft against over-testing and for safe shut-down procedures in case of facility malfunctions.

*) These parameters are related to the maximum dynamic performance stipulated in Figures 3 and 4 and will increase if the acceleration requirements decrease.

TEST TABLE / SYSTEM GEOMETRY

The test table is the mechanical interface between the test article and the hydraulic actuators. It must meet the following requirements:

a) The lowest elastic natural frequency in loaded condition should be above 100 Hz, to reduce resonance problems in the operating frequency range of the facility.

b) The location of the actuator attachment points should be optimized to minimize distortions, which could be induced by the basic elastic modes.

c) The mass should be minimized in order to reduce the installed power.

The system geometry is a major element effecting the vibration modes of the table and therefore its natural frequency. Also the positioning of the actuators around the table has a large influence on the actuator forces and oil flow requirements. Therefore a compromise has to be found. The trade-off was based upon finite element models for six different options (Figure 5). For each actuator arrangement, the rigid body modes and the first three elastic modes were calculated. The model took into account the stiffness, position, and mass of all actuators; the table was modelled as a simple box structure. The main conclusion of this study was that the interfaces of the horizontal servo actuators must be located as high as possible in order to minimise the coupling factor between translation modes along X and Y axes and rotation modes around X and Y axes. Horizontal and vertical actuators must be located as far as possible from the vertical centre-line of the table. For the elastic modes, the asymmetric arrangements of options 1 and 2 have low stiffness (compared with the other four options) and an uneven weight distribution. This would be more difficult to control and would also require a larger seismic foundation to absorb asymmetric forces. The finite element analysis on the square table showed clearly that the torsional resonance is in most cases much lower than that of the other modes (Figure 6). This was one of the main reasons why options 5 and 6 were rejected. If the X-Y actuators are positioned on the nodal lines of the torsional mode, the actuators will have no influence on the resonance frequency, whereas positioning the X-Y actuators on the table corners would considerably reduce the torsional frequency. Consequently, the best choices for the X-Y actuators are options 3 and 4. With respect to the Z-actuators, option 4 has the advantage that all actuators are positioned on the table-sides. In this configuration it is possible to cut the corners of the table (Figure 7), thereby forming an octagonal table and saving approximately 3000 kg on the overall table mass. As this corner mass has a large influence on the torsional mode of the table, it would also raise its first natural frequency. Option 4, however, leads to a mass and stress concentration at the co-located actuator/table interfaces. Option 3 was finally chosen as the optimum arrangement, because it allows to remove the corners, but also distributes the lumped masses of the actuators evenly around the table. The interface of the Z-actuators with the table is illustrated in Figure 8. It is located as far as possible from the table centre line to cope with the turning moments.

ACTUATOR FORCES

The actuators dynamic force and flow requirements for various arrangements were computed with a specially developed computer program called 'MAP', which permits the input of table motions at different frequencies and amplitude levels in any of the six degrees of freedom. The program can calculate the maximum dynamic force and flow requirements for all of the system's eight actuators at any given point in time, or the dynamic force, flow, velocity, displacement and acceleration requirements of the actuators over the test period. With this tool, the optimum positions of the actuators can be determined for any given specification. Figure 9 shows the data supporting the selection of actuators. For commonality reasons, 630 kN (static force) standard actuators (type Mannesmann-Rexroth) were selected for X, Y and Z. These have a piston rod diameter of 160 mm and are equipped with hydrostatic pocket bearings. The analysis illustrated in Figure 10 shows that the maximum dynamic forces lie below 500 kN for all single translatorial or rotational motions. This is also the case with translatorial motions that do not lie on the systems main axis, but are arbitrary vectors in space. The largest dynamic force requirements (580 kN) were found in a combination of rotation about the X and Y axis simultaneously (rotation about a diagonal axis). As the dynamic force requirement of 580 kN arises during the part of the sine wave cycle where the velocity is at its minimum (i.e. the acceleration is at its maximum), an actuator with a static force capacity of 630 kN is adequate.

TABLE DESIGN

The design for the test table takes into account all the above conclusions, as well as the results of investigations with respect to materials and construction. The use of steel and aluminium was considered, and showed that the price disadvantage of an aluminium table can be justified with its high internal damping and the reduction in the overall mass. The high internal damping of aluminium can reduce some of the difficulties associated with multi-axes control systems and the reduction in the table mass reduces the demands on the power of the hydraulic system. Furthermore, a double layer box structure (Figure 11), manufactured from welded aluminium plates, was selected. The rectangular boxes of this structure have the dimensions 0.786 m x 0.786 m x 0.800 m and a plate thickness of 20mm. These dimensions were chosen as a compromise between the need for high web plate local modes and sufficient space to allow for good welding conditions. Plate resonance problems were experienced with other tables of this size, because the natural frequencies of the individual plates were very similar. Therefore, access holes of varying sizes will be cut into the plates to avoid these effects.

SUSPENSION OF SEISMIC FOUNDATION

Air springs and steel springs (Figure 12) combined with viscose dampers were investigated. For both suspension systems the dynamic performances were very good and very similar. The highly damped steel springs were selected because they have the following advantages:

- Easy on-site implementation. They can be introduced into the form work of the seismic foundation before pouring the concrete by prestressing the steel spring boxes to their final working height.

- High damping ratio in each degree of freedom, which allows operations at very low frequencies.

- High reliability and no maintenance.

The configurations which were investigated are illustrated in Figure 13. In option 1 the seismic mass is suspended at the level of its centre-of-gravity. This provides de-coupling of lateral and the respective rotational rigid body modes. The second option with a base-suspension does not provide this de-coupling but leads to a simpler construction and a lower mass. A detailed analysis of the dynamic performance has shown no essential differences between the two options for this application. Therefore option 2 has been selected for cost reasons.

CARDANIC BEARINGS

The table motions in six degrees of freedom (D.O.F.) are achieved with eight linear actuators, which will be connected between table and foundation by means of bearings having at least two rotational degrees of freedom.

The main technical requirements placed on these bearings are:

- low mass, due to the dynamic application [< 500 kg];
- backlash free, as this will affect the mechanical noise;
- high stiffness, as this will affect the actuator's overall natural frequency [> 10^{10} N/m];
- low friction coefficient [< 0.005];
- stiffness and friction should be as linear as possible (Figure 14);
- acceptable life rating even when subjected to unfavourable operating conditions, such as shock or high frequency loading with very small radial movements.

Figure 15 shows 3 groups of bearings, which were investigated. The first group relies on the use of dissimilar materials (Steel/ PTFE) sliding over each other (usually referred to as plain bearings). This type of bearing can meet the stiffness requirements. However, it is essential that the bearing is pre-loaded to prevent excessive wear, which leads to unacceptable friction levels. The second group uses the hydrostatic principle and assures no metal-to-metal contact and thereby leads to very low friction levels. Unfortunately the bearing has to be extremely large to achieve the required stiffness and would therefore lie outside the weight limit. The last and possibly the only acceptable solution for such an application is the pre-loaded cylindrical roller bearing, because it is most suited for taking large forces. With this type of assembly the bearing stiffness can be optimised by the use of two, three or even four row bearings, providing a very compact solution with friction levels very much lower then that of plain bearings. Interference fits are used to pre-load such a bearing without the possible risk of alignment errors. The pre-load can be very accurately controlled by preheating the bearing housings before assembly. The influence of high frequency vibration on fretting corrosion can to a certain extent be reduced by optimising the pre-load. Lower pre-loads increase the relative motion between the rolling elements and the raceways, while higher pre-loads increase the contact pressure. As the life of the bearing will be reduced by contamination or local heat build-up it is extremely important that a continuous supply of filtered oil to the bearings is guaranteed. The oil also functions as a coolant, dispersing local heat and maintaining a satisfactory operating temperature.

CONTROL SYSTEM

The dynamic behaviour of the test facility is governed by the performance of the control system. An analysis of existing control systems for multi axes test facilities was carried out during the conceptual design phase for the HYDRA facility. It showed clearly that most of these systems have disadvantages that can be avoided by modern control concepts. Most of the conventional control algorithms available on the market are designed for universal applications convenient for various multi-axes environments (e.g. electrodynamic shakers and hydraulic shakers with lower accuracy requirements). They consider the test facility as an abstract object where "n" drive signals produce "m" sensor signals. A mathematical description of these dependencies is assumed by linear mapping given by the so-called "transfer matrix". A desired load to the test item is achieved by iterative signal adaption which consists of two independent steps:

- Automatic identification of the transfer matrix of the loaded facility before the test starts by applying low-level noise.
- Repeated iterations of the test signals including manual correction procedures.

The optimization of the control signals depends on the test item, the reference signals and the acceptable tolerances. It is a complex procedure, which requires engineering knowledge and experience with respect to the facility behaviour as well as the test item. Furthermore, the experimenter might limit the number of iterations to avoid undue exposure of the test item. The new digital control algorithm applied for HYDRA is based on mathematical modelling of the actuators, servo valves, and the overall geometric arrangement. A computational function is implemented to synchronise the actuator controllers in order to avoid warping distortion stresses in the shaker table. Further attention is given to the "dead time" of the facility caused by the hydraulic system, anti aliasing filters, and sensors. To achieve the required accuracy of the test signals, a control loop time of 100 microsec is necessary. This requires a powerful computer system. Furthermore, resonance frequencies of the test item can be determined on line to reduce the input level automatically (notching).

The goals of the modern control techniques as used by the HYDRA control algorithm are:

- Accurate control by non linear actuator control algorithms.
- Control signal generation for optimized use of the special geometric arrangement of the HYDRA actuators.
- Minimized stress at the table, caused by the over determination arising from eight actuators at the table and six degrees of geometrical freedom.
- Granted convergence of the control procedure.
- Minimized manual intervention by the use of self adapting kinematic closed loop controllers.
- High security by a distributed control system.

A customized computer system is employed to support the software of the control algorithm. The treelike structured controller network is shown in Figure 16. It is based on parallel processing with highly distributed computing power, permitting:

- on-line modelling of the actuators;
- on-line, adaptive modelling of the kinematic situation;
- highly accurate data acquisition;
- well structured to identify and react to critical situations.

This leads to the following advantages for the user:

- no special knowledge of operators is required;
- item independent control structure;
- automatic emergency shut down;
- preventive maintenance by an expert system;
- secure shut down procedures;
- automatic notching;
- expandability of number of notching channels.

The overall supervisor and control concept is subdivided into several parts (Figure 17) for data acquisition and emergence control on different levels, the kinematic controller, the pilot sequence controller and the so called supervisor system. Each subdivision can operate as stand alone system as well, allowing different levels of emergency shut down procedures, if one of the components or interface lines fails. In parallel to the specialized closed loop control processing units, a three-processor system (based on the UNIX operation system) allows the comfortable and mostly standardized handling with external ETHERNET access. External data can be used as reference signals. Integrated data analysis can be applied in connection with the item data analysis system. Common data base structures with extensive backup options are used to prevent loss of data even if one of the storage media or power supply fails.
As a conclusion, the distributed supervisor and control concept allows redundant operations with high reliability and as a consequence with a maximum security for the test item. The operations are well controlled without requirements for specialized knowledge of the facility operators.

FACILITY IMPLEMENTATION

The implementation of the HYDRA facility is scheduled in two phases:

Phase I "Design and Engineering"
Phase II "Procurement, Installation and Acceptance"

Phase I has started on February 1st, 1992 and will be terminated in May 1993. This phase is subdivided into 4 subphases, each being completed with a formal review:

- Concept Design Review (May 1992)
- Preliminary Design Review (September 1992)
- Critical Design Review (January 1993)
- Final Design Review (April 1993)

Phase II will be started by the end of 1993 and the facility will be operational for spacecraft testing by the middle of 1996. According to the present schedule of ESA the first test will be performed on the Polar Platform (PPF), which will be launched with ARIANE V in 1998.

CONCLUSION

The concept design of a 6-degree-of-freedom hydraulic shaker for spacecraft testing has been sucessfully completed. The design work has resulted in a baseline, which provides confidence that the final facility design and its subsequent implementation will meet the demanding specifications. After successful implementation, it will provide ESA with a very efficient tool for dynamic testing of large structures.

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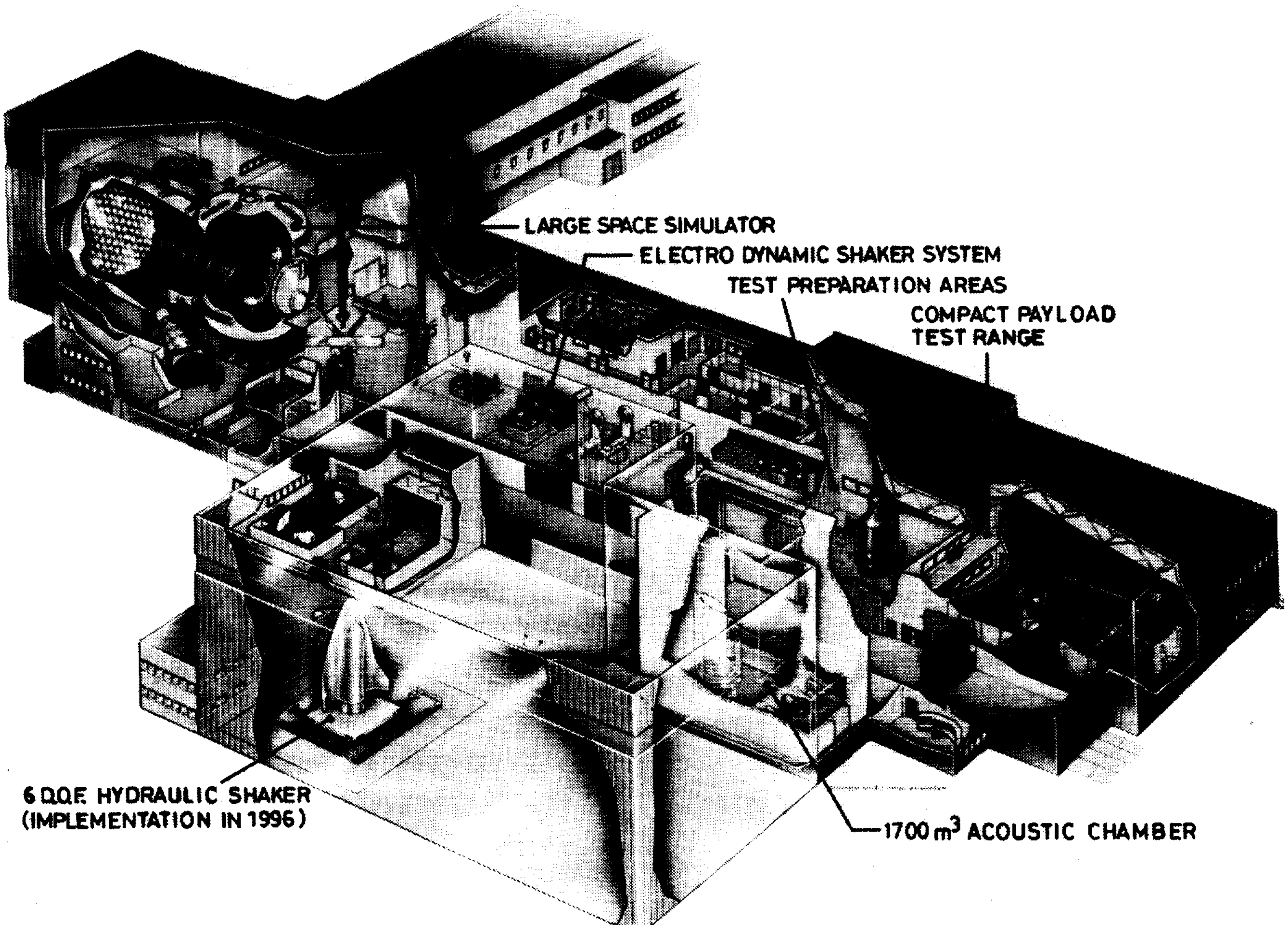


Fig 1 ILLUSTRATION OF THE ESA TEST CENTRE AT THE EUROPEAN SPACE TECHNOLOGY CENTRE (ESTEC)

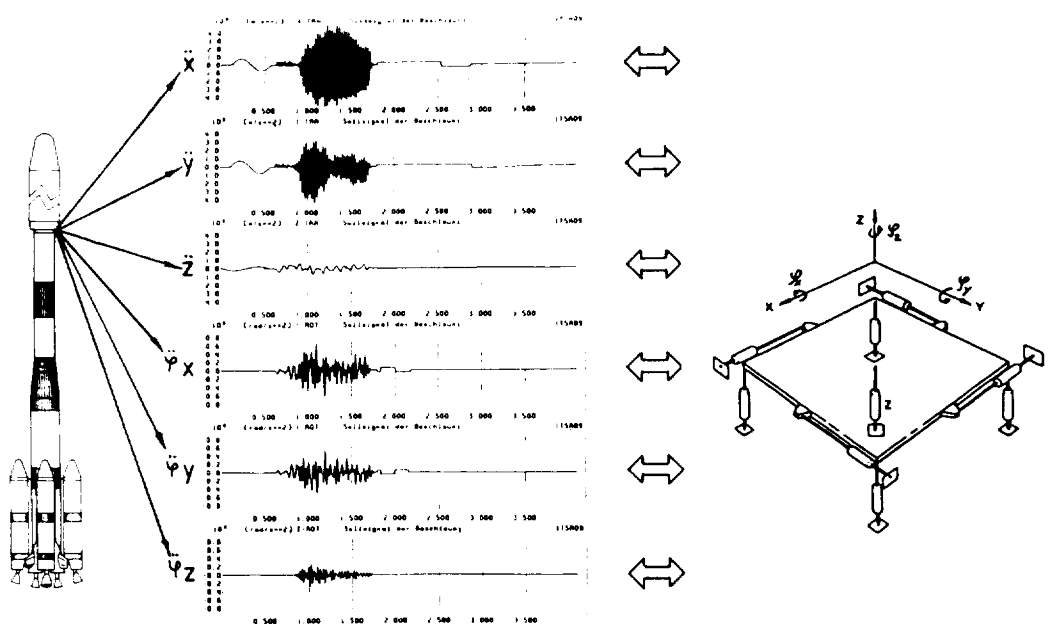


Fig 2 TYPICAL TRANSIENT AT THE PAYLOAD INTERFACE OF ARIANE IV IN 6 DEGREES-OF-FREEDOM

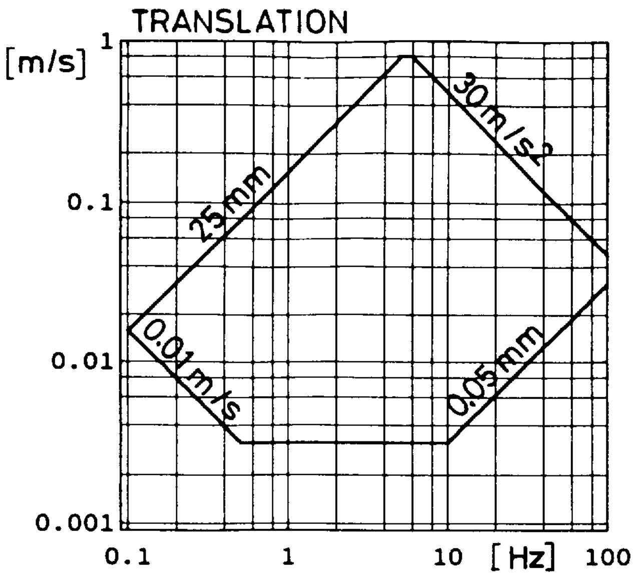


Fig.3 PERFORMANCE DIAGRAM FOR X/Y AND Z TRANSLATION WITH TEST TABLE LOADINGS.

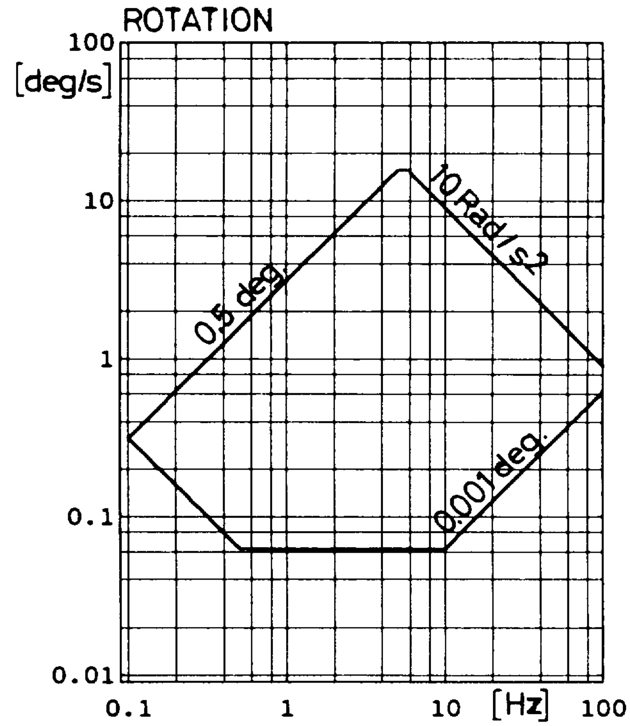


Fig.4 PERFORMANCE DIAGRAM FOR ROTATION ABOUT THE X/Y AND Z AXIS WITH TEST TABLE LOADINGS.

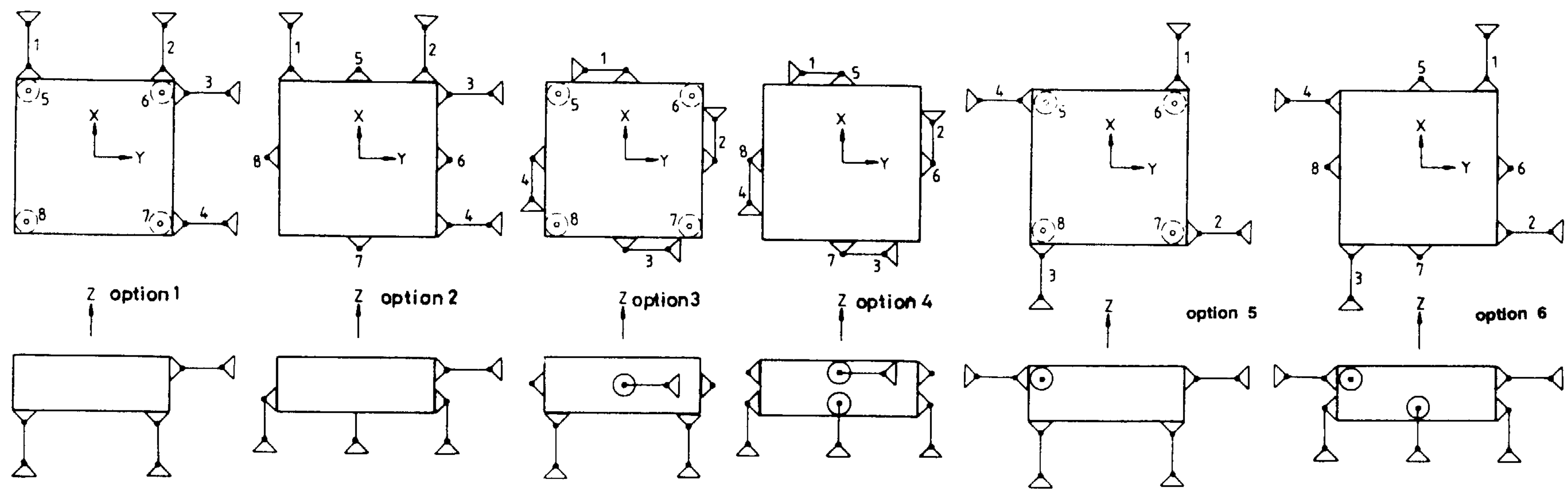


Fig.5 CONFIGURATIONS FOR SYSTEM GEOMETRY TRADE-OFFS

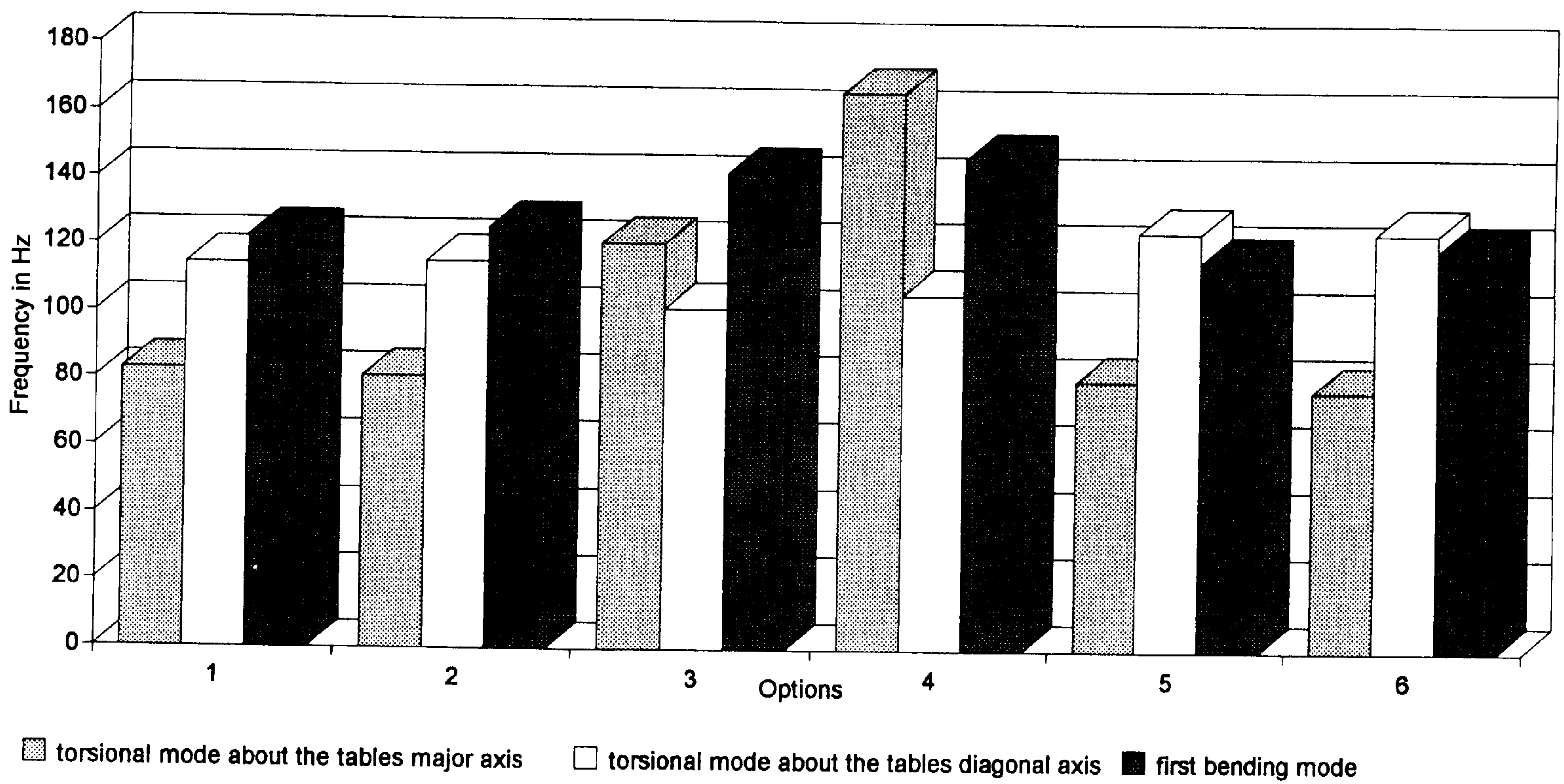
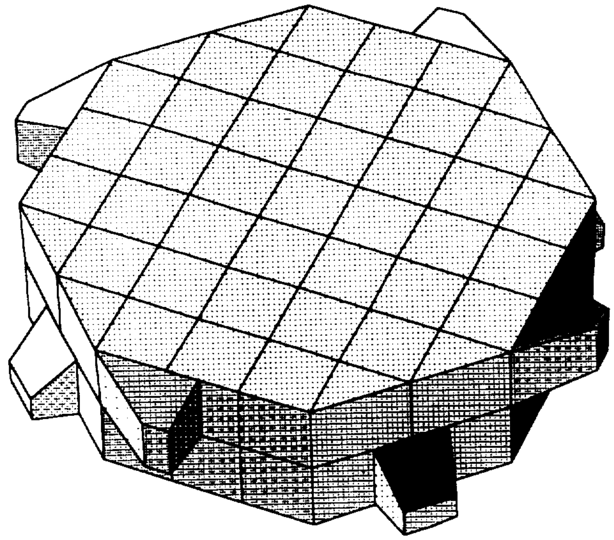
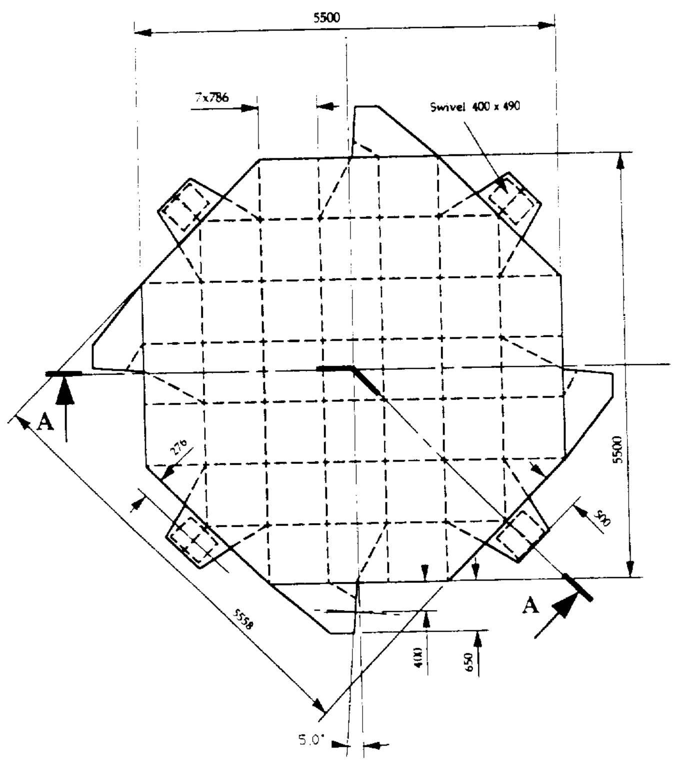


Fig.6 MODES OF DEFORMATION OF THE TEST TABLE



PANORAMIC VIEW



PLAN VIEW

Fig.7 OCTAGONAL TEST TABLE

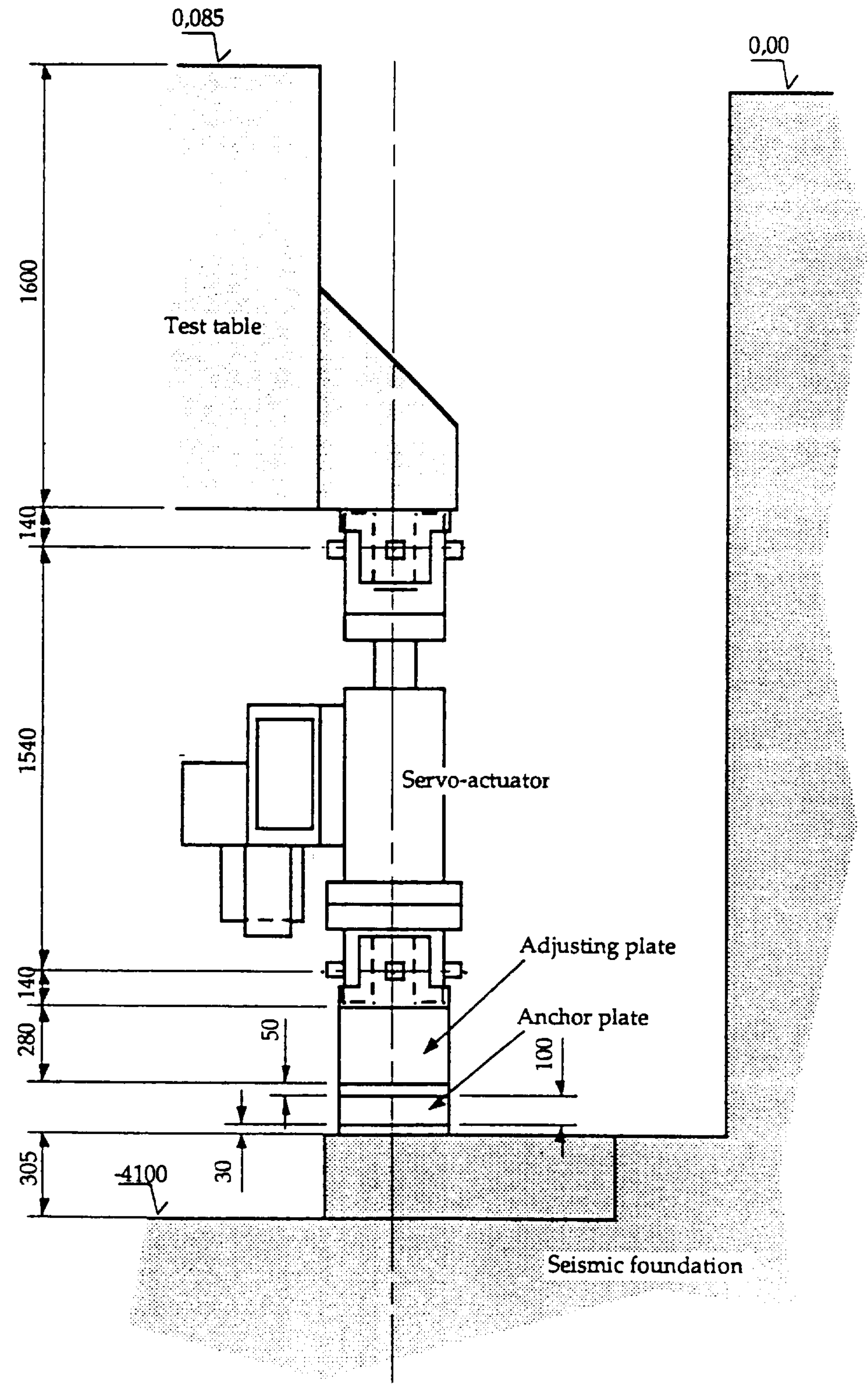


Fig.8 VERTICAL SERVO-ACTUATORS (WORKING STROKE: +/- 85mm)

MAP	Actuator	Foundation x	Foundation y	Foundation z		Table x	Table y	Table z		Actuator stroke	start position	piston Ø	rod Ø
1	X1	-1610	3324	-450	mm	-70	3200	-450	mm	170	85	233	160
2	X2	1610	-3324	-450	mm	70	-3200	-450	mm	170	85	233	160
3	Y1	-3324	-1610	-450	mm	-3200	-70	-450	mm	170	85	233	160
4	Y2	3324	1610	-450	mm	3200	70	-450	mm	170	85	233	160
5	Z1	-2175	-2175	-3290	mm	-2175	-2175	-1750	mm	170	85	233	160
6	Z2	2175	-2175	-3290	mm	2175	-2175	-1750	mm	170	85	233	160
7	Z3	2175	2175	-3290	mm	2175	2175	-1750	mm	170	85	233	160
8	Z4	-2175	2175	-3290	mm	-2175	2175	-1750	mm	170	85	233	160

	weight		width	length	depth		C of G x	y	z		moment of inertia Ix	Iy	Iz	
Moving mass	23000	kg	5500 (8 sided)	5500	1600	mm	0	0	-700	mm	93000	93000	120000	kg m²

	weight		C of G x	y	z		moment of inertia Ix	Iy	Iz	
Test article	7000	kg	-355	-355	5000	mm	40000	40000	15000	kg m²

Fig.9 SYSTEM MODEL FOR ACTUATOR SELECTION

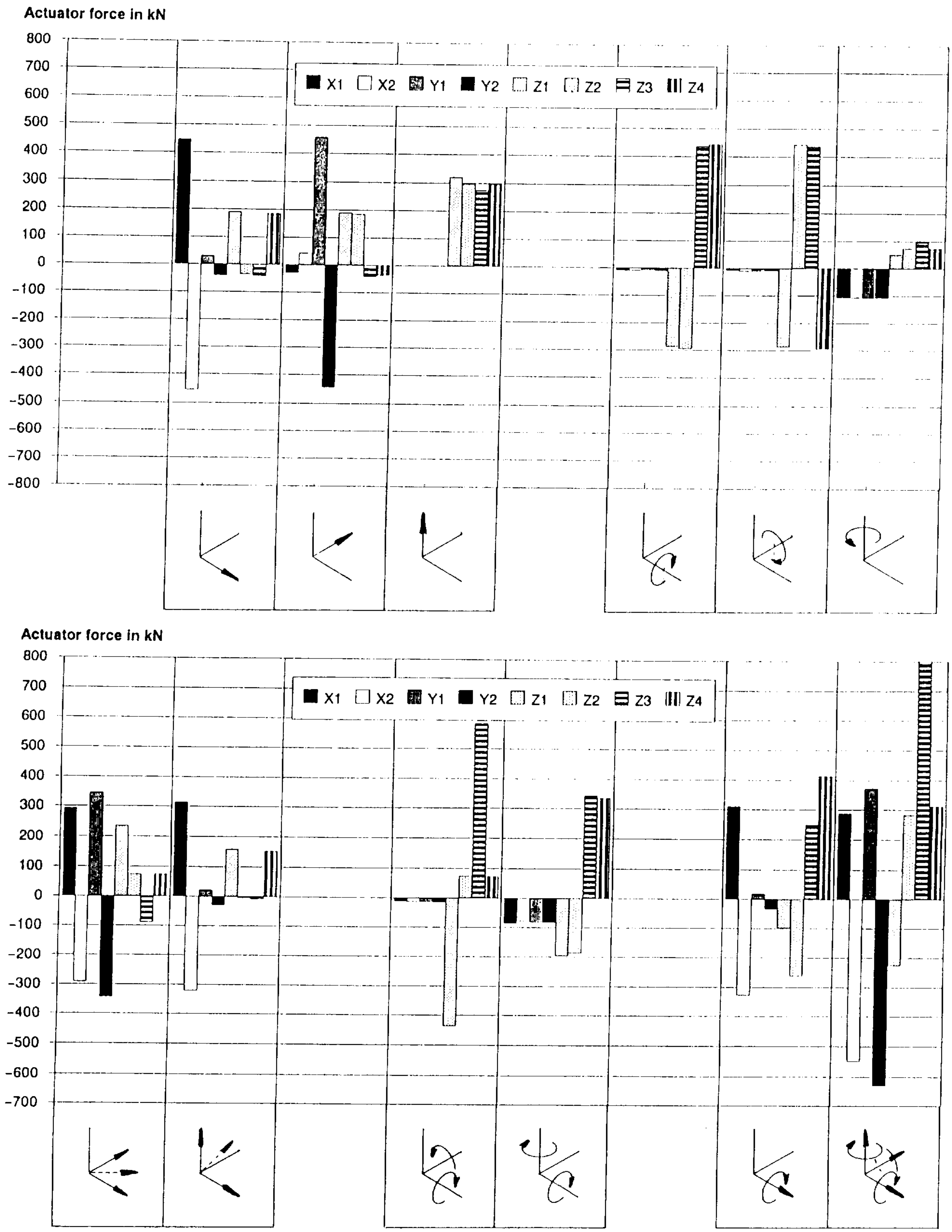


Fig.10 ACTUATOR FORCE REQUIREMENTS (AT MAXIMUM ACCELERATION)

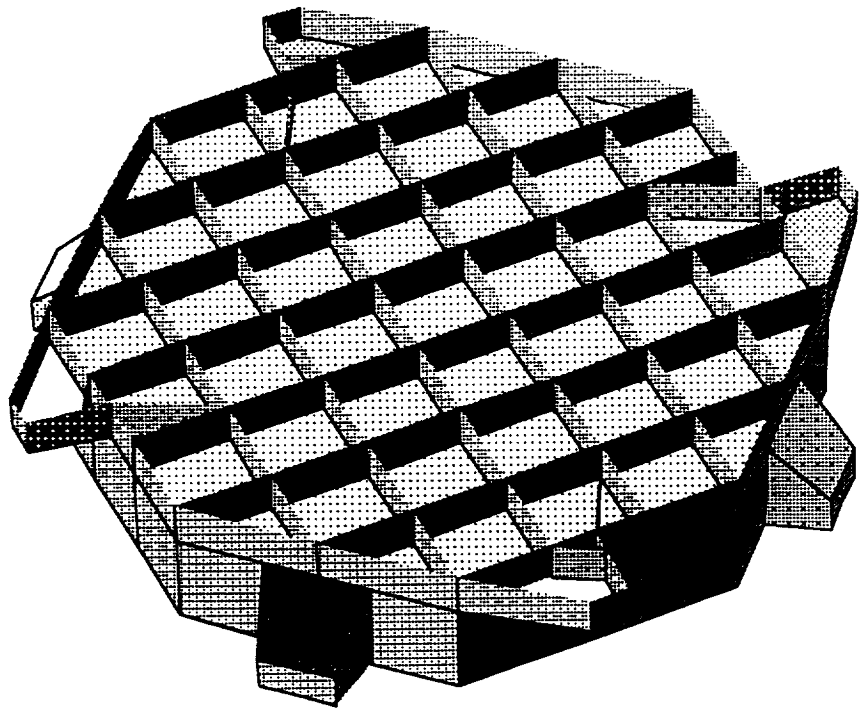


Fig.11 SECTION THROUGH HORIZONTAL ACTUATOR ANCHOR SYSTEM

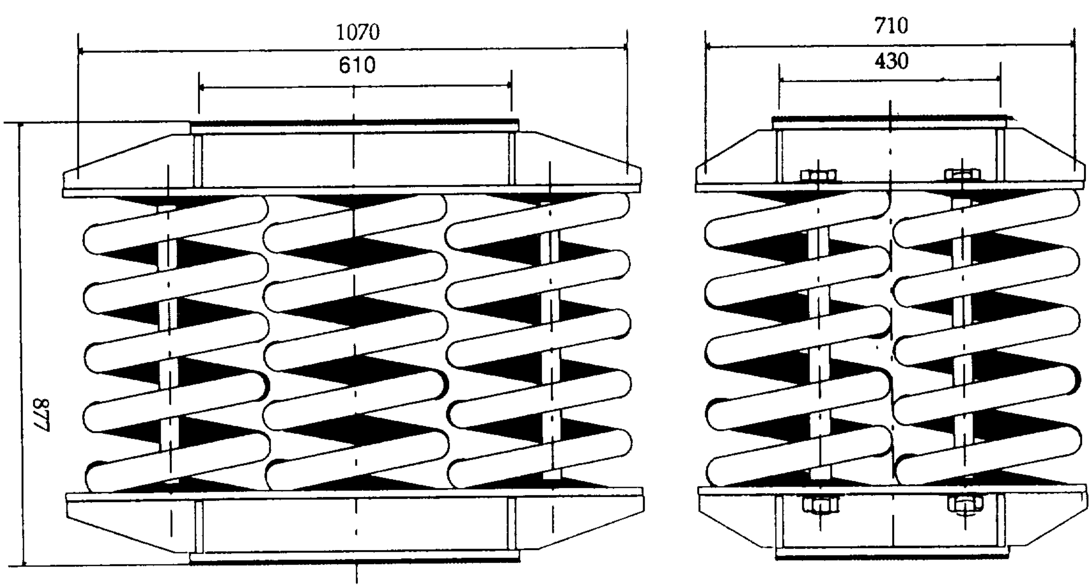


Fig. 12 STEEL SPRING BOXES

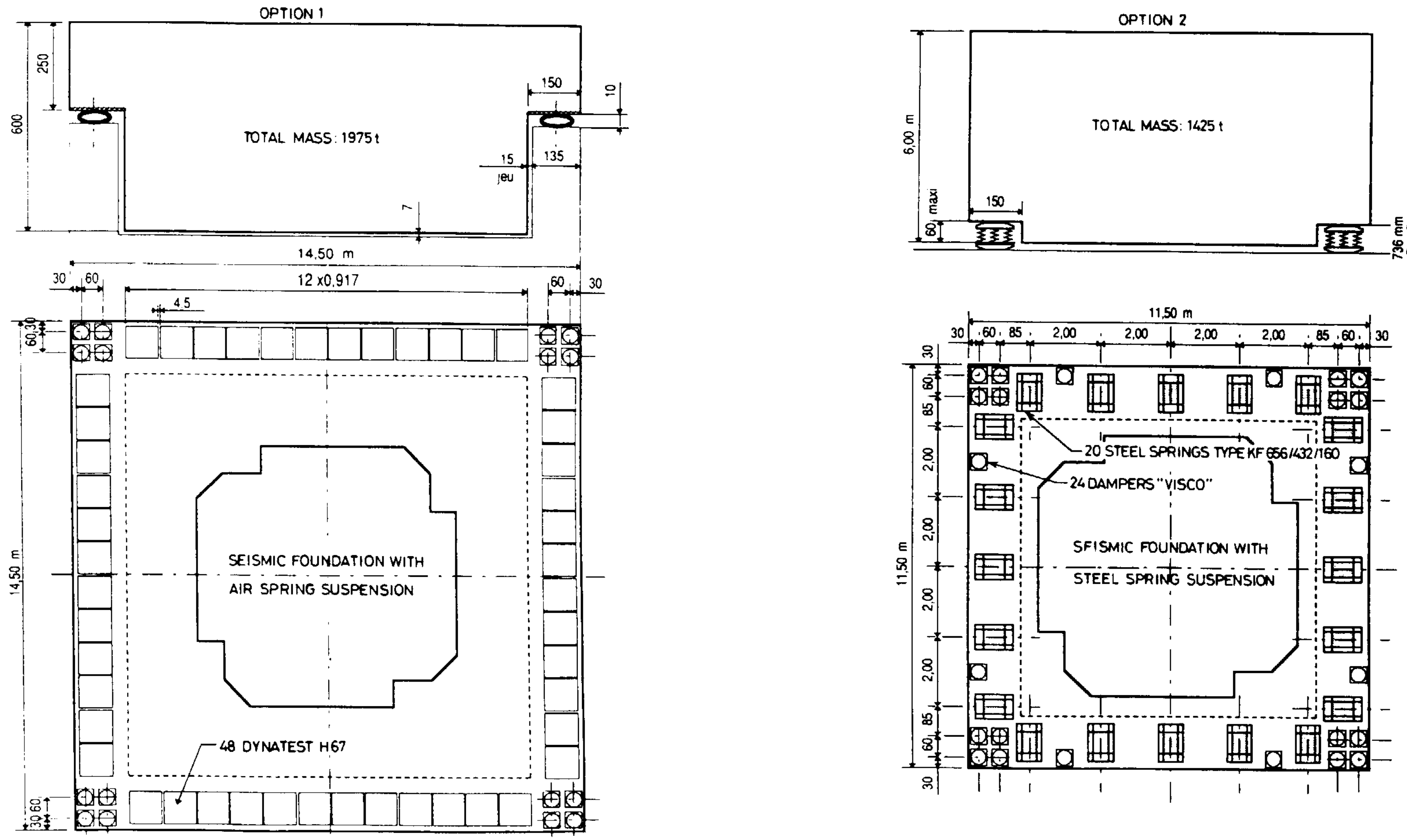


Fig.13 OPTIONS FOR SEISMIC FOUNDATION AND ITS SUSPENSION

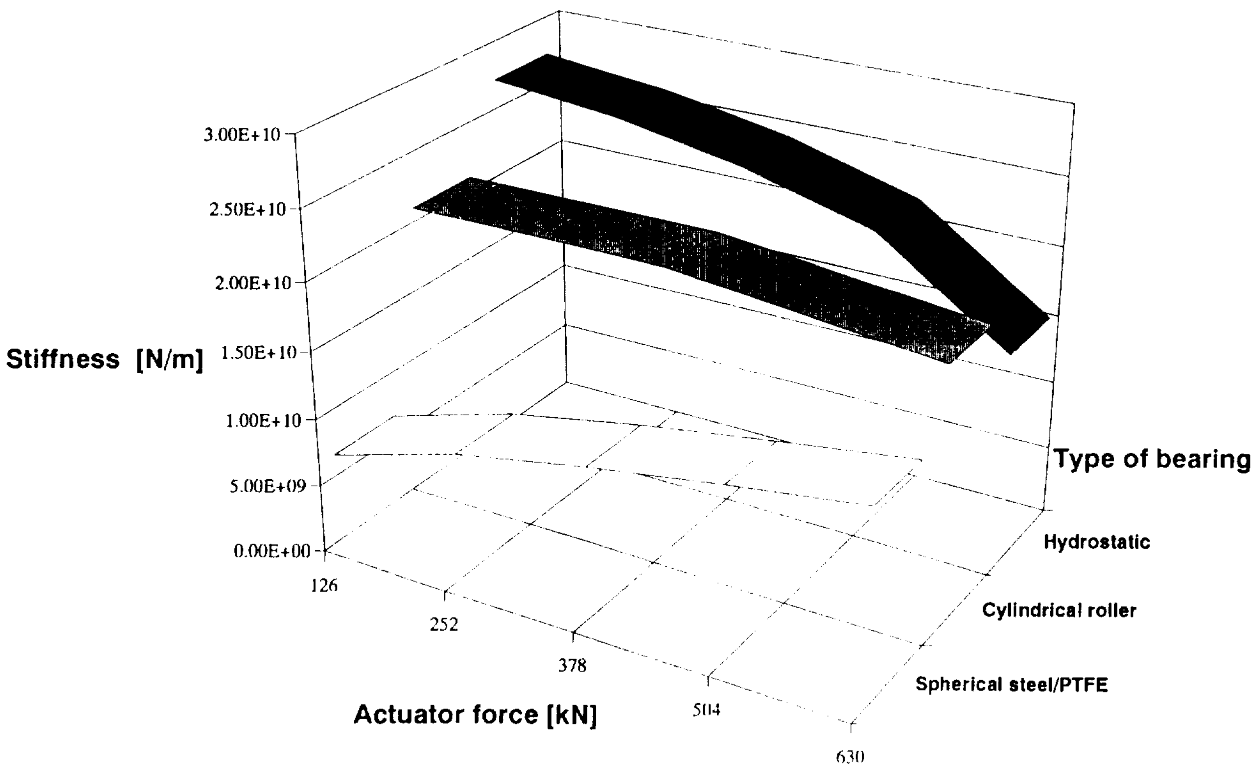


Fig.14 BEARING STIFFNESS COMPARISON

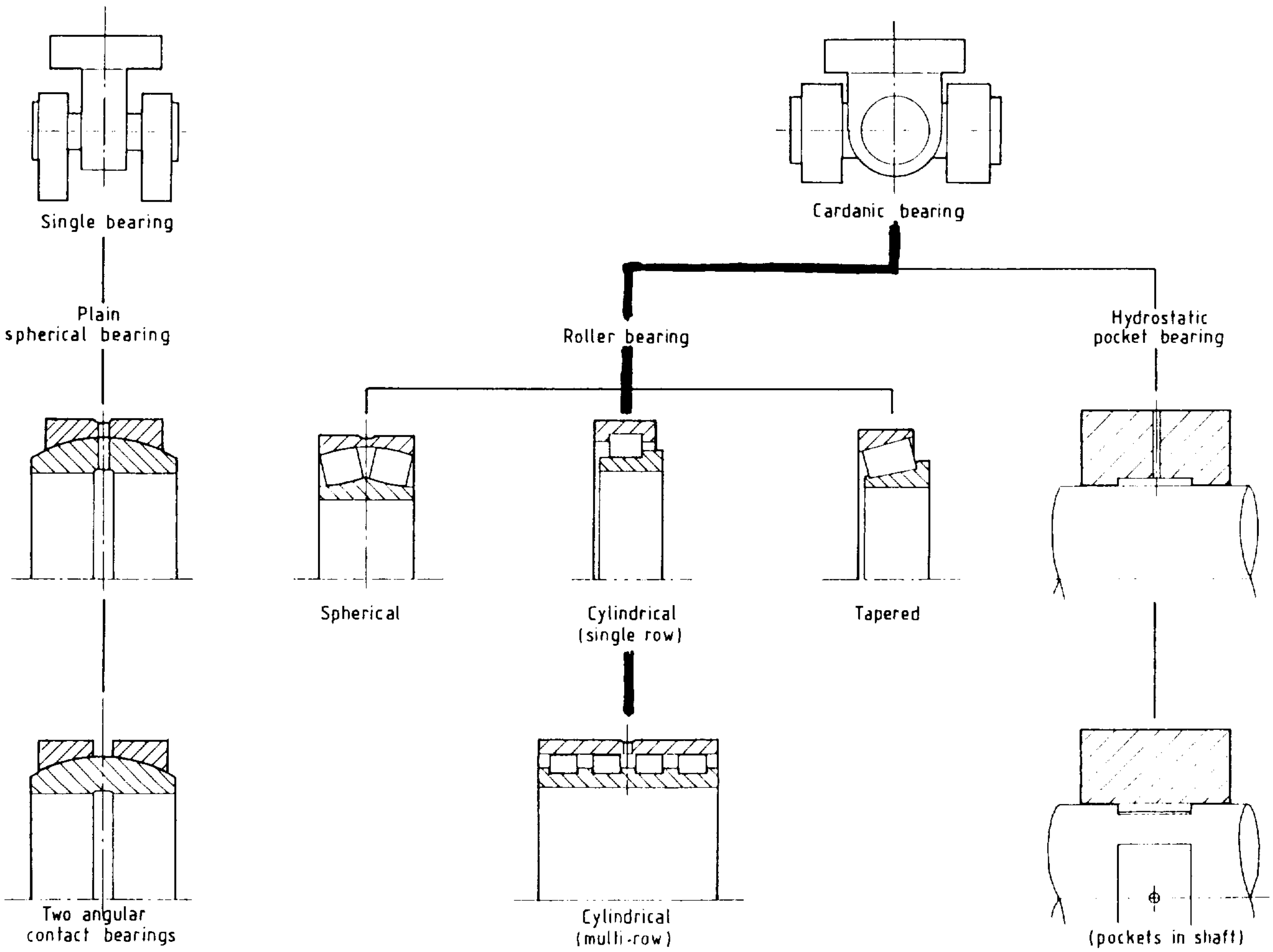


Fig.15 BEARING TRADE-OFF STUDY

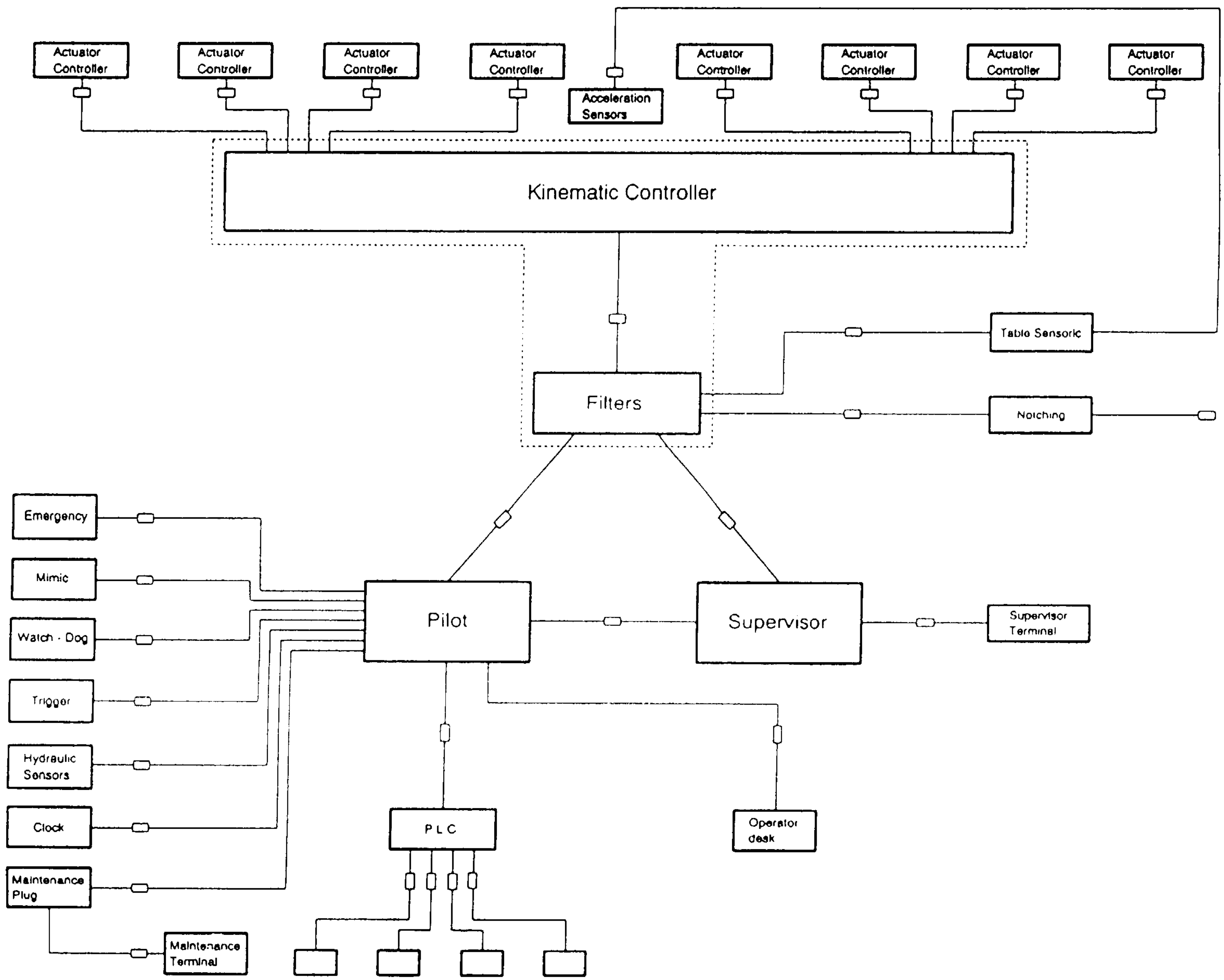


Fig.16 HYDRA CONTROL DISTRIBUTION AND STRUCTURE

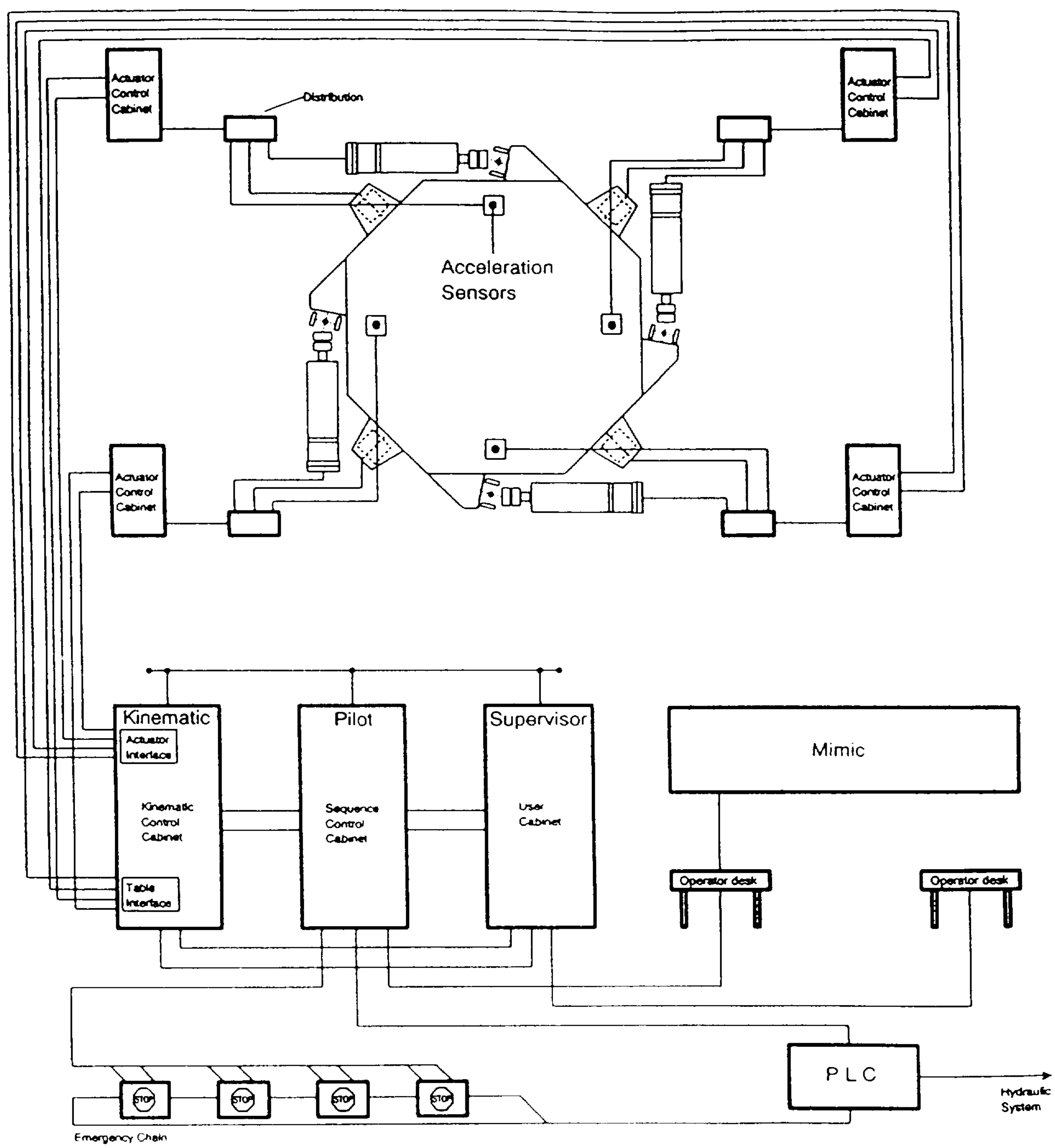


Fig.17 HYDRA CONTROL COMPONENT ARRANGEMENT