Performance Test of Mechanical Snubber for Fast Breeder Reactor

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SANWA TEKKI KOGYO CO., LTD

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With the purpose of developing a radiationproof mechanical snubber for use with the primary loop of the Fast Breeder Reactor, we have manufactured a prototype of mechanical snubber mechanism which is mainly featured a ball bearing screw and rotary flywheel. This snubber has equivalent mass provided by rotating inertia of flywheel connected with a ball bearing screw.

To confirm its performance, we have carried out a series of comprehensive tests such as, vibration test, static workability test, resonance test, and endurance test.

As the outcome of the above tests, the performance of the mechanical snubber is justified to be regarded as an equivalent mass. For instance, in case of the snubber installed with the piping, it is possible to do seismic design under which the piping system is regared as a flexible structure, since the snubber effects to the piping in its direction as a huge mass. Further, the Vibration Test has resulted in that the initial displacement, during which the resisting force is rised, is less than half value of the performance of the hydraulic snubber. This clarified its just performance.

In addition, as to the larger capacity mechanical snubber, those tests did not result in a sufficient performance due to excessive static friction of the mechanism which amplifies the rotating inertia.

This work was performed under the contract between Power Reactor & Nuclear Fuel Development Corporation and Sanwa Tekki Corporation.

CONTENTS

1. Purpose and Policy of Experiment

The vibration-proof mechanical snubber which is to be used for the primary coolant piping loop of the Fast Breeder Reactor is subject to be exposed to a high dose of gamma ray. A hydraulic snubber capable of withstanding a high radiation was developed during the years of 1969 - 1972. But as no oil to withstand such a high radiation as 109R which is considered with a Fast Breeder Reactor is available, development of a mechanical type snubber has come to be considered.

In this experiment, the test purpose was set to found the performance of a vibration-proof snubber out of several types of test specimens in order to evaluate the feasibility of practical use of a mechanical snubber foatured with the resistance mechanism of rotating inertia.

For this purpose, seven types of mechanical snubbers and twelve sets of test specimens were prepared. The type and capacity of these test specimens are as shown in Table 1. It was the policy of this experiment to perform a series of comprehensive with these specimens comprising vibration test, static workability test, resonance test, and endurance test, and thus to draw a performance diagram of the mechanical snubber based on the results of these respective tests.

2. Design of Test Specimens.

The measurements of the exterior shape of the specimens are as represented by Fig. 1, 2, and 3. Fig. 1 represents Specimens No. MSA-1, 2 and 3. Its functional structure is that (10) a flywheel is fitted to (1) a ball screw by (11) a parallel key and then via (12), (14) bearings, it is fixed to (13) a load bearing plate, while on the other part, (6) the piston is connected to (2) the ball nut and is encased into (17), (21) housing cases.

Its kinetic function is deviced that when one side of its (7) ear is fixed to a pipe and the other part to the building, and a vibration is applied to the piston, the ball-screw will rotate by a linear motion of the ball-nut. At this instant, the flywheel which is connected to this ball-screw will generate a resistance caused by its moment of inertia, and this energy will prevent the vibration.

Fig. 2 represents Specimens No. MSA-4 and 5, which have the similar functional mechanism as those of the above mentioned specimens, except that the outer diameter of the flywheel is smaller and an amplifying device is incorporated in between the ball-screw and the flywheel in order to increment the anti-vibration effect.

Fig. 3 shows Specimens No. MSA-6 and 7, which are prepared for the purpose of testing whether the equivalent mass (Me) of the mechanical snubber resonates with the spring, and of describing their resonant curves considering the combination of the specimen and the coil spring as a single system of particles.

As to the design details of these specimens, please refer to the Appendixes at the end of this report.

3. Test Method.

3-1. Static Workability Test.

The test was performed with use of the test equipment as shown by Fig. 4 (flow chart). The thermal expansion and contraction of the piping were assumed in this experiment. The test method was taken so as to move the piston slowly (0.01, 0.05, 0.1, 0.2, 0.3 cm/sec) to measure the resistance of the specimen during this five stages of slow movements of the piston.

3-2. Vibration Test.

The test was performed with the test unit which is represented by Fig. 5 (flow chart). With the combination as shown in Table 2, the specimens were applied with different degree of frequency and force, and thus the displacement appearing on the specimens was measured.

3-3. Endurance Test.

The test was conducted with use of the test equipment as shown by Fig. 5. In this test, the Number of seismic tremors to which the mechanical snubber may possibly be subject was assumed. The vibration length applied to the specimens was based on the following formula:

T = L . fnn. t	T:	Vibrating hour (hr)
	L:	Life of the plant (40 yrs)
	fn:	Times of Earthquake (once/yr)
T = 6.60 hr = 7hr	t:	Seismic hour (10 minute)

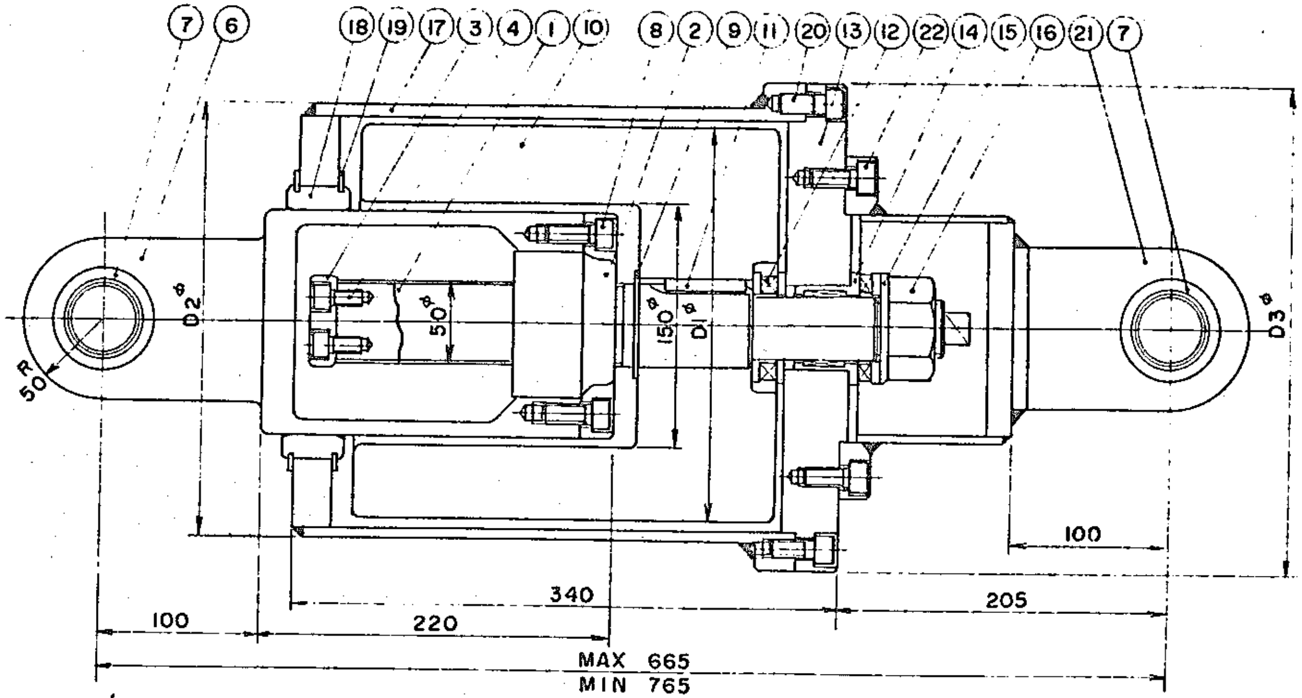
Each one set of Specimens MSA-1 and 5 was applied a vibration of 10Hz for 7 consecutive hours under 5,000 kg vibrating force. Then the tests under the above items of 3-1 and 3-2 were respectively performed, and the results were compared.

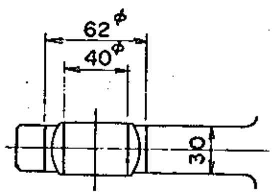
3-4. Resonance Test.

The test was performed with use of the test equipment as shown in Fig. 5. For this test, a combination of the specimen and the coil spring was assumed as a model of one system of particles, and it was to determine whether it was possible to handle the equivalent mass of the specimens No. MSA-6 and 7 the same as the actual mass. For this, it was to describe the resonant curve and to measure the load and displacement of the specimen and the coil spring by changing the frequency in the range of 1 Hz to 20 Hz.

Table 1. Kind of Specimen

Specimen No.	Screw Lead (cm)	Equivalent Mass ($kg.sec^2/cm$)	Speed Increase Ratio	Spring Constant	Remarks	Required No. of Specimen
MSA-1	0.5	1,000	1			2
MSA-2	0.6	1,000	1			2
MSA-3	0.8	1,000	1			2
MSA-4	0.5	500	96		with anplifier	2
MSA-5	0.6	2,500	104		"	2
MSA-6	0.8	0.03179	1	73.6	Resonant test specimen	1
MSA-7	0.8	0.03719	1	97.3	"	1
					Total	12





Unit : m/m

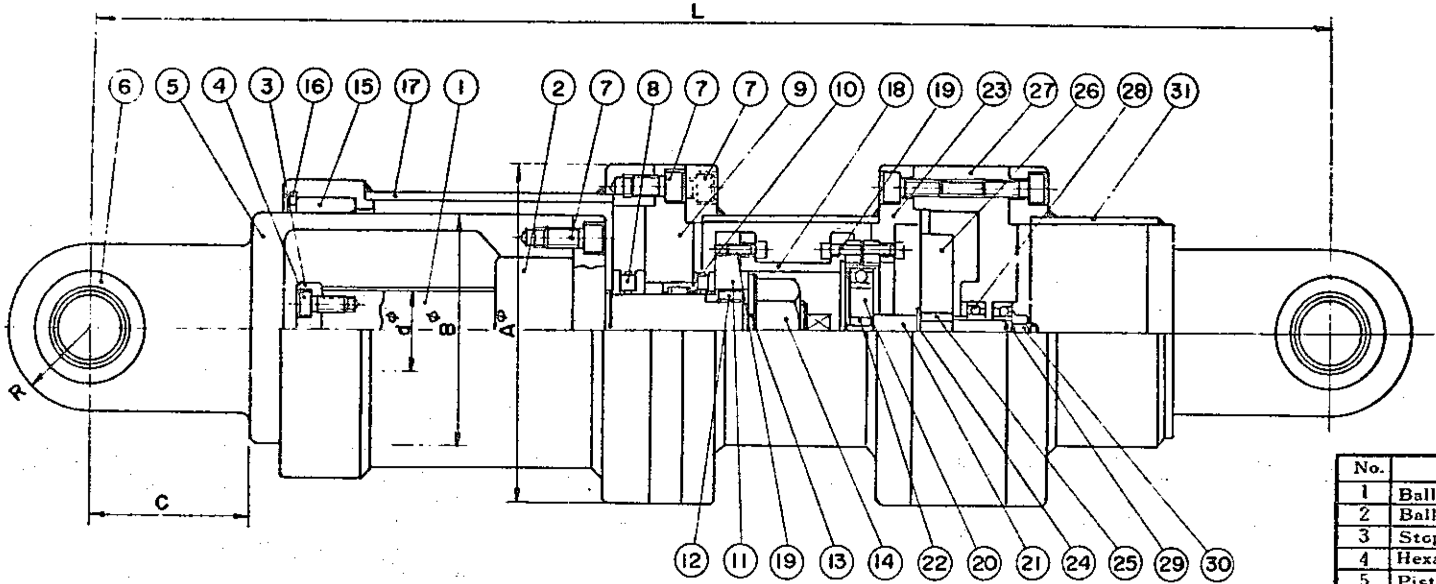
Specimen No.	Equivalent Mass	Design load	D_1	D_2	D_3
M S A 1	$1\times10^3\ \frac{kg\cdot sec^2}{cm}$	10×10^3 kg	242	267.4	300
M S A - 2	〃	〃	263	318.5	350
M S A 3	〃	〃	302	355.6	400

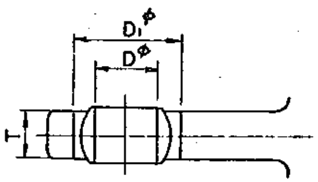
Fig. 1 Exterior Figuration Measurement of Specimen
(MSA-1, 2 and 3)

No.	Name of parts	Req'd No.
1	Ball screw	1
2	Ball nut	1
3	Stopper	1
4	Hexagon soket head bolt	2
5		
6	Piston	1
7	Spherical bearing	1
8	Hexagan soket head bolt	8
9	Snap ring	1
10	Flywheel	1
11	Sunk key	1
12	Thrust bearing	1
13	Load bearing plate	1
14	Compound bearing	1
15	Flat washer	1
16	Locking nut	1
17	Case	1
18	Bush	1
19	Snap ring	1
20	Hexagon soket head bolt	8
21	Ear	1
22	Hexagon soket head bolt	8

Table 2. Vibration Test Combination Table

Freq. (Hz) \ Load	1,000 kg	2,000 kg	3,000 kg	4,000 kg	5,000 kg	8,000 kg	10,000 kg	12,000 kg
0.5	MSA-4	MSA-4	MSA-4 MSA-1 2 3	MSA-1·2·3	MSA-1·2·3	MSA-5	MSA-5	MSA-5
1.0	"	"	"	"	"	"	"	"
3.0	"	"	"	"	"	"	"	"
5.0	"	"	"	"	"	"	"	"
10.0	"	"	"	"	"	"	"	"
15.0	MSA-4	MSA-4	MSA-4 MSA-1 2 3	MSA-1·2·3	MSA-1·2·3	MSA-5	MSA-5	MSA-5



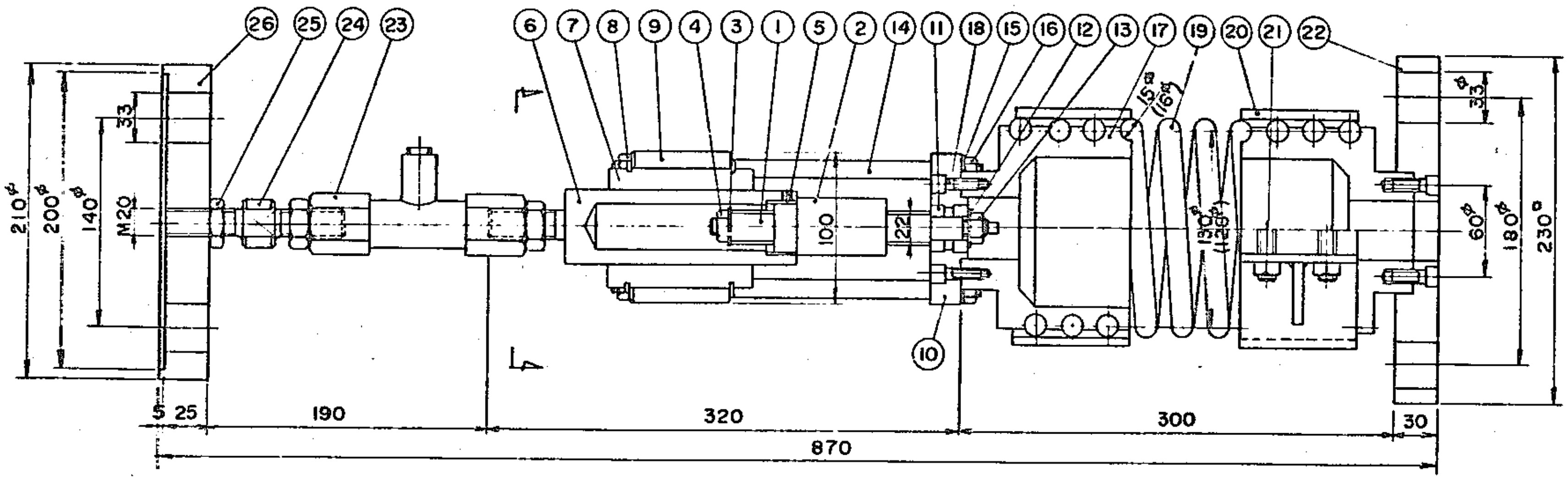


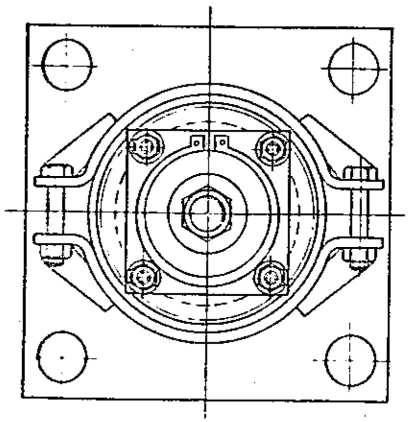
Unit : m/m

Specimen No.	Equivalent Mass (kg·sec²/cm)	Design Load (kg)	d		L		A	B	C	D	D_1	R	T
			Dia	Lead	MAX	MIN							
MSA−4	500	2×10^3	20	5	575	475	125	65	50	20	32	25	16
MSA−5	2500	10×10^3	50	6	875	775	200	140	100	40	62	50	30

Fig. 2 Exterior Dimension of Specimen
(MSA-4 and 5)

No.	Name of parts	Reqd No.
1	Ball screw	1
2	Ball nut	1
3	Stopper	1
4	Hexagon soket head bolt	2
5	Piston	1
6	Spherical bearing	2
7	Hexagon soket head bolt	40
8	Thrust bearing	1
9	Load bearing plate	1
10	Compound bearing	1
11	Flange	1
12	Sunk key	1
13	Flat washer	1
14	Locking nut	1
15	Bush	1
16	Snap ring	1
17	Case	1
18	Joint	1
19	Hexagen socket head bolt	18
20	Amplifier	1
21	Shaft	1
22	Sunk key	1
23	Case	1
24	Snap ring	2
25	Sunk key	1
26	Flywheel	1
27	Case	1
28	Ball bearing	1
29	Flat washer	1
30	Locking nut	1
31	Ear	1





No.	Name of parts	Reqd No.
1	Ball screw	1
2	Ball nut	1
3	Flat washer	1
4	Locking nut	1
5	Check screw	1
6	Piston	1
7	Linear motion bearing	1
8	Snap ring	2
9	Flange	1
10	Load flange	1
11	Ball bearing	2
12	Flat washer	8
13	Locking nut	8

No.	Name of parts	Reqd No.
14	Ty-rod	4
15	Flat washer	1
16	Locking nut	1
17	Spring bearing	2
18	Bolt with hexagonal hole	12
19	Coil spring	1
20	Band	2
21	Hexagonal bolt nut	8
22	Flange	1
23	Load cell	1
24	Stud bolt	4
25	Lock nut	3
26	Flange	1

Fig. 3 Exterior Dimensions of Specimen

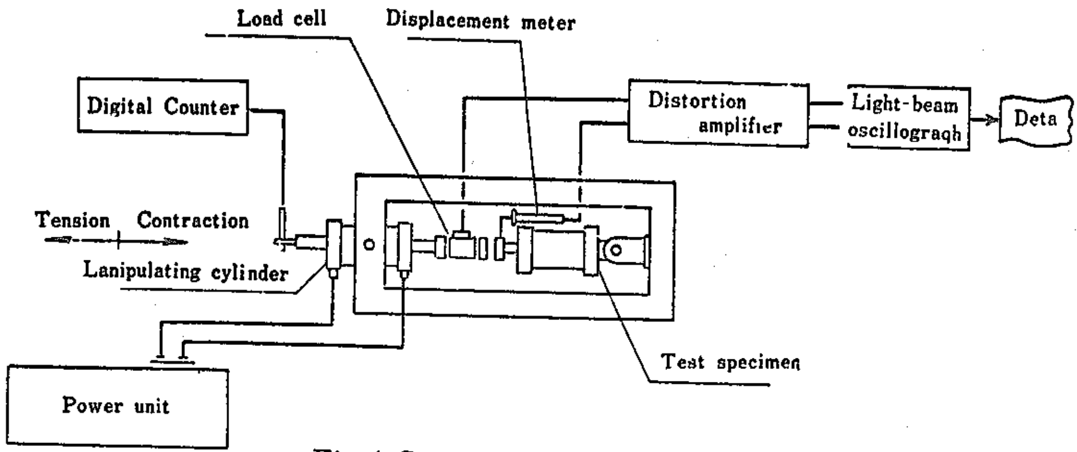


Fig. 4 Static Workability Test Diagram

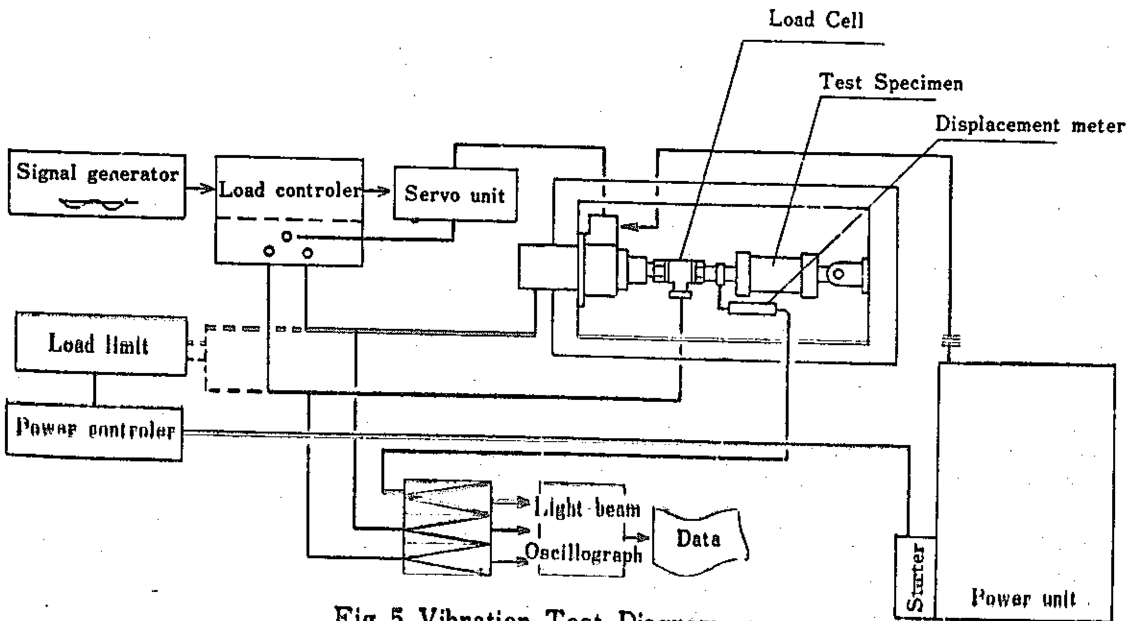


Fig. 5 Vibration Test Diagram

4. Results of Experiment and Consideration.

4-1. Static Workability Resistance.

The actually monitored values of the static workability resistance are given in Table 3 and 4. Table 3-A represents the test in which the test specimen was held vertically demonstrated a satisfactory workability of the specimen when Load of about 30 kg mightier than flywheel weight was added to the specimen. Starting resistance in this case was far smaller than the calculated value (when the friction coefficient (μ) of the bearing is assumed as 0. 1), and it was found out that at least the friction coefficient of the ball screw and the roller bearing was smaller than 0. 1.

Table 3-B shows the workability resistance of the specimen which was held horizontally to the test equipment and moved at a slow speed. According to the result of the test, the static workability resistance gave 2 or 3 times as heavy as the calculated load value. In the vertical test, it was about 1/3 - 1/2 of the calculated values and the friction coefficient at the bearing was sufficiently small. A suspicion of involvement of other factors are considered. It is considerable, as one reason, that since the flywheel of MSA-1 - 3 was of a cylindrical form and of such a design as to hold one end of the flywheel on a bearing, when it was laid horizontally, a bending moment was applied on the wheel's bearing and caused an increased friction resistance, which obstructed a smooth rotation movement. In order to find out the horizontal workability resistance irregulavity its value was meassured in six times test of MSA-1A, and as the results, Table 3-C was given. The figures shown by the table indicate a considerable degree of ununiformity. Although it shows certain concurrence with the calculated values in the area where figures are small.

It was, therefore, reflected that consideration in the design and fabrication of the specimens was not sufficient after all to approximate them to the theoretical values.

As to MSA-4, as it failed to workd even when a heavier load than normally allowable was applied to the specimen during the test, the test was in a fear that the specimen might breakdown.

MSA-5 showed three times as high values as the calculated values in its static workability resistance test. Owing to the resistance of the accelerator which was incorporated in it, there happened no natural fall down.

The static workability resistance after the endurance test showed a similar result as represented by Table 4. Generally, the static workability resistance showed as several times values as the calculated ones, and the values of those which used the amplifier were too large as values of the actually employable ones. This was originated in mainly the starting torque. Originally, it had been planned to measure the workability resistance at each of the five stages of test speed. But due to the difficulty of speed adjustment, the test was performed at merely two stages of speed.

4-2. Performance.

Table 5-9 show the results of vibration tests, and particularly Table 9 represents the results of the vibration test after the endurance test. Fig. 6 is the graph prepared by compensating the figures of Table 5-9 assuming MSA-1 - 3 worked under 4,000 kg and MSA-5 under 10,000 kg respectively.

Looking at the result, the figures in each case are smaller than the maximum allowable values of displacement. In performance, they demonstrate better results than those of the hydraulic type snubber.

MSA-1 - 3 showed an irregularity centering on 0.5 mm displacement, while MSA-5 showed an irregularity at around 1.2 mm displacement. These irregularities were originated from the ununiformity arising from the manufacturing allowance of the specimen. The reason for the absence of the vibration frequency 1 Hz for MSA-1 - 3 in the table was because that the holding frame of the test equipment or the volumic elasticity of the snubber's operating oil and Me of the test specimen resonate and made it difficult to measure the load. As for the post-endurance test performance, it was approximately the same as the pre-endurance test performance in the case of MSA-1A, and in the case of MSA-5A, as the test specimen became so loose and dis-shaped, no tensile direction load was measured.

4-3. Resonance Test Results.

The resonance test results are given in Table 10 and 11. Fig. 7 represents a graph plotted and prepared from both the displacement ratio per each frequency of the coil spring to the mechanical snubber in order to obtain the resonance curves.

Table 10 shows the measurement results of the coil spring's spring constant. Looking at the resonance test results the maximum flexion of the spring is about 15 mm, and the mean value of the spring constant at each 5 mm up to its flexional amount of 15 mm was adopted as the mean spring constant.

The specimens were of identical specifications for MSA-6 and 7, and the equivalent mass give by calculation was Me = 0.03179 kg sec^2/cm. (Refer to Appendices for the calculated equivalent mass of MSA-6 and 7.)

When the specimen's Me is retroactively computed in the reverse order from the test results, and compared with the calculated values, it is as shown in the following table:

Specimen No.	Resonant Vibration Frequency (Hz)	Spring Constant (kg/cm)	Test value Me (kg.sec^2/cm)	Calculation Value Me (kg. sec^2/cm)	Error (%)
MSA-6	7.0	71.3	0.03690	0.03179	16.1
MSA-7	8.0	80.0	0.03170	0.03179	0.3

Looking at the comparative results, not with standing the fact that MSA-6 and 7 are the identical specimens, the errors are large. The reason for this can be considered, judging from the resonant curve shown in Fig. 7, in which the resonant curve of MSA-6 is seen distorted, that the loosening and clattering effect of the specimen as well as the resulting friction might have greatly worked against the specimen.

5. Conclusion.

The above results of the experiment may be summarized in the following manner:

As clearly evidenced from the resonance test results, it is possible to handle the mechanical snubber as an equivalent mass by equialize the energy of its rotating inertia to the energy of its mass, and it was found that its value no much differs from the calculated value.

It is also considered that the mechanical snubber has a superior performance of hydraulic type, and that this mechanical type can be used as a quake-proof snubber in place of the hydraulic snubber. Judging from the results of the performance test, the problematic point in the performance of the mechanical snubber lies in the fact that, as in the case of MSA-1 - 3, where the mechanism is simply to transfer the ball screw rotation to the flywheel, each specimen showed somewhat greater horizontal direction static workability resistance. This, however, was not any basic defect of the mechanical snubber and can be definitely corrected and solved by an improved design and manufacturing, and it is thought possible to apply these experience of this experiment to the actual mechanical snubber to make it really workable with the actual reactor.

On the other part, the model which has the mechanism of transferring the ball screw rotation to the flywheel by means of an amplifier showed a greater static workability resistance, and no helpful result sufficient to be applicable to the actual was obtained. Consequently, unless a small size amplifier with a small starting torque incorporated can be developed, this kind of model will be sopeless to be used as an adequate mechanical snubber.

6. Postscript.

The experiment of this time was purported to find out such a mechanical snubber which has a suitable performance for use with the Fast Breeder Reactor. For this purpose, a series of tests were conducted on the specimens of seven types of mechanical snubbers especially prepared for this experiment. As the results, it was found out that a mechanical snubber could replace the so-called hydraulic snubber as a practical and useful quake-proof snubber, and also a success was obtained to some extent as to its theoretical endorsement.

There are, however, two problems left unresolved. The first one is that, although a prospect was found to attain more or less an acceptable performance by the simple form of transferring the ball screw rotation to the flywheel, it is thought necessary to have a further study as to whether this mechanism is worthwhile to be actually employed, since in an actual use, especially when load is heavy, the flywheel's outer diameter has to be made larger and its weight increases inevitably.

The second problem is how to minimize the static workability resistance of the snubber with uses an amplifier. In this experiment, although a certain degree of success was attained in developing a small size, light weight amplifier model with a larger speed amplifying ratio, when it was incorporated into a specimen of the mechanical snubber, it failed to demonstrate a satisfactory performance.

By equipping an amplifier with the mechanical snubber, the snubber itself becomes light in weight and small in size, and the equivalent mass increases, which naturally makes it handy and fittable into a small space. It is, therefore, considered that a further effort is needed to improve the mechanical snubber toward this goal so that a smaller-size, light-weight mechanical snubber of larger capacity will be developed for use with even a larger capacity fact breeder reactor of the future.

Taking this opportunity, as the last concluding works, we, the writers of this REPORT, cordially express our sincere appreciation of the assistance and suggestion which were rendered to us in connection with this experimental work by Messrs. Kinji Asano, Somei Kato, and Seiya Tamura at the Japan Atomic Power Co., Ltd., and Messrs. Osamu Kawaguchi and Hideki Ito at the Power Reactor & Nuclear Fuel Development Corporation.

Table 3A. Results of Static Workability Test (Vertical Test)

Specimen No.	Flywheel weight (kg)	Specimen weight (kg)	Calculated starting resistance (kg)	1		2		3	
				Speed (cm/sec)	Resistance (kg)	Speed (cm/sec)	Resistance (kg)	Speed (cm/sec)	Resistance (kg)
MSA-1A	68. 3	120	267	0. 97	98	1. 03	98	0. 75	98
MSA-1B	85. 3	120	267	Not meas-red (fell-down)	/	/	/	/	/
MSA-2A	8. 53	160	274	0. 51	115	0. 39	115	/	/
MSA-2B	8. 53	160	274	Not meas-red (fell-down)	118	/	/	/	/
MSA-3A	120. 6	220	285	0. 34	150	0. 45	150	0. 44	150
MSA-3B	120. 6	220	285	Note meas-red (fell-down)	148	/	/	/	/

Note: Refer to Appendices for the calculated starting resistance.

Table 3B. Static Workability Test Results

(Horizontal Test)

Specimen No.	Flywheel Weight	Specimen Weight	Calculated Resistance	1			
				Speed	Resistance	Speed	Resistance
	(kg)	(kg)	(kg)	(cm/sec)	(kg)	(cm/sec)	(kg)
MSA-1A	68. 3	120	347	0. 04	750	0. 1	1, 000
MSA-1B	68. 3	120	347	0. 04	1, 000	0. 1	980
MSA-2A	85. 3	160	404	0. 05	1, 250	0. 1	1, 500
MSA-2B	8. 53	160	404	0. 05	1, 100	0. 1	1, 000
MSA-3A	120. 6	220	428	0. 05	880	0. 1	1, 000
MSA-3B	120. 6	220	428	0. 05	750	0. 1	1, 000
MSA-4A	0. 16	25	628	/	/	/	/
MSA-4B	0. 16	25	628	/	/	/	/
MSA-5A	0. 44	45	1, 570	0. 05	4, 200	0. 1	4, 500
MSA-5B	0. 44	45	1, 570	0. 1	4, 600	0. 1	4, 400

Note: Refer to Appendices for computation of calculated starting resistance.

Table 3C. Irregularity of Horizontal Static Workability Test of MSA-1A

Test Frequency		1	2	3	4	5	6
Tensile side	Speed (cm/sec)	1. 18	0. 55	0. 59	0. 11	1. 15	1. 37
	Resistance (kg)	1040	590	440	300	300	890
Contraction side	Speed (cm/sec)	0. 89	0. 64	0. 58	0. 49	1. 18	1. 08
	Resistance (kg)	1040	590	590	590	590	890

Table 4. Results of Static Workability Test After Endurance Test

Specimen No.	Flywheel weight (kg)	Specimen weight (kg)	Calculated resistance (kg)	1		2	
				Speed (cm/sec)	Resist (kg)	Speed (cm/sec)	Resist (kg)
MSA-1A	68.3	120	347	0.05	820	0.1	950
MSA-5A	0.44	45	1,570	0.05	3,600	0.1	6,800

Table 5

Specimen No.	Frequency (HZ)	Load (kg)		Displacement (mm)	
		Tension	Compression	Tension	Compression
MSA−IA	0.5	3000	3000	0.7	0.7
	1.0				
	3.0	3000	3000	0.5	0.5
	5.0	2700	2700	0.4	0.4
	10	2700	2700	0.4	0.4
	15	2700	2700	0.4	0.4
MSA−1A	0.5	3750	4250	0.7	0.8
	1.0				
	3.0	3750	4350	0.5	0.6
	5.0	3750	4250	0.5	0.6
	10	4250	4250	0.5	0.5
	15	4250	4250	0.5	0.5
MSA−1A	0.5	5500	4900	1.2	1.1
	1.0				
	3.0	6000	5100	0.7	0.7
	5.0	6000	5250	0.7	0.7
	10	5270	5000	0.6	0.6
	15	5000	5000	0.6	0.6
MSA−1B	0.5	3250	3750	0.8	0.8
	1.0				
	3.0	3250	3750	0.7	0.7
	5.0	3100	4500	0.7	0.7
	10	3000	3750	0.7	0.7
	15	3250	3750	0.7	0.7
MSA−1B	0.5	4500	5000	0.8	0.8
	1.0				
	3.0	3250	4000	0.7	0.7
	5.0	4000	4500	0.7	0.7
	10	4000	4500	0.7	0.7
	15	4000	4000	0.7	0.7
MSA−1B	0.5	5000	5400	0.8	0.8
	1.5				
	3.0	4500	5000	0.7	0.7
	5.0	5000	5500	0.7	0.7
	10	5500	5500	0.7	0.7
	15	5000	5250	0.7	0.7

Note: This table is applicable to Table 6, Table 7, Table 8, Table 9

Table 6

Specimen No.	Frequency (HZ)	Load (kg)		Displacement (mm)	
		Tension	Compression	Tension	Compression
MSA−2A	0.5	2750	3000	0.7	0.7
	1.0				
	3.0	3250	3500	0.5	0.5
	5.0	3000	3250	0.5	0.5
	10	3000	3250	0.5	0.5
	15	3000	3250	0.5	0.5
MSA−2A	0.5	4000	4500	1.0	1.0
	1.0				
	3.0	3750	4250	0.6	0.6
	5.0	3750	4000	0.6	0.6
	10	4000	4500	0.6	0.6
	15	4000	4500	0.6	0.6
MSA−2A	0.5	5500	5500	1.3	1.3
	1.0				
	3.0	4900	5200	0.7	0.7
	5.0	5000	5750	0.7	0.7
	10	5000	5500	0.7	0.7
	15	5200	5500	0.7	0.7
MSA−2B	0.5	3500	3200	0.8	0.8
	1.0				
	3.0	3500	3000	0.7	0.7
	5.0	3200	3000	0.7	0.7
	10	3000	2800	0.6	0.6
	15	3000	2800	0.6	0.6
MSA−2B	0.5	4000	4000	0.8	0.8
	1.0				
	3.0	4000	4500	0.7	0.7
	5.0	4000	4500	0.7	0.7
	10	4000	4000	0.7	0.7
	15	4000	4000	0.7	0.7
MSA−2B	0.5				
	1.0				
	3.0				
	5.0				
	10				
	15				

Table 7

Specimen No.	Frequency (HZ)	Load (kg)		Displacement (mm)	
		Tension	Compression	Tension	Compression
MSA−3A	0.5	3500	2500	1.2	0.8
	1.0				
	3.0	2500	3500	0.5	0.3
	5.0	2500	3000	0.5	0.3
	10	3000	3000	0.5	0.3
	15	3000	3000	0.5	0.3
MSA−3A	0.5	5500	3500	1.7	1.3
	1.0				
	3.0	3500	4500	0.6	0.4
	5.0	3500	4500	0.6	0.4
	10	4000	4000	0.5	0.5
	15	4000	4000	0.5	0.5
MSA−3A	0.5	5500	2500	1.3	1.1
	1.0				
	3.0	5200	6600	0.7	0.8
	5.0	5000	6200	0.7	0.8
	10	5500	5500	0.7	0.7
	15	5500	5500	0.7	0.7
MSA−3B	0.5	3000	3000	0.8	0.7
	1.0				
	3.0	3000	3500	0.3	0.2
	5.0	3000	3500	0.3	0.2
	10	3000	3500	0.3	0.2
	15	3700	3700	0.2	0.2
MSA−3B	0.5	3700	4500	1.2	0.9
	1.0				
	3.0	3700	4800	0.4	0.3
	5.0	3700	5000	0.4	0.3
	10	4000	4200	0.4	0.4
	15	4000	4000	0.4	0.4
MSA−3B	0.5	6000	6000	1.6	1.4
	1.0				
	3.0	5000	6500	0.5	0.4
	5.0	5000	6500	0.5	0.5
	10	5000	5500	0.5	0.5
	15	5000	5500	0.5	0.5

Table 8

Specimen No.	Frequency (HZ)	Load (kg)		Displacement (mm)	
		Tension	Compression	Tension	Compression
MSA−5A	0.5	4000	7500	0.8	0.9
	1.0	4000	7000	0.8	0.8
	3.0	4000	8000	0.7	1.0
	5.0	4000	7500	0.7	0.9
	10	4000	8500	0.7	1.0
	15	4000	8000	0.7	1.0
MSA−5A	0.5	5500	11000	1.0	1.5
	1.0	5500	8500	0.9	1.1
	3.0	5500	8500	0.9	0.9
	5.0	5500	8000	0.9	0.9
	10	6500	6500	0.9	0.9
	15	6000	10000	0.9	1.1
MSA−5A	0.5	6000	13000	1.3	1.5
	1.0	4000	13500	0.6	1.0
	3.0	2500	10000	0.6	1.0
	5.0	2500	10000	0.6	1.0
	10	3500	11500	0.6	1.1
	15	3500	11500	0.6	1.1
MSA−5B	0.5	4000	4500	0.8	0.9
	1.0	3800	4000	0.8	0.8
	3.0	3500	4000	0.7	0.7
	5.0	3500	4500	0.7	0.7
	10	4500	4000	0.7	0.7
	15	5000	3000	0.7	0.7
MSA−5B	0.5	7500	7500	1.6	1.1
	1.0	8000	9000	1.6	1.1
	3.0	8000	6000	1.0	0.8
	5.0	8000	7500	1.0	0.8
	10	8000	8000	1.0	0.8
	15	9000	8500	1.0	0.8
MSA−5B	0.5	7500	8000	1.6	1.3
	1.0	7000	7800	1.6	1.2
	3.0	7500	7500	1.0	1.0
	5.0	7500	8000	1.0	1.0
	10	7500	7500	1.0	1.0
	15	8000	8000	1.0	1.0

Table 9

Specimen No.	Frequency (HZ)	Load (kg)		Displacement (mm)	
		Tension	Compression	Tension	Compression
MSA−1A	0.5	3000	3000	0.7	0.7
	1.0				
	3.0	3000	3000	0.4	0.4
	5.0	3500	3200	0.4	0.4
	10	3500	3500	0.4	0.4
	15	3500	3500	0.4	0.4
MSA−1A	0.5	4500	4500	0.8	0.8
	1.0				
	3.0	5000	6000	0.6	0.6
	5.0	3500	3500	0.5	0.5
	10	4000	4000	0.5	0.5
	15	4000	4000	0.5	0.5
MSA−1A	0.5	5000	5000	0.8	0.8
	1.0				
	3.0	5000	6000	0.6	0.6
	5.0	5000	5500	0.6	0.6
	10	5000	5500	0.6	0.6
	15	5500	5500	0.6	0.6
MSA−5A	0.5	0	5500	0.4	0.6
	1.0	0	6000	0.4	0.6
	3.0	0	6000	0.2	0.6
	5.0	0	5500	0.2	0.6
	10	0	6000	0.2	0.6
	15.	0	4500	0.2	0.6
MSA−5A	0.5	0	6500	0.4	0.6
	1.0	0	6500	0.4	0.6
	3.0	0	6500	0.4	0.6
	5.0	0	6500	0.4	0.6
	10	0	6500	0.4	0.6
	15	0	6500	0.5	0.6
MSA−5A	0.5	1500	10500	0.6	1.0
	1.0	1000	9000	0.5	1.0
	3.0	1000	9000	0.5	1.0
	5.0	1000	9500	0.5	0.9
	10	1500	9500	0.5	0.9
	15	2000	9000	0.5	0.9

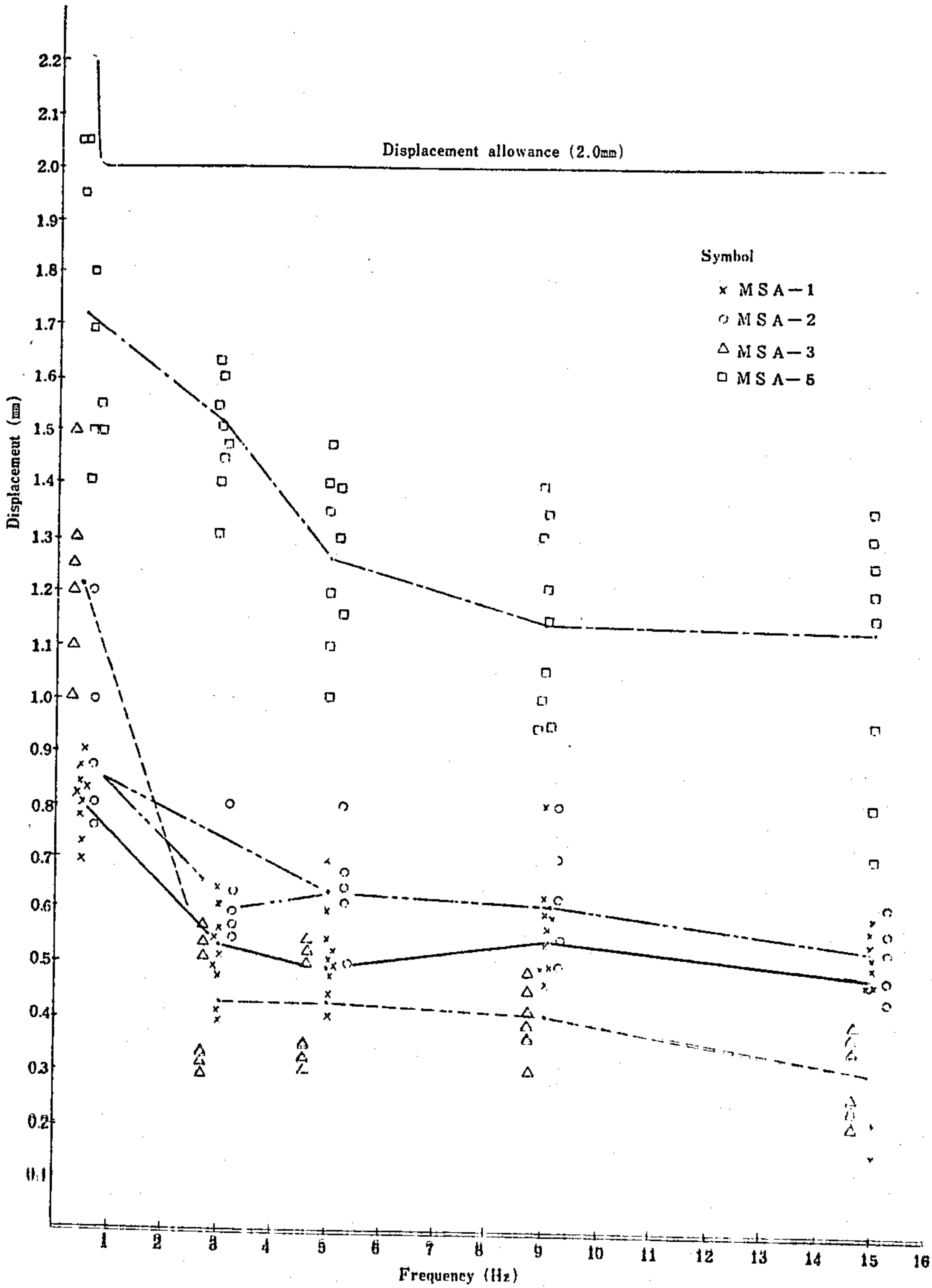


Fig. 6 Vibration Test Results

Table 10. Spring Constant Measurement

Deflection (mm)	5	10	15	20	25	30	Mean spring constant (kg/mm)
MSA-6	38 (7.6)	71 (6.6)	107 (7.2)	138 (6.2)	172 (6.8)	208 (7.2)	7.13
MSA-7	39 (7.8)	80 (8.2)	120 (8.0)	158 (7.6)	202 (8.8)	242 (8.0)	8.0

Table 11. Resonance Test Results

Total amplitude (mm)

Frequency (HZ)	MSA-6			MSA-7		
	Snubber	Spring	Ratio	Snubber	Spring	Ratio
1.0	5.4	2.6	0.48	6.0	0.8	0.13
2.0	5.4	2.6	0.48	6.0	0.9	0.15
3.0	5.8	2.6	0.45	6.2	1.6	0.26
4.0	5.8	2.8	0.48	6.2	1.9	0.31
5.0	6.0	3.6	0.6	6.4	3.0	0.45
6.0	6.0	8.0	1.33	6.6	6.2	0.94
7.0	6.2	10.2	1.65	7.0	14.8	2.11
8.0	6.4	6.6	1.03	6.2	14.8	2.39
8.5	6.6	6.8	1.03	6.6	14.8	2.24
9.0	7.2	7.2	1.0	6.6	12.4	1.89
9.5	6.8	7.0	1.03	6.6	9.0	1.36
10.0	6.0	6.0	1.0	6.2	7.2	1.16
10.5	5.6	5.0	0.89	5.6	5.6	1.0
11.0	5.6	4.4	0.79	5.6	4.7	0.84
12.0	4.4	3.6	0.82	4.4	3.8	0.86
13.0	3.8	3.2	0.84	4.0	3.0	0.75
14.0	3.4	2.4	0.71	3.4	2.6	0.76
15.0	3.0	2.2	0.73	3.0	2.2	0.73
16.0	2.7	2.0	0.74	2.6	1.8	0.69
17.0	2.4	1.5	0.63	2.4	1.7	0.71
18.0	2.2	1.4	0.64	2.2	1.5	0.68
19.0	2.0	1.3	0.65	2.0	1.3	0.65
20.0	1.8	1.1	0.61	1.8	1.0	0.56

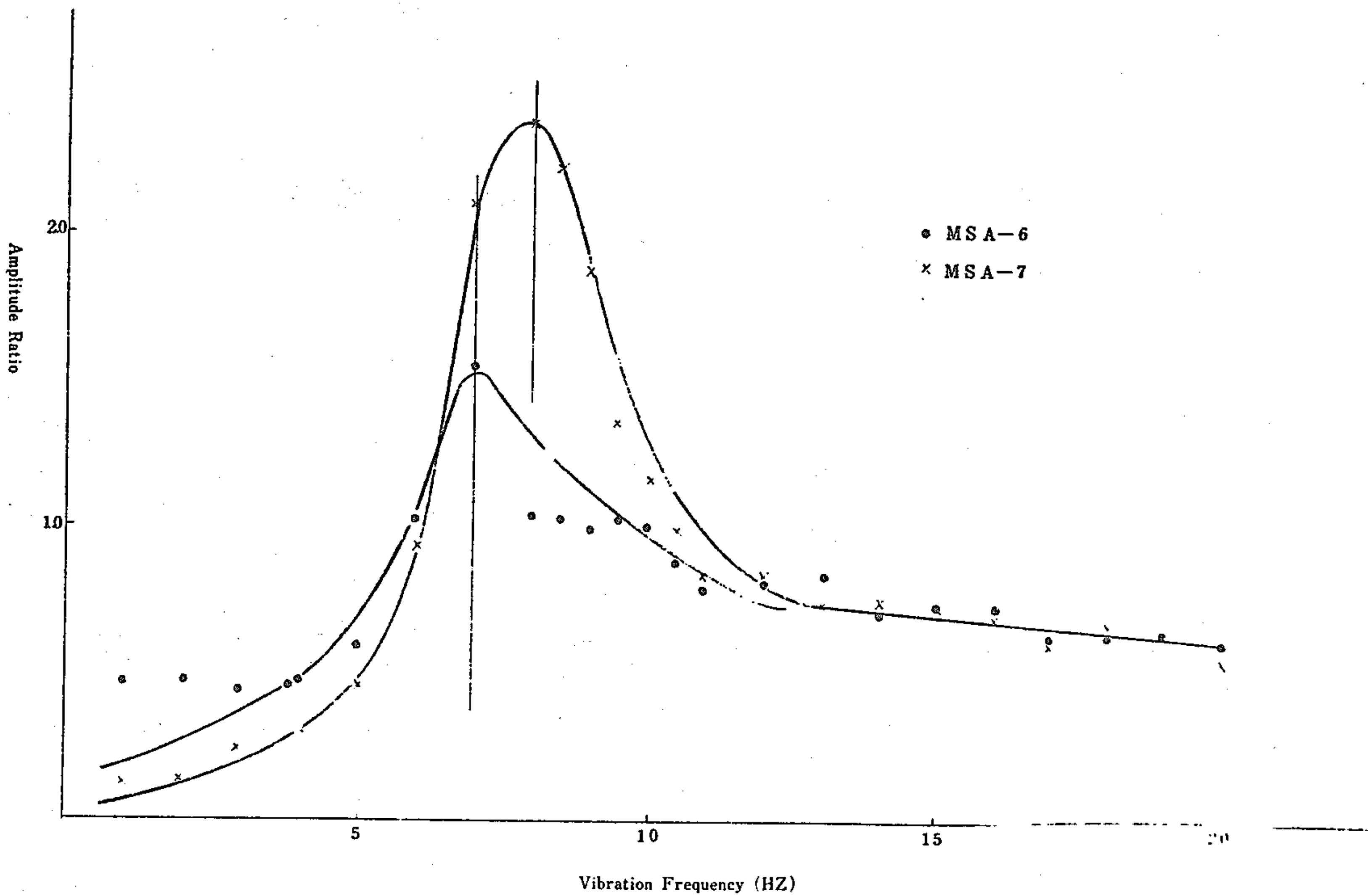


Fig. 7 Resonance Test Results

APPENDICES

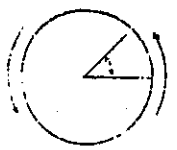
Detailed Design of Specimens

I. Theoretical Basis of Mechanical Snubber

I. Relation between Displacement and Load

A consideration was developed to see how displacement would occur on a mechanical sunbber by the force of inertia of the ball screw lead (pitch) and the flywheel when the mechanical snubber is applied a vibration-load of certain sine-waves.

1-1. Motion of Flywheel



In the rotation angle ϕ of a rigid body:

$$N = I\frac{d^2\phi}{d\,t^2} \quad \cdots\cdots\cdots\cdots \quad (1)$$

Motion of a rigid body with a fixed axis

$$I = \int r^2\, dm$$

r: Radius

m: Mass

1-2. Motion of Ball Screw.

From the relation between the screw's thrust and torque:

$2\pi \cdot N = \ell \cdot P$

$$\therefore N = \frac{P\sin\omega t \cdot \ell}{2\,\pi} \quad \cdots\cdots\cdots\cdots \quad (2)$$

$P\sin\omega t$: Input frequency

ℓ: Screw's lead

N: Torque

1-3. Relative Motion of Screw and Flywheel.

From Equation (1) and (2):

$$\frac{d^2\,\phi}{d\,t^2} = \frac{P\,\ell}{2\,\pi\,I}\ \mathrm{Sin}\omega t$$

$$\frac{d\,\phi}{d\,t} = \frac{P\,\ell}{2\,\pi\,I\omega}\ \mathrm{Cos}\omega t + C_1$$

$$\phi = \frac{P\ell}{2\pi I\omega^2} \text{Sin}\omega t + C_1 + C_2 \quad \cdots\cdots\cdots\cdots\cdots (3)$$

Assuming that the vibration frequency is sine wave motion, give Equation (3) the condition of $t = 0$, $\phi = 0$, then,

$$\phi = \frac{P\ell}{2\pi I\omega^2} \text{Sin}\omega t + C_1 t \quad \cdots\cdots\cdots\cdots\cdots (4)$$

Giving Equation (4) the condition of ϕ ($t = \frac{1}{2}$) = 0, and then from $C_1 \ t = 0$

$$\phi = \frac{P\ell}{2\pi I\omega^2} \text{Sin}\omega t \quad \cdots\cdots\cdots\cdots\cdots (5)$$

On the other part, the relation between the ball screw's axial displacement and the rotation angle of the flywheel:

2π ; ℓ = ϕ; d(t)

$$\therefore \ d(t) = \frac{\ell\phi}{2\pi} \quad \cdots\cdots\cdots\cdots\cdots (6)$$

d(t) : Screw displacement

ℓ : Screw lead

ϕ : Rotation angle of flywheel

Assuming the period as T, $\omega = \frac{2\pi}{T}$ $\cdots\cdots\cdots\cdots\cdots$ (7)

Substituting it with the equations of (5), (6) and (7),

$$d(t) = \frac{P\ell^2 \ T^2}{16\pi^4 I} \text{Sin} \left(\frac{2\pi}{T} \cdot t\right) \quad \cdots\cdots\cdots\cdots\cdots (8)$$

The maximum of the absolute values of d (t) in Equation (8) is at the time of t = T/4, 3T/4, and assuming the value at that time as d, it is;

$$d = \frac{P\ell^2}{16\pi^4 \pm f^2} \quad \cdots\cdots\cdots\cdots\cdots (9)$$

Provided that,

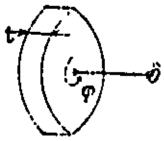
$$f = I/T$$

$$I = 1/2\pi\rho \ t R^4$$

2. Relation between Inertia Force and Equivalent Mass in Mechanical Snubber.

Assume the rotary inertia force of the flywheel as an equivalent and apparent mass effectively working in the support direction of the snubber.

2-1. Energy (E) Generated When Snubber Works Only by δ.



$$\frac{1}{2}(I\phi^2 + M\delta^2) = \frac{1}{2}(I(\frac{\phi}{\delta})^2 + M)\delta^2 = E$$

Energy (e) when the general mass ME moves only by δ, it is, $E = \frac{1}{2} Me\,\delta^2$. (Here, Me is an equivalent mass)

$$\frac{1}{2}(1\ (\frac{\phi}{\delta})^2 + M)\delta^2 = \frac{1}{2}Me\,\delta^2$$

$$Me = I\ (\frac{\phi}{\delta})^2 + M \quad \cdots\cdots\cdots\cdots \quad (10)$$

2-2. Relation between Ball Screw Lead and Flywheel.

$$\phi = \frac{2\pi}{\ell}\delta \quad \cdots\cdots\cdots\cdots \quad (11)$$

From Equation (10) and (11),

$$Me = \frac{4\pi^2}{\ell^2}\ I + M$$

Here, from $\frac{4\pi^2}{\ell}\ I \gg M$

$$Me = \frac{4\pi^2}{\ell^2}\ I \quad \cdots\cdots\cdots\cdots \quad (12)$$

Summarizing the theory of the mechanical snubber,

it is, $P = \frac{16\pi^4 f^2 d}{\ell^2} I \quad \cdots\cdots\cdots\cdots \quad (9)$

From Equation (12), $Me = \frac{4\pi^2}{\ell^2} I \quad \cdots\cdots\cdots\cdots \quad (12)$

From Equation (9) and (12), $\frac{P}{Me} = 4\pi^2 f^2\ d \quad \cdots\cdots\cdots\cdots \quad (13)$

P = Kg (load)

Me = $Kg \cdot sec^2/cm$ (equivalent mass)

d = cm

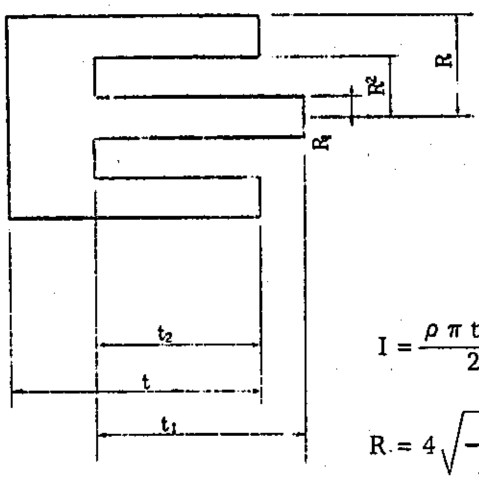
f = 1/sec

II. Design of Specimens

1. MSA-1, 2 and 3.

The performances of MSA-1, 2 and 3 are all identical, while the configuration measurements of flywheel differ from each other because of the difference of lead in each ball screw. The design of flywheel at the time of $Me = 1{,}000\ kg \cdot sec^2/cm$,

$Me = \frac{4\pi^2}{\ell^2} I$		Requirement (I) $kg \cdot cm/sec^2$
MSA-1	$\ell = 0.5$ cm	$I = 6.339$
MSA-2	$\ell = 0.6$ cm	$I = 9.128$
MSA-3	$\ell = 0.8$ cm	$I = 16.23$



$R_1 = 2.5$ cm

$R_2 = 7.5$

$t = 26$

$t_1 = 30.5$

$t_2 = 17.5$

$$I = \frac{\rho \pi t R^4}{2} + \frac{\rho \pi t_1 R_1^4}{2} - \frac{\rho \pi t_2 R_2^4}{2}$$

$$R = \sqrt[4]{\frac{2I}{\rho \pi t} - \frac{t_1 R_1^4}{t} + \frac{t_2 R_2^4}{t}}$$

MSA-1	$R = 12.1049$ cm
MSA-2	$R = 13.1608$ cm
MSA-3	$R = 15.0805$ cm

2. MSA-4 and 5.

	Ball screw lead (cm)	Acceleration ratio	Me $Kg \cdot sec^2/cm$	Width of flywheel (cm)
MSA-4	0.5	96	500	1.0
MSA-5	0.6	104	2,500	2.0

$I = \frac{Me\,\ell^2}{4\pi^2}$ $\qquad \ell = L/G$ $\qquad$ L: Ball screw lead (cm)

G: Acceleration ratio

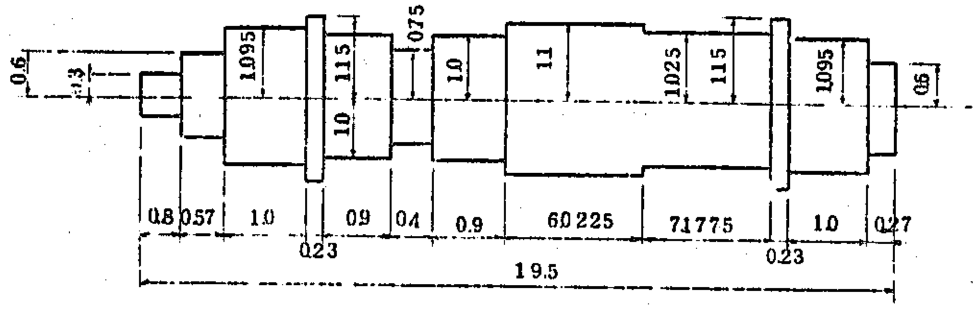
$R^4 = \frac{2I}{\rho \pi t}$ $\qquad \rho = \frac{7.85}{980} \times 10^{-3}\ kg \cdot sec^2/cm^4$

MSA-4 R = 2.28013 cm

MSA-5 R = 3.0219 cm

3. Equivalent Mass of Test Specimens MSA-6 and 7.

3-1. Flywheel's Me.



$$Me = \frac{4\pi^2}{\ell^2} I + M \ (kg \cdot cm \cdot sec^2)$$

$$I = \rho\pi R^4 t/2 \ (kg \cdot sec^2/cm)$$

$$M = \rho\pi R^2 t \quad (kg \cdot sec^2/cm)$$

$I = \rho\pi/2$ $(0.3^4 \times 0.8 + 0.6^4 \times 0.57 + 1.095^4 \times 0.23 + 1.0^4 \times 0.9 + 0.75^4 \times 0.4$
$\times 1.0^4 \times 0.9 + 1.1^4 \times 6.0225 + 1.025^4 \times 7.1775 + 1.15^4 \times 0.23 + 1.095^4$
$\times 1.0 + 0.6^4 \times 0.27) = 0.0002774$

$M = \rho\pi$ $(0.3^2 \times 0.8 + 0.6^2 \times 0.57 + 1.095^2 \times 1.0 + 1.15^2 \times 0.23 + 1.0^2 \times 0.9$
$+ 0.75^2 \times 0.4 + 1.0^2 \times 0.9 + 1.1^2 \times 6.0225 + 1.025^2 \times 7.1775 + 1.15^2$
$\times 0.23 + 1.095^2 \times 1.0 + 0.6^2 \times 0.27) = 0.000501$

$$Me = \frac{4\pi^2}{0.8^2} \times 0.0002774 + 0.000501 = 0.01759$$

3-2. Case's M

	M
i) Linear motion bearing	0.00177
ii) Flange	0.00011
iii) Load bearing flange	0.00160
iv) Tie rod	0.00123
v) Spring's bearing plate	0.00724
vi) Band	0.00225
	0.01420

Consequently, MSA-6, 7's Me = 0.01759 + 0.01420 = 0.0317 kg$\cdot$ sec^2/cm

4. Calculation Method of Starter Torque of Mechanical Snubber.

4-1. Premises of Calculation.

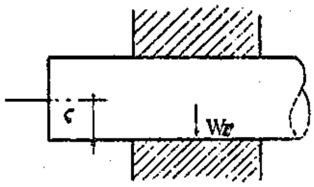
a) Friction coefficient

Assume the friction coefficient of the entire bearing section as $\mu = 0.1$.

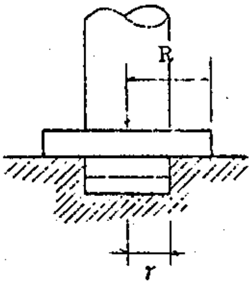
b) The relation between the ball screw thrust (P) and torque (T) is expressed by the following formula:

$$P = \frac{2\pi \cdot T}{\ell(1-\mu)}$$ ℓ : Pitch of ball screw

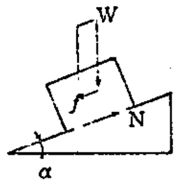
c) Friction at the bearing section



$$T_H = \mu M_T r$$



$$T_V = \frac{3}{2} \mu W \frac{R^3 - r^3}{R^2 - r^2}$$



4-2. Start-up Resistance of Snubber.

4-2-1. MSA-I

i) Flywhell weight 68.3 kg (w)

ii) Ball screw weight 6.00 g

iii) Ball screw pitch angle 1°48' (α)

Ball screw lead 0.5 cm

iv) Weight of test specimen 120 kg (W_T)

$N = \mu W$

$f = W \sin\alpha$

a) Siarting torque in the case of a vertical test:

$N = 0.1 \times 68.3 = 68.3$

$f = 68.3 \times \sin 1° 48' = 2.18$

$$T_V = \frac{3}{2} \times 0.1 \times 74.3 \times \frac{3^3 - 2^3}{3^2 - 2^2} = 18.814 \text{ kg} \cdot \text{cm}$$

The thrust P against Tv is 1, (b), from which,

$$P = \frac{2\pi \cdot Tv}{\ell\,(1\text{-}M)} = \frac{2\pi \times 18.814}{0.5 \times 0.9} = 262.55$$

Consequently, the start-up torque (F) of MSA-1 is:

$Fv = N - f + P = 6.83 - 2.18 + 262.55 = 267.2 = 267$ kg

b) Starting resistance in the case of horizontal test.

$T_H = \mu W_T\, r = 0.1 \times 120 \times 2 = 24$ kg·cm

$N = \mu W_T = 0.1 \times 120 = 12$

$$P = \frac{2\pi \cdot T_H}{\ell\,(1-\mu)} = \frac{2\pi \cdot 24}{0.5 \times 0.9} = 346.9 = 334.9 \text{ kg}$$

$F_H = P + N = 12 + 334.9 = 347$ kg

4-2-2. MSA-2 and 3.

From the same calculation of MSA-1, MSA-2 and 3 are:

MSA-2	Vertical	Fv = 274
	Horizontal	F = 404
MSA-3	Vertical	Fv = 285
	Horizontal	F = 428

4-2-3. MSA-4 and 5

As for MSA-4 and 5, the same consideration as above mentioned is applicable. But starting torque of accelerator is included among starting torque in this case.