TABLE 1 RESULT OF PRELIMINARY TESTS

Test no.		2-1	2-2	2-3	3-1	3-2	3-3	4-1	5-1	5-2
N_0 rpm		510	469	416	511	469	411	463	480	465
N_0/N_n%	aimed	100	90	80	100	90	80	90	94	90
	experi.	102	93.8	83.2	102.2	93.8	82.2	92.6	96.0	93.0
s %	aimed	10	10	10	20	20	20	13.3	16.7	16.9
	experi.	8.13	10.8	11	19.8	19.7	19.3	13.3	17.3	17.5
t_s sec		11.3	9.9	13.0	13.1	13.4	11.0	11.5	11.8	12.2
N_s rpm		505	492	465	500	487	450	485	485	485
t_N sec			21.5	34.7		33.8	38.0	27.3	25.0	30.2
ν_0		1.25	0.94	0.52	0.82	0.52	0.42	0.63	0.73	0.83
ν_s		0.73	0.63	0.78	0.32	0.31	0.73	0.42	0.32	0.42
ν_N			0.63	0.68		0.31	0.42	0.42	0.32	0.42

N = Normal revolutions per minute
N_0 = Speed at which preopening was started
s = Preopening of valve at final stage (%)
N_s = Speed when opening s was reached
t_s = Time required for opening to reach s from 0
t_N = Time required for speed to reach N
ν_0, ν_s = Amplitude of vibrations at opening 0 and s, respectively
ν_N = Amplitude when speed N was reached

be somewhat less reliable than those of the wire strain gage-oscillograph combination, so the values of ν_0, ν_s, ν_N are indicated in ratios to the value corresponding to the normal speed 500 rpm with delivery valve fully closed, the amplitude under these conditions being measured with the oscillograph. The reading of the televibrometer in this instance was 60/1000 mm.

Fig. 8 shows the change with time of the values of valve opening s_t, pump speed N, and amplitude ratio ν_t. The results indicate that when N_0 is too small, the time required for attainment of full speed becomes unduly long, over and above which the

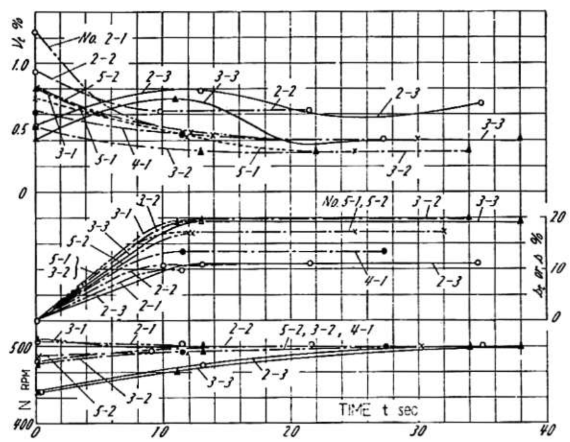

FIG. 8 RATIO ν_t OF VIBRATION AMPLITUDE AT INTERMEDIATE TIME t ON PRELIMINARY TEST
(s_t: valve opening at time t.)

vibrations are magnified by the reverse flow. The optimum values were found to be in the range N_0 = 90–94 per cent and s = 16.7–20 per cent, under which conditions it was ascertained that the vibrations at starting up could be reduced to less than half.

Test Runs on Preopening. The Tōhoku Power followed the preliminary tests with the installation of electrical and mechanical apparatus constructed by Hitachi for manipulating the valves in preparation for the adoption of the preopening method. Test runs were performed from June 25–30, 1954, at which the author again participated, the methods of measurement being the same as before. The maximum vibration amplitude was found to have dropped from the (15–18)/1000 mm recorded during the preliminary tests to 2.5/1000 mm at full speed and with valve fully open, while at starting up, the same had been reduced from the original (55–60)/1000 mm down to (17.5–18)/1000 mm. The cause of this improvement is believed to have been the reinforcement of the by-passage part at the main valve of the water turbine and other pump accessories found to have caused resonance during the previous tests of July, 1953. These measures were analogous to those taken at Grand Coulee, but were carried out according to the original considerations of Mr. K. Honda, Chief of the Design Section of Kameari Works, Hitachi, Ltd., and his colleagues.

The adoption of the preopening method under these conditions proved that the vibration amplitude had been reduced as follows, respectively, to 29 and 44 per cent of previous values:

	Closed starting	Preopened starting
Pump No. 1	17.5	5/1000 mm
Pump No. 2	18	8/1000 mm

Fig. 9 illustrates the changes with time of the speed N, opening of the delivery valve s_t, and vibration amplitude V.

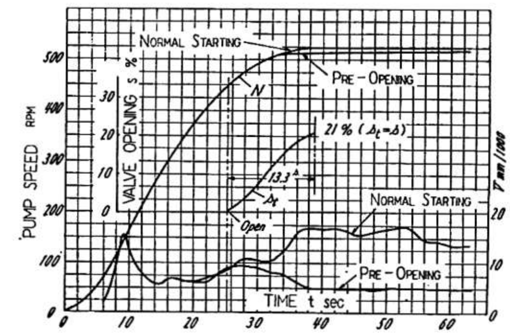

FIG. 9 VIBRATION AMPLITUDE V AT INTERMEDIATE TIME t ON TEST RUN

ACKNOWLEDGMENT

In conclusion, it must be acknowledged with appreciation that valuable assistance was given by Assistant Professors S. Saitō and H. Murai in performing the calculations involved in the work reported in the paper, while the author is greatly indebted to Mr. M. Shirakawa, Vice-President of the Tōhoku Electric Power Company, Ltd., and to Mr. M. Abe, Chief of the Electrical Construction Section of the same company, for their courtesy in permitting the publication of the experimental results.

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Discussion

R. T. KNAPP.[3] This paper is very timely because it focuses attention on a type of problem that is relatively new in the centrifugal-pump field and also offers a new method of attack for its solution.

[3] Professor of Hydraulic Engineering, California Institute of Technology, Pasadena, Calif.

Hydraulic turbines and pumps are close relations and have many things in common, particularly in their hydrodynamic characteristics. In fact, a good turbine can be operated as a pump with high efficiency and a good pump is also an excellent turbine. However, there is one striking difference in their fields of application. Hydraulic turbines are usually installed in large units. Thus a turbine whose output is measured in hundreds of horsepower is considered small and is rarely seen. On the other hand, the vast majority of centrifugal pumps are driven by motors of under 100 hp, and pumps requiring thousands of horsepower to operate them are very uncommon and have only been developed in relatively recent times. Thus pump manufacturers and users have had relatively little experience with troubles peculiar to large installations.

There is another characteristic difference between the two types of machines. Turbines normally are started and stopped under no-load conditions. Usually, the basic nature of the installation is such that they cannot be made to operate as a pump nor can conditions occur under which they will rotate in the reverse direction. Pumps, on the other hand, either start under load or against a closed discharge valve, which in itself imposes extremely complicated hydraulic conditions. Also, in many installations, they may be made to run in the reverse direction as turbines, at least for brief periods. These facts create comparatively little difficulty with small units where starting times are inherently very short and the machines themselves are compact and rigid. However, troubles like the ones the author discusses are beginning to arise in the transient operation of large units.

The author's use of the complete characteristic diagram to explore and calculate the characteristics of special methods of starting is most interesting. In the past the use of the characteristic diagram has been restricted largely to the calculation of water hammer produced during normal or abnormal shutdown of large pumps.[4] This new use should serve to emphasize the wide possibilities inherent in this type of diagram.

There is still one weak point in the present situation. This weak point is not with the author's method, but rather with deficiencies in laboratory methods of testing model pumps. Thus in Figs. 3 and 4 of the paper various possible starting procedures are plotted, the evaluation of the relative desirability of which requires field testing. The only reason that field tests are necessary is that the normal laboratory tests do not yield the necessary information about the amount of vibration-producing pressure fluctuations along the different paths. It seems quite certain that the pressure fluctuations occur in the model pump as well as in the prototype but the relative vibration is usually very much smaller because in the past the model pump has never been a structural model but only a hydrodynamic one. Thus it seems quite clear that there is a need to improve model-testing techniques so that troubles of this kind can be anticipated and avoided in the field installation.

Author's Closure

The author wishes to express his appreciation of the understanding discussion given by Prof. Robert T. Knapp.

His suggestion that model testing techniques should be improved from the viewpoint of a structural model is a valuable one.

Though there are some difficulties in seeking the principles of similarity in elastic vibration concerning the model and its prototype, the author believes the model techniques of pumps ought to develop in the future along the line of Prof. Knapp's suggestion.

[4] "Complete Characteristics of Centrifugal Pumps and Their Use in the Prediction of Transient Behavior," by R. T. Knapp, Trans. ASME, vol. 59, 1937, pp. 683–689.

"Waterhammer Analysis," by John Parmakian, Prentice-Hall Civil Engineering and Engineering Mechanics Series, New York, N. Y., 1955; see chapters 11 and 12.