

Hybrid Shaker Technology for Wide-band Vibration Test Systems

Mr. Katsuhiko Nakamura¹, Mr. Kazuyoshi Ueno¹
Dr. John Goodfellow²

¹IMV Corporation, R&D Centre, 2-6-10 Take-jima, Nishi-yodogawa-ku,
Osaka, 555-0011, Japan
E-Mail: ueno@imv.co.jp

²IMV Europe, Devonshire Business Centre, Works Road,
Letchworth, Herts, SG6 1GJ, UK
E-Mail: john.goodfellow@imv-tec.com

1.0 Overview

Through its excellent linear response that extends over several thousand Hz, ED (electro-dynamic) shaker systems have been widely applied to vibration testing carried out by various kinds of industries. The current market trend is for longer displacement testing, reflecting the recent requirements from development activities of innovative technology such as electric cars. It is possible but generally not economic to apply ED shakers to testing that requires a displacement larger than 200mm. The fundamental restriction comes from the implementation of the magnetic circuit that should grow in size to achieve the required long stroke. However, it is not only the excellent accuracy in the acceleration or velocity domain that is required and is possible with ED shakers, but even in the accuracy achievable in displacement domain has been required from ED shakers.

On the other hand, in the fields of earthquake resistance testing and high displacement transport testing, for which the traditional servo-hydraulic shaker technology has been mainly applied, there are increasing needs for sophisticated and diversified testing for accurate waveform replication. Most of these new applications are requiring higher accuracy in replication of the waveform; in other words, the requirement for higher frequency response is increasing in this field and the ED shaker is the best fitting technology for higher frequency response.

To comply with such requirements for widening the frequency band of vibration testing, the “Hybrid Shaker” technology has been developed by combining the ED shaker technology which dominates in high frequency response and the AC-servo motor shaker which economically provides the capability of long stroke vibration. This technology has opened a really wide-band world of vibration testing that covers a frequency range from 0.1Hz to several hundred Hz.

Economy in footprint, (and without the need for hydraulic oil giving) economy in maintenance and laboratory cleanliness, are other valuable features of this technology.

2.0 Increasing requirements for tests with long stroke vibration

2.1 Automobile (Car body and car mounted parts)

Examples of requirements for long stroke vibration

(A) Ride comfort test (for seats and driving simulation)

A seat ride comfort test is carried out by replicating the floor vibration waveform acquired in a running car as the vibration on the test table on which the specimen seat is set. In the development stage, sensory examination by humans sitting on the specimen seat is tried, and very accurate replication of the vibration waveform from 0.1Hz to 100Hz is sometimes required.

In recent years, some users (manufacturers) who require realistic vibration replication apply 6-DOF ED shaker systems for vibration testing. The stroke of such systems is limited to around 200mm_{p-p} due to the limitation of the fundamental design of the ED shaker. Therefore, the measured waveform is necessary to be high-pass-filtered to remove very low frequency components. If this maximum stroke limitation can be extended, then the reality of replication at the very low frequency can be more improved.

(B) Vibration simulation for a whole vehicle

Each of the tyres of a fully assembled car (1, 2 or 4 wheels) is placed on its own vibration table comprising one or more ED shakers. The vibration table is termed a "Post" (correspondingly creating a 1-Poster, 2-Poster or 4-Poster system), and simulation of the running car condition can be examined. Hydraulic shakers, which are easy to manufacture for long-stroke equipment, have often been applied to this type of system. Requirements for rough road simulation and stepped road simulation often exceed 200mm_{p-p} stroke. Traditionally, the vibration simulation is focused over the range from 1Hz to several tens of Hz, but recently higher bandwidth of up to 300Hz is sometimes required and for such requirement, an ED shaker is used.

Since the ED shaker is typically limited in stroke up to 200mm_{p-p}, the measured waveform data has to be high-pass-filtered to remove very low frequency components. If this maximum stroke limitation can be extended, then replication for a rough terrain road surface requiring displacement with very low frequency components up to and including a high frequency road vibration of several hundred Hz would be possible.

2.2 Transportation

Random vibration tests having a PSD profile determined as an approximation of actual transportation vibration is widely employed in transportation vibration testing. The random vibration bandwidth of the testing vibration depends on the test standard employed. For example, ASTM transportation standard (ASTM D4728-6) requires 1Hz to 300Hz, and MIL transportation standard (MIL-STD-D810G-METHOD 514.6) requires 5Hz to 500Hz. So, 1Hz to 500Hz bandwidth is required for a system compatible to the both standards, and the ED shaker is most appropriate for this bandwidth.

It is required in the MIL-810G 514.6C-1 standard that the frequency of the dominant resonance (of the component under test) should be included in the testing bandwidth even if it exists in the frequency band lower than 10Hz. So in such a case, the displacement requirement for the test equipment may exceed 100mm_{p-p} stroke.

As the vibration system used for transportation testing often requires a large capacity for specimen size and weight loaded on the table, there is a tendency that the equipment is also applicable to earthquake resistance testing. In such a case, the system is required to have a very wide frequency bandwidth of 0.1Hz ~500Hz and a very large stroke of 500mm_{p-p}.

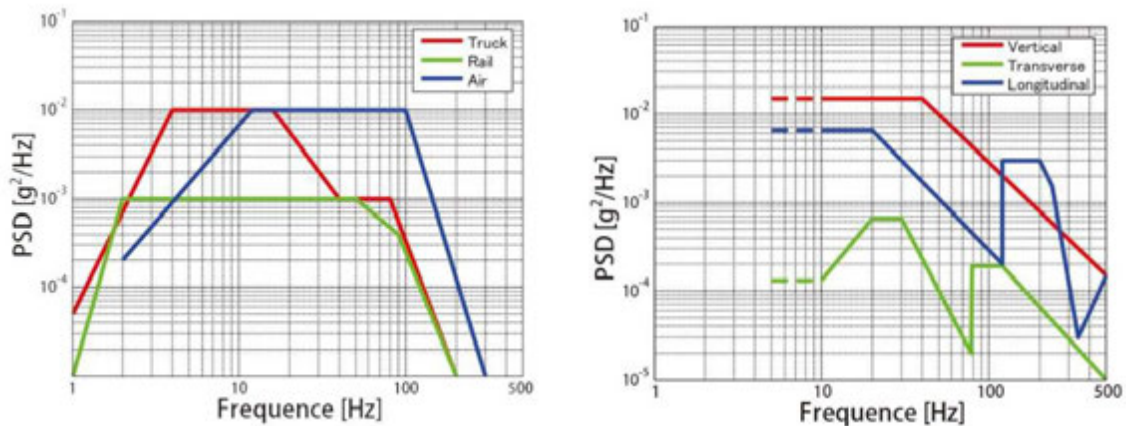


Figure 1 Random vibration test requirements (left: ASTM D4728-6, right: MIL-810G)

2.3 Earthquake resistance

In an earthquake resistance test, an accurate waveform replication of the representative earthquake acceleration waveforms recorded during past events is required for the vibration table, including the mounted specimen. As the dominant damage created by an earthquake is caused by exposure of the

buildings to high velocity and/or large displacement of the ground, it is required for the test equipment to have not only a good accuracy replication capability but also a very large stroke specification.

Examples of maximum Acceleration, Velocity and Displacement of representative earthquakes are shown below:

	El Centro	Hanshin (Kobe)
Max. Acceleration	3.42 m/s ²	8.18m/s ²
Max. Velocity	34.8cm/s	91.0 cm/s
Max. Displacement	185 mm _{p-p}	350 mm _{p-p}

Note:

1. El Centro (1940): NS direction acceleration data

2.Hanshin(1995) : NS direction acceleration data

Table 1 El Centro and Hanshin earthquake

Requirements for replication of these earthquakes and the maximum performance of IMV's earthquake resistance test system DS-3600-24L Hybrid Shaker which employs the Hybrid Shaker mechanism are overlaid on a same graph shown below:

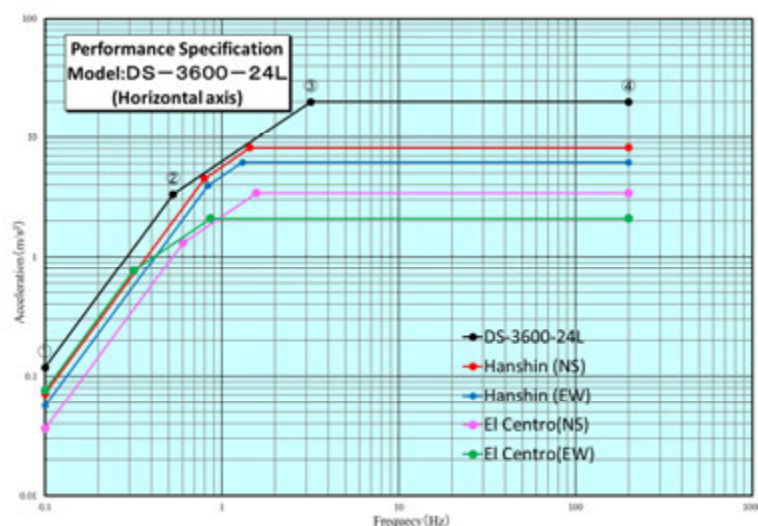


Figure 2 Rating performance of DS-3600-24L and requirements by earthquakes

3.0 Vibration generating machines

3.1 Electro-Dynamic shaker

The ED shaker is equipment that applies vibration to a specimen loaded on the armature table which is linearly vibrated by the electromagnetic force. This EM force is generated at the armature coil, which is placed in the static magnetic field. The magnetic field is generated by the field coil (or by a permanent magnet in the case of small shakers). The force is directly generated by the electromagnetic interaction between the static magnetic field and the armature current which is the drive current fed from the amplifier ($F=Bli$, where B is magnetic flux density, l is armature coil length and i is armature coil current). There is no other influence from the characteristics of materials and mechanical mechanisms that are used to build the ED equipment. Since by good design, influences from mechanical friction and damping can be minimized, the ED actuator has an excellent linear response. The usable maximum frequency is limited by the axial resonance of the coil-table structure, and the typical value is several thousand Hz. Contrary to the high frequency performance, to extend the very low frequency response requiring high displacement, is not economic in cost nor in size of the ED shaker. This limitation comes from the requirement to extend the static magnetic field over the length of the required displacement.

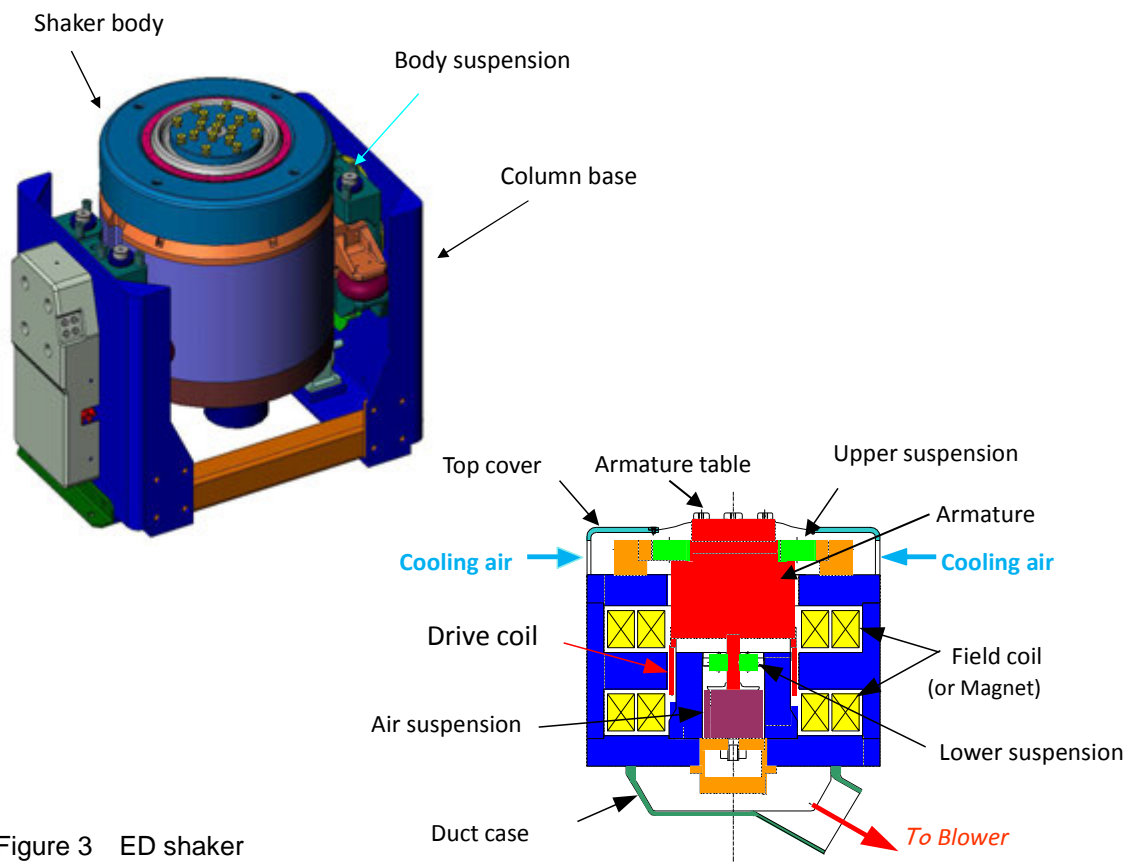
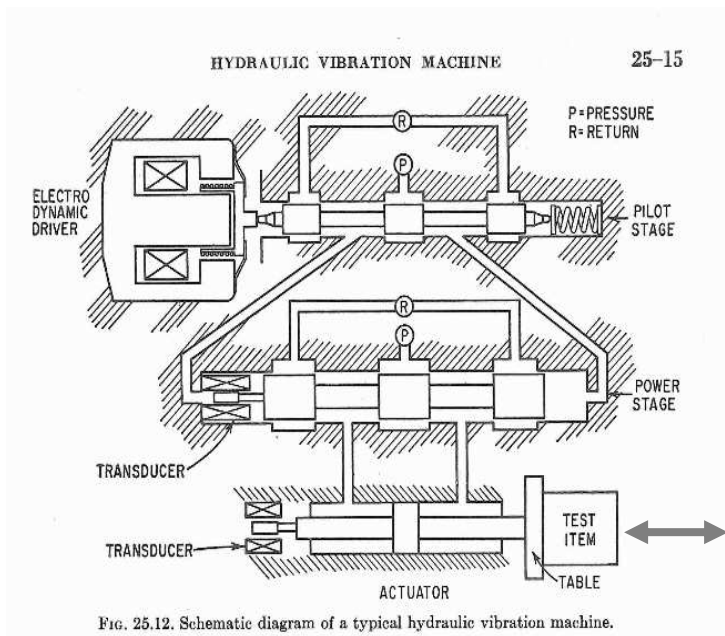


Figure 3 ED shaker

3.2 Hydraulic shaker

The hydraulic shaker is equipment that applies vibration to a specimen mounted on the actuator rod which is formed from a piston driven by hydraulic oil of which flow is controlled by a servo-valve constructed using a tiny ED shaker. The hydraulic piston actuator is convenient for designing very large force systems or very large stroke systems.

On the other hand, the force is generated through the motion of the actuator rod that is driven and controlled by the flow of hydraulic oil that is energized by being compressed by the mechanical work done by a hydraulic pump. Therefore the performance characteristics of this hydraulic shaker essentially depend on the characteristics of the oil and the operation mechanisms of the equipment. The maximum usable frequency is limited by the resonance of the oil-column formed at both sides of the piston, and typical maximum frequency value is from several tens of Hz to several hundred Hz.



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Figure 4 Schematic diagram of a typical Hydraulic shaker

3.3 AC-servo motor driven shaker

The AC-servo motor driven shaker is equipment that generates linear vibrating motion by converting rotational motion of an AC-servo motor through a 'ball screw & nut' mechanism. Such a mechanism has been widely used as a positioning system (e.g. in machine tools) by slowly driving a motor to place a target at a desired position along the line of axis. By utilising the principle that the linear step of the screw is proportional to the rotated angle of the screw,

relatively low frequency linear vibration can be generated. By controlling the AC-servo motor to make angular rotation proportional to the desired displacement, low frequency vibration is generated. This simple mechanism is very convenient to extend the maximum stroke; longer displacement is achievable just by extending the length of the 'Ball screw'.

On the other hand, when the rotational speed of the screw exceeds a certain limit, the proportional relationship between the rotation angle and the step of the screw is destroyed because of the backlash existing at the contact surfaces of the screw and the nut. Then the generated linear vibration becomes different from the required reference (expressed as a proportion of the angular rotation). As the existence of the backlash is fundamental for the "Ball screw & nut" mechanism, this effect can't be eliminated. The maximum usable frequency is limited by this effect, and has a typical value of several Hz.

In addition, a supporting structure is inevitably required to suspend the ball screw, and the friction at the bearing contact surface sometimes causes difficulty in controlling the motion of an object suspended by the bearing mechanism. This can stimulate a self-excited oscillation through a 'stick-slip' effect by the rapidly varying friction force at the bearing surface.

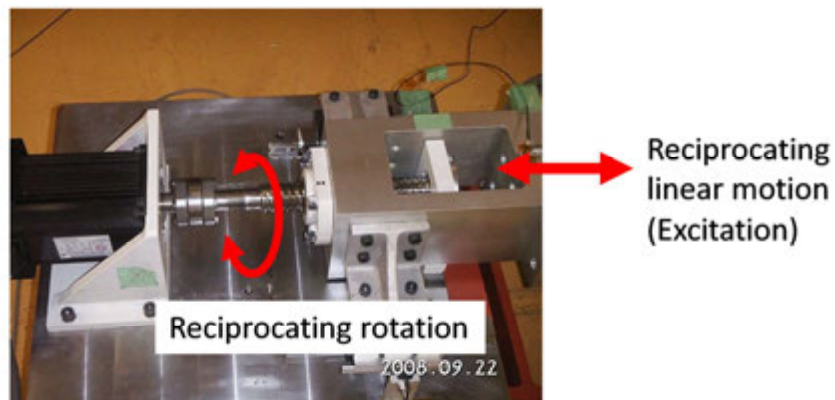
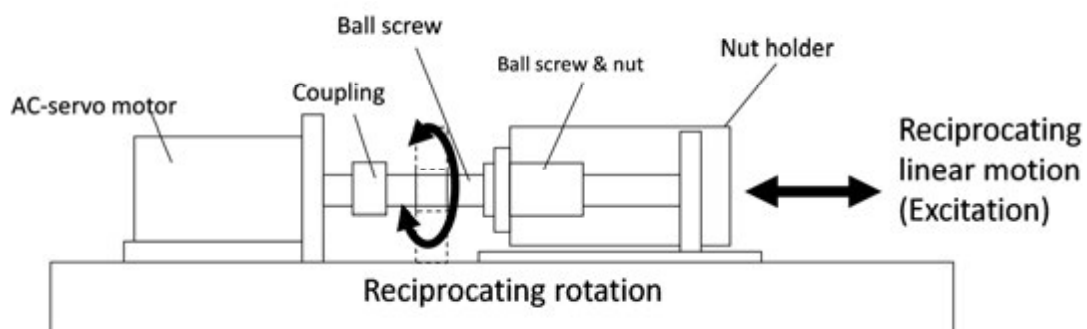


Figure 5 AC-servo motor driven shaker

3.4 Hybrid connection of the shakers

As briefly viewed above, each shaker type has certain features and each has a usable frequency band. For the purpose of realising a wide frequency band shaker, it would be effective to create a hybrid connection of several shakers by taking the key advantage from each shaker, that is to divide the frequency range to a lower and a higher band, and to apply the most appropriate type of shaker to each band, and to control the whole band as a unified single system.

To be more precise, it would be most practical to apply an AC-servo shaker to the frequency band lower than $\sim 1\text{Hz}$ where the displacement is dominant and to apply an ED shaker for the rest of the band higher than $\sim 1\text{Hz}$. Features of these actuators are listed in the table below.

As seen in this table, by adding an AC-servo shaker to cover the very low frequency range of less than $\sim 1\text{Hz}$, it is possible to implement a very wide frequency band shaker system of 0.1Hz to several hundred Hz. The excellent linearity of the ED shaker is also maintained.

Features of the Actuators

(of those having appropriate capacity for the system under consideration)

	ED Shaker	AC-servo motor driven shaker
Type	Linear vibrator A self-contained vibration generator	Rotational motor Reciprocating linear vibration is generated by converting the reciprocating rotational movement using Ball-screw mechanism.
Force	Shock max. $\sim 20\text{kN}$	Shock max. $\sim 20\text{kN}$
Velocity	Shock max. $\sim 2.0\text{m/s}$	Shock max. $\sim 1.0\text{m/s}$
Displacement	Shock max. $\sim 250\text{mm}_{\text{p-p}}$	Shock max. $\sim 1000\text{mm}_{\text{p-p}}$
Frequency	DC to $\sim 1000\text{Hz}$	DC to $\sim 50\text{Hz}$
Dimension, Weight	$\sim \phi 0.4\text{m} \times 1.0\text{m}$ $\sim 500\text{kg}$	$\sim \square 0.4\text{m} \phi \times 1.5\text{m}$ $\sim 300\text{kg}$
Strong point	Linear response up to high freq. (suits for $\sim 1\text{Hz}$ to 1000Hz system)	Larger displacement is easy to realize (suits for DC to $\sim 1\text{Hz}$ system)
Weak point	Possible maximum stroke would be $\sim 300\text{mm}$	Nonlinear and noisy response caused by the Bearing backlash is inevitable.

Table 2 Features of ED shaker and AC-servo shaker

4.0 Hybrid Shaker basic design

4.1 Mechanical structure

The basic mechanical design of the Hybrid Shaker is shown in Figure 6. This equipment generates the required vibration by combined action of the two shakers; an ED shaker is mounted on a moving base plate that is driven by an AC-servo shaker.

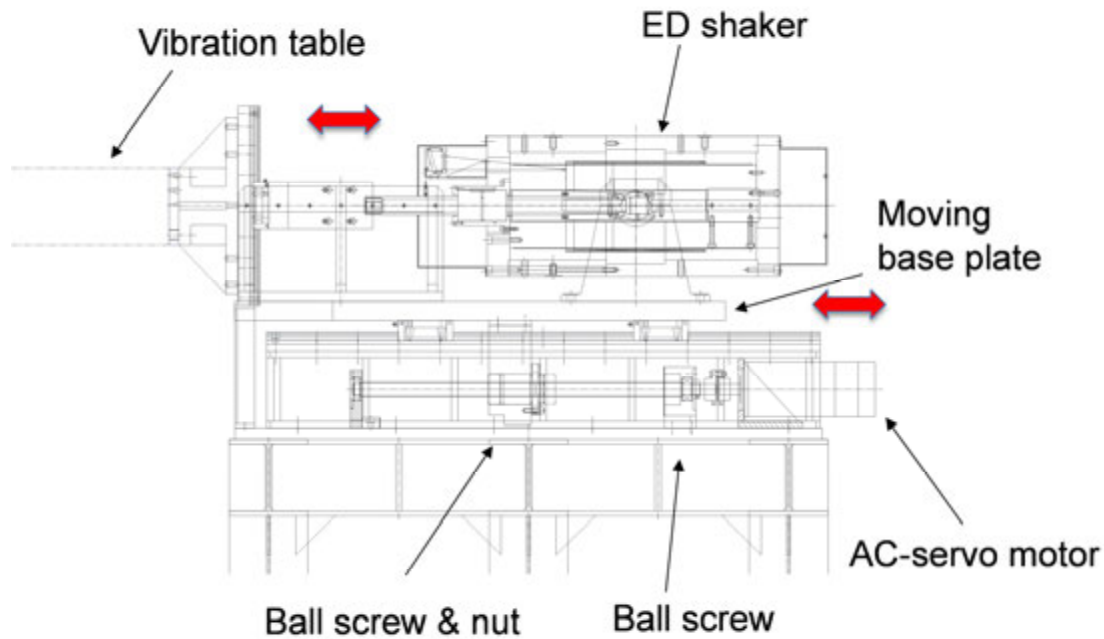


Figure 6 Hybrid Shaker mechanical structure

4.2 Vibration control

A schematic diagram of a vibration control system for the Hybrid Shaker is shown in Figure 7. In this configuration, K2, IMV's vibration controller, controls the system as a whole so that the Hybrid Shaker covers the whole wide bandwidth, by controlling each shaker independently but simultaneously to work at each frequency band. That is, K2 generates two drive signals, one for lower band (AC-servo shaker) and one for the higher band (ED shaker), to replicate the required reference waveform in the motion of the vibration table. The response of the controlled system to the input signals from K2 is detected as the response acceleration signal, and this response is the feedback input to K2. K2 compares the acceleration response with the required reference and modifies the data for the algorithm generating the drive signal to improve the replication accuracy in the next loop.

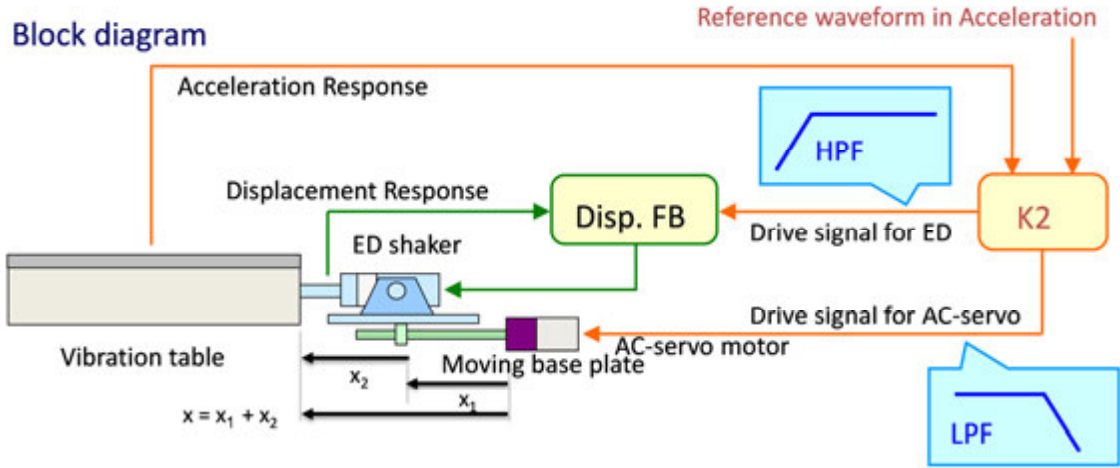


Figure 7 Schematic diagram for Hybrid Shaker controller

For more detail, K2 must work as a controller which deals with two drive signals and one response signal. In this scheme, the controlled system is described as a 2-input, 1-output system as expressed in below:

$$[Y] = [H_{AC} \quad H_{ED}] \begin{bmatrix} D_{AC} \\ D_{ED} \end{bmatrix} \quad (1)$$

where, D_{AC} denotes the drive signal component for the AC-servo and D_{ED} denotes that for the ED shaker, and Y denotes the response signal which is a scalar. H_{AC} is the transfer function from the input point of the AC-servo shaker to the control point on the vibration table, and H_{ED} is that of the ED shaker. These components form the transfer function matrix of the controlled system of (1, 2) size. Here, we have to consider that we are intending to control the two shakers as two perfectly band-separated systems; that is, the transfer function matrix $H(f)$ must be comprising of perfectly band-separated components bounded at some cross-over frequency of f_c :

$$\begin{aligned} 0 < f \leq f_c : & \quad H_{AC} \neq 0, H_{ED} = 0 \\ f_c < f \leq f_{\max} : & \quad H_{AC} = 0, H_{ED} \neq 0 \end{aligned} \quad (2)$$

To ensure that this proposal is implemented, weighting of the transfer function matrix components is executed as below:

$$\begin{bmatrix} H_{AC} & H_{ED} \end{bmatrix} = [W_{LPF} H_{AC} \quad W_{HPF} H_{ED}] \quad (3)$$

where W_{LPF} and W_{HPF} are weighting factors that take below values, and they take a role of an ideal band-separation filter of cross-over frequency of f_c :

$$\begin{aligned} 0 < f \leq f_c : \quad & W_{LPF} = 1, W_{HPF} = 0 \\ f_c < f \leq f_{\max} : \quad & W_{LPF} = 0, W_{HPF} = 1 \end{aligned} \quad (4)$$

To get a drive signal vector D that will generate any given reference R , a desired R is to be substituted for Y in Eq. (1), and the inverse matrix of $H(f)$, $H^{-1}(f)$ should be multiplied from the left-hand side. But since there exists no inverse matrix for this non-square matrix $H(f)$, the Moore-Penrose pseudo-inverse matrix $G(f)$ ($=H^+(f)$), which is a (2,1) size matrix, is used:

$$\begin{bmatrix} D_{AC} \\ D_{ED} \end{bmatrix} = \begin{bmatrix} G_{AC} \\ G_{ED} \end{bmatrix} [R] \quad (5)$$

where, G_{AC} and G_{ED} are the components of the pseudo-inverse matrix $G(f)$ of the transfer function matrix $H(f)$ of the controlled system, and the relationship can be written as:

$$\begin{bmatrix} G_{AC} \\ G_{ED} \end{bmatrix} \begin{bmatrix} H_{AC} & H_{ED} \end{bmatrix} = \begin{bmatrix} 1 & 0 \\ 0 & 1 \end{bmatrix} \quad (6)$$

As seen above, the matrix $G(f)$ has a functionality of equalising the characteristics of the transfer function of the controlled system, it is often called the 'Equalization matrix'.

To practically implement the above control scheme, it is necessary for the ED shaker to behave as a rigid body in the low frequency band governed by the AC-servo shaker. This will enable the ED shaker to transfer all of the vibration generated by the AC-servo shaker to the vibration table. For this purpose, a displacement Feed-Back (FB) loop is inserted as an inner control loop around the ED shaker to fix the armature's nominal position of the ED shaker to a defined fix point on the moving base plate on which the ED shaker is mounted. As a result, the vibration table position x can always be determined as the sum of the AC-servo shaker armature table position x_1 , and the ED shaker armature table position x_2 :

$$x = x_1 + x_2 \quad (7)$$

From (7), by denoting the displacement of each shaker armature position from its nominal position as Δx_1 , Δx_2 , Δx , then:

$$\Delta x = \Delta x_1 + \Delta x_2 \quad (8)$$

and as the velocity of each shaker armature v_1 , v_2 , v is defined as the ratio of these quantities to the time duration Δt , by dividing the equation (8) by Δt ,

$$v = v_1 + v_2 \quad (9)$$

just the same relationship between a_1 , a_2 , a is induced from (9) as,

$$a = a_1 + a_2 \quad (10)$$

Usually, the reference of a vibration test is given by the specification of an acceleration signal that is to be replicated on the vibration table. This hybrid shaker system replicates the required reference acceleration signal 'a' as the sum of the acceleration a_1 and a_2 generated by each shaker, by observing (10). The vibration control itself is performed using the acceleration signal, but the relationships (9), (8), (7) will also stand if the acceleration control is performed with suitable accuracy.

4.3 Necessity for the displacement FB control

Displacement FB control of the ED shaker has been introduced to build a stiff mechanism that transfers the force generated by the AC-servo shaker to the vibration table. In addition to this, the FB control has another important contribution to improve the performance of this system, in which mechanical bearings are utilised as essential and convenient parts. This can be observed by considering the experimental results of a small system used in the feasibility study of the Hybrid Shaker.

In this equipment, the horizontal table is supported by a linear guide that provides movement on a linear, axial line restricted by ball-bearings and a sealing mechanism. When the table moves in the horizontal direction, there is always an associated 'bearing noise', which is relatively large since the mass of the table is relatively small compared to the noise-generating forces caused by the linear guide. This noise is of course not desirable, but difficult to avoid.

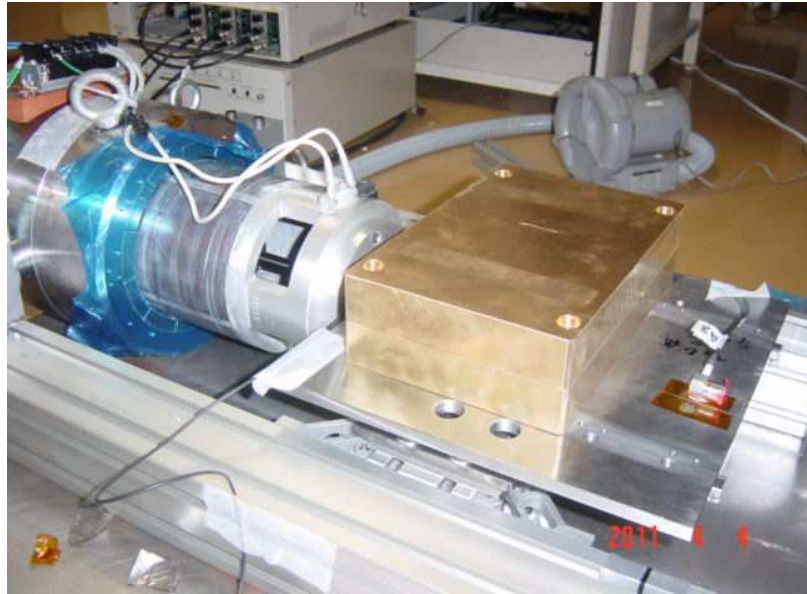


Figure 8 A small horizontal shaker system used for the feasibility study

Most importantly, the fact that the static friction force from the linear guide is considerable, makes it very difficult to control the vibration table; the table does not move at all at first. When a stronger force is applied by the shaker, the table finally starts to move. Once it starts to move, the frictional force shows a sudden and quick decrease, as the dynamic friction force is significantly smaller than the static one. Then the force-friction balance at the table is lost and the table is accelerated in the direction of the applied force. An over-travelling error event of the table can occur and the system will be stopped. The system is unstable and it is very difficult to control such a system that uses a linear guide system, when using a typical vibration controller. To make the control possible, it is necessary for the vibration controller to be able to respond quickly and appropriately to such rapid changes of the frictional force.

A displacement FB controller is useful for this purpose. When a FB controller is inserted as shown in Figure 9, the change of displacement caused by the change of the frictional force is detected. The FB loop works to control the shaker to cancel the excessive or insufficient displacement immediately and in a real-time manner. As a result, when considering the vibration controller which is controlling the system from the perspective of the outer control loop, the controlled system behaves in a stable manner (as if such instability did not exist). Figure 10 shows experimental data demonstrating the effectiveness of the inner FB loop.

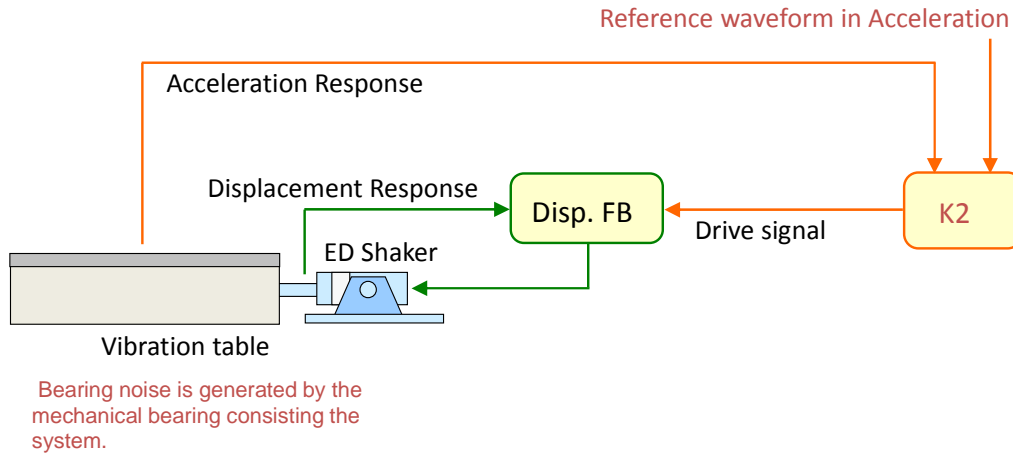


Figure 9 Control system of the small horizontal shaker

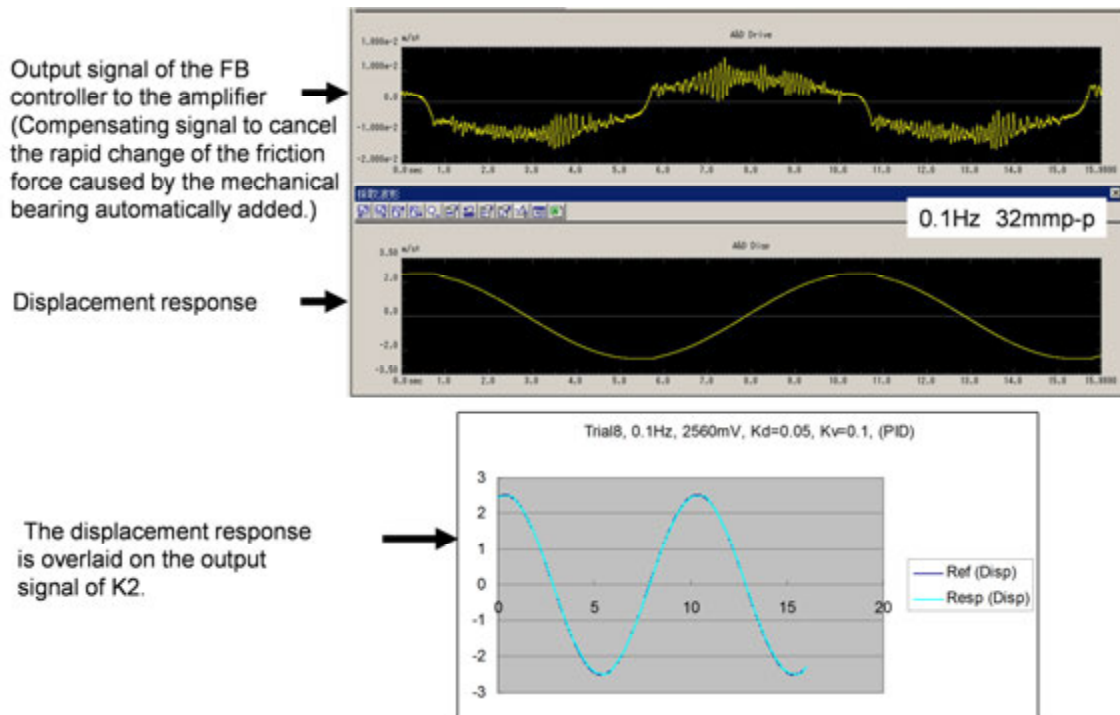


Figure 10 The rapidly varying friction force has been cancelled by the FB control

In the case of the Hybrid Shaker, the FB loop inserted in to the ED shaker control loop is working in just the same way as demonstrated in the example of the small shaker system above. This is greatly contributing to stabilise the characteristics in the low frequency band, which can be disturbed by the existence of noise from the mechanical linear bearings.

5.0 Hybrid Shaker realization

5.1 Earthquake resistance test system DS-3600-24L

This is a system for seismic tests with simultaneous horizontal and vertical excitation. A Hybrid Shaker is implemented in the horizontal exciter part.



Figure 11 Earthquake resistance test system DS-3600-24L

5.2 Examples of experiment

5.2.1 Reference acceleration waveform virtual band-separation

To observe the operation of the control system on actual waveform data, the reference waveform 'a' has been filtered by a band-separating filter and virtually divided into two waveforms, a_1 and a_2 as seen below. Also velocity waveforms and displacement waveforms obtained by integrating the separated acceleration waveforms are shown. The graphs demonstrate how the frequency components are separated in to the two bands, which are that then used to operate the two types of shaker.

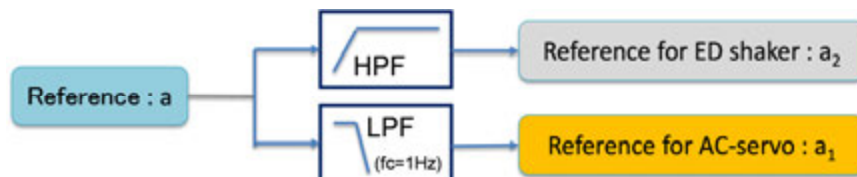


Figure 12a Virtually band-separated acceleration references

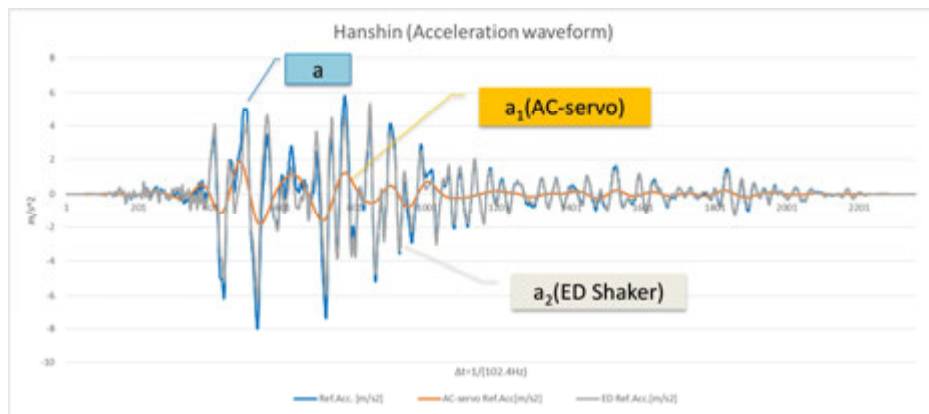


Figure 12b Virtually band-separated acceleration references (Hanshin horizontal NS)

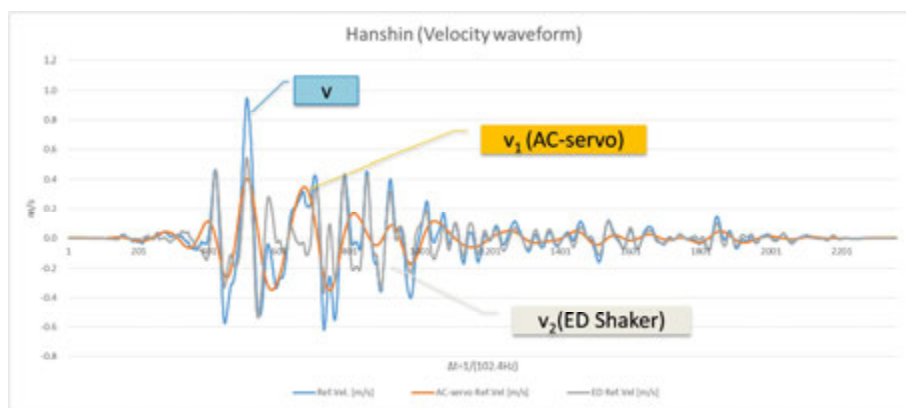


Figure 13 Virtual velocity references achieved by integrating each separate acceleration waveform

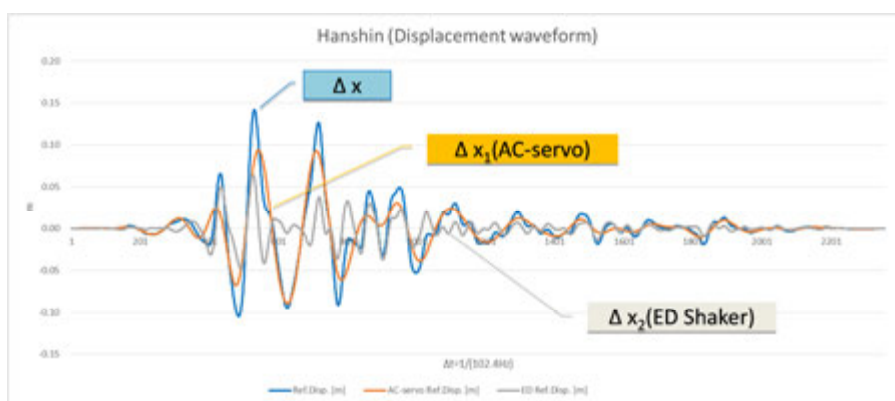


Figure 14 Virtual displacement references achieved by integrating each separated velocity waveform

5.2.2 Actual excitation results

The above examined waveform (Hanshin NS direction) was fed as the reference, and excitation was performed with a 1000kg mass payload mounted on the table.



Figure 15 The system with 1000kg mass mounted

In the Figure 16 the response waveform record is overlaid on the reference, and a zoomed graph of the centre portion of it is shown in Figure 17.

Response acceleration waveform (red) is overlaid on the reference (blue).
Good replication of the given reference (Total rms error 13.7 %)

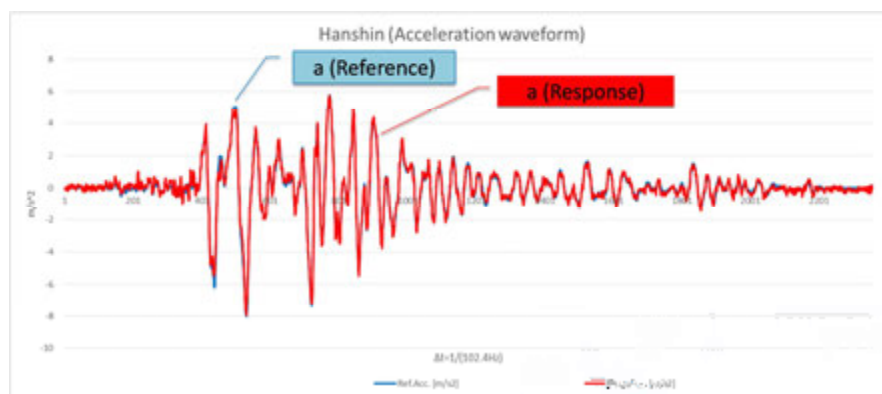


Figure 16 Control result (response waveform with 1000kg mass load mounted)

Enlarged display of the central part of the acceleration waveform
(Total rms error 13.7%)

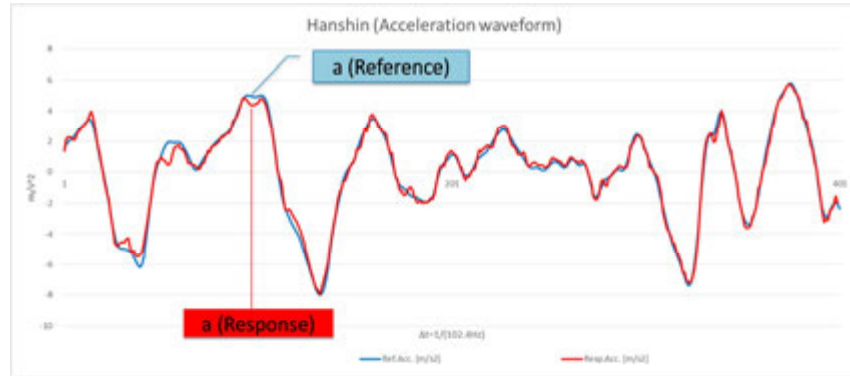


Figure 17 Zoomed centre portion

The actual displacement record of the ED shaker armature detected by the displacement sensor for the FB control in Figure 7 is overlaid on the virtual displacement reference of the ED shaker in Figure 18. In spite of the fact that vibration control is done on the acceleration response, the response displacement waveform is also showing a good match to its reference.

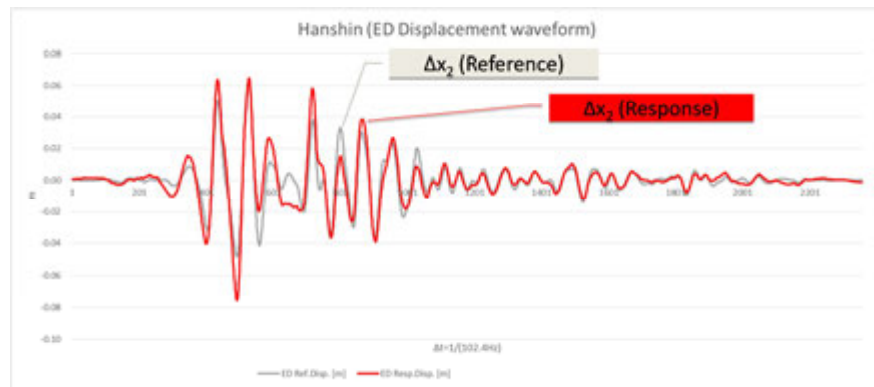


Figure 18 Displacement response compared with the virtual reference (with 1000kg mass load)

6.0 Vertical exciter of DS-3600-24L (supplement)

6.1 Design strategy

Although the Hybrid Shaker design is not employed in the vertical exciter of the DS-3600-24L system, the improvement of the characteristics by FB control is also applied to this part.

From the point of view of geometrical stability, a four vertical shaker configuration (using the same type of shaker) aligned to the corner points of a

square table is preferred for this type of vertical exciter. A 'Multi-point control' technique is suitable to control the four shakers, but in this system, an approach to deal with the four shakers acting as a single shaker is employed. The individual shaker characteristics of each shaker are equalized using the displacement FB inner control loop inserted to each shaker's individual (multi-point) control loop.

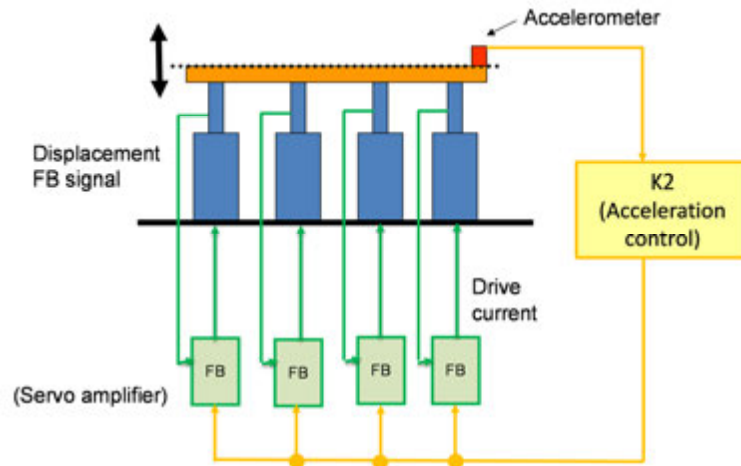


Figure 19 Construction of the vertical exciter of the DS-3600-24L system

6.1 Construction of the control system of the whole system

The schematic diagram of the control system of the DS-3600-24L as a complete system is shown in Figure 20. Centre Positioner (CP) is a FB control to hold the original point of the AC-servo shaker.

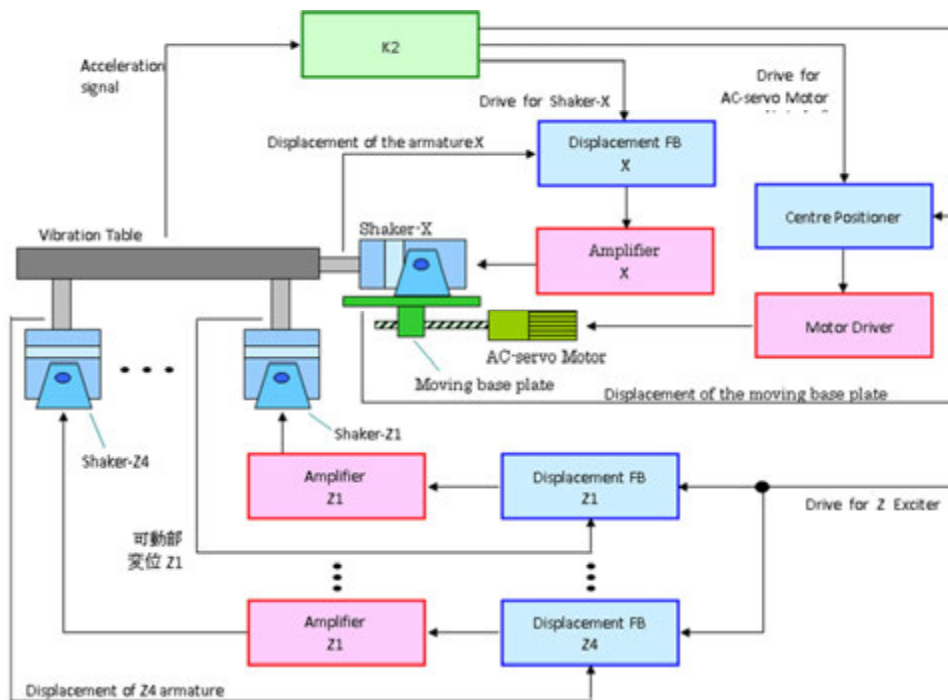


Figure 20 Construction of the control system of the DS-3600-24L

6.2 Control results

Vertical vibration of the simultaneous excitation results of Hanshin (1995) waveform replication by DS-3600-24L with a 1000kg mass load are shown below. Corresponding horizontal results are already shown in the section 5.2.

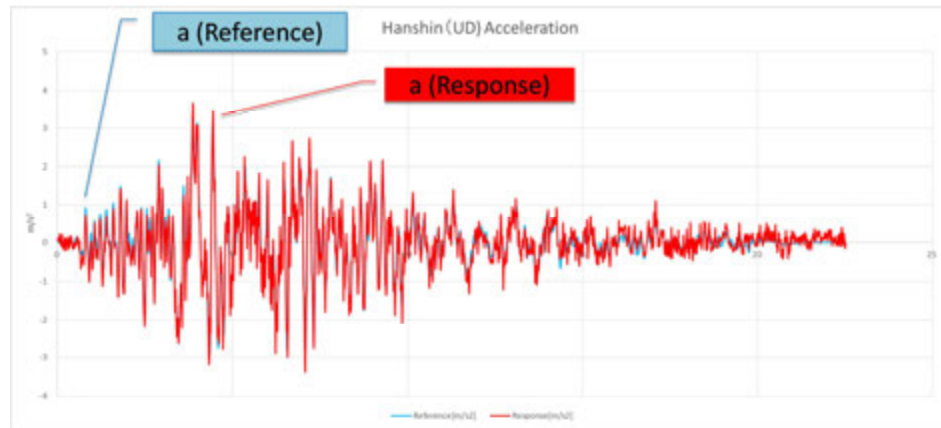


Figure 21 Hanshin waveform (UD: vertical) replicated by DS-3600-24L

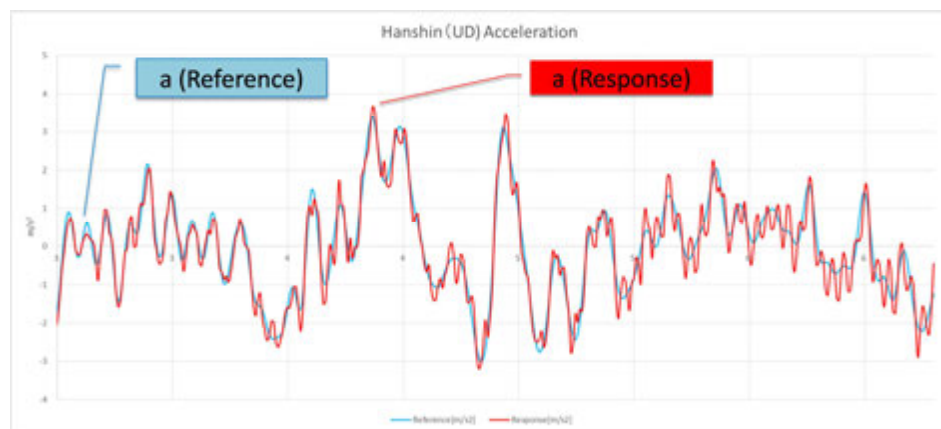


Figure 22 Zoomed centre portion of Figure 20

As seen in the zoomed graphs of the centre portion, there exists an excessive response in the components higher than 10Hz. This excessive error might be caused by a mechanical resonance around the table structure. When more accurate control at a higher bandwidth is desired, introduction of multi-point control, controlling each of the four shakers independently might be useful.

7.0 Conclusion

A newly developed, innovative technology of a Hybrid Shaker system has been explained. The purpose of this technology is the realisation of a very wide-band vibration test system, with very low frequency, high displacement capability. The system is realised in a compact footprint, using two different types of band-separated shakers. The foot print of this system can be $1/3 \sim 1/4$ of that of conventional servo-hydraulic types of system.

It is expected that various types of vibration test that had previously only been possible with a very large test facility are becoming possible to be achieved easily and more accurately through this technology.