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船體振動の流體力學的制振法

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Abstract.

A Hydrodynamic Method of Damping the Vibrations of a Ship.

By Katsutada Sezawa, *Kogakuhakusi, Member*, and Wataru Watanabe.

The authors suggest a hydrodynamic method for damping the vibration of a ship. Submerged blades of moderate elastic resistance and inertia mass that are fitted to a ship's hull, are possible to fairly damp the vibrations of that ship for any frequency of the forced vibration, particularly in certain speed range of the ship. The nature of the problem is ascertained mathematically and confirmed through model experiments. It is shown, in some example, that the maximum amplitude of the ship (among different vibrational frequencies) with a damper, is merely 1/9 that of the case of no damper. The applicability of the result of the investigation to the actual case is also discussed.

The details of the problem as well as its theoretical and experimental treatments are shown in the Appendix towards the end of the paper.

前回の論文では、水の粘性、表面波、船體の内部的力のために船體の强制振動が減衰され、その振幅が一定以上になり得ぬ事を理論的に說明して置いた。然し夫等の力では減衰性に限界のあるのは言ふまでもない譯である。この論文では船體に特別の制振器を取付け、流體力學的作用で其の振動を制振する方法を擧げようと思ふ。その方法と云ふのは餘り難かしいのではなく、元良式安定器の如き舵面板を船から水中へ突き出し、之に適當の彈性力と質量とを持たせればよいのである。之は Inglis が出した制振器と稍似てをるけれども、この場合には制振器と船との間には何等の摩擦力をも置かぬ事、及び制振器の面が流體力學的の復原力と減衰力とを受けると云ふ差異がある。從つて此の場合には船體が或範圍の前進力を持たねばならぬ事が必要である。又、制振器の一定の質量を必要とするのは Inglis の場合と餘り變りがないのである。然し此の制振器があれば凡ゆる强制振動週期でも其の效力を發揮し力學的制振器の特徴を完全に備へて居るものである。

この研究を試みる爲には試驗水槽が必要な譯であるけれども、我々には其の便宜がなく、其の代りに已むを得ず小さな風洞を用ひた。然し之でも相當の程度まで性質がわかるのみでなく、實驗も容易に試みる事が出來るのである。

試驗に船の代表として用ひた模型は兩端自由の梁（極く弱い**ばね**で重さを支へる）であつて、其の質量は主として 3 箇所に集中してある。これでも船體振動の性質は可なりわかるものである。Fig. 1,

Fig. 2 の如く、其の上方に舵面板があり、之を船體を代表する梁に固定したり、又は夫れに對して相對的に振動したりするやうにするのである。固定した場合と振動した場合を夫々 A, B とする。A, B の夫々の場合につき種々の風速の風を與へて見た。Table II の如く、風速が高くなるに從つて夫々 A1, A2, A3, A4, A5; B1, B2, B3, B5 に相當する實驗を試みたのである。

强制力としては Fig. 1, Fig. 2 の如く室内の床の上にある直流電磁石 G と梁に附着せる可動線輪 C とがある。C には毎秒 1 回から 10 回（電流の方は 100 回まで出來る）までの交流を通じ、その電壓と電流の强さとは一定に保つのである。この様にして種々の振動數で實驗を試みた結果の一二を Figs. 3〜6 に示して置いた。之から共振曲線を作ると A, B の場合につき夫々 Fig. 7, Fig. 8 の如き結果となるのである。

A の場合でも風速が高くなるに從つて制振力が働く事は働くけれども、B の場合程ではない。又風速零の場合でも、B の方が A の方よりも多少の效果がある。然し何と云つても、B の方の風速の高い所では、凡ゆる振動數で其の振幅が少くなり、例へば B5 の場合、即ち風速が 24.5 米/秒、換言すれば Reynolds 數が $24.4.10^4$ の場合には共振に當たる最大振幅が共振でない場合の最小振幅の 4 倍位にしかならぬ事がわかつたのである。この最大振幅は制振器固定の場合に比較して著しく小さく、制振器固定風速零の場合の 1/9 位しかなく、制振器固定で同風速の夫れに比較しても 1/3 位しかないのである。又、A の場合には共振が 1 つしかなかつたものが B の場合には 2 つの共振のある事は其の性質から明かな所である。この性質がある事と、制振器が大きな振動の代理を爲す事がある故に、凡ゆる週期に於て振幅が大きくならぬものである。

この實驗を試みるに先立つて、その自己振動の性質を調べてあつたから、夫等の結果を用ひて强制振動の場合の性質を全く數學的に豫知する事が出來た。A1, A5; B1, B5 の場合につき正確なる數理を當嵌めて出した結果は夫々 Fig. 9, Fig. 10 に濃い破線、濃い實線を以て示してある。制振器の振幅も細い破線、細い實線で示してあるが、之に依つて見ると制振器の振幅は夫れ程大きくならなくてもよい事がわかる。B5 の場合が其の他の場合に比して著しい效力のあるのは此の數理の結果からも明かであるが、是等の數理的結果が實驗の結果と可なり能く一致せる性質のあるのは注意すべきである。殊に Fig. 10 の濃い實線と Fig. 8 の濃い實線とを比較して見ると、計算及實驗と云ふ別の方法から出した結果が殆ど重なり合ふ位能く一致して居るのである。

計算の結果を實物へ持つて行く事には異論があらう筈もない。模型實驗の結果を實物に當嵌めるには種々の相似の法則が必要である。然し之は附錄の終りに委しく述べてあるやうに、適當の注意次第で可能にし得る事がわかつた。

實際問題で如何なる制振器を取付ければ良いかは容易にわかりさうもないが、bilge keel の一部分に撓み性を與へ且つ之を適當に重くするのも凡ゆる場合に無意味とも考へられない。

この研究では便宜上航空研究所の風洞を用ひた。十二三年位前には風洞の代りに水槽を用ひた事も

あつたのである。然し飛行機の胴體の振動も船體の振動と多少似た所があるから、其の場合の制振法をも此の風洞によつて研究する望を持つて居るのである。玆に和田航空研究所長其の他の方に謝意を捧げる。

終りに此の種の研究を始めるに際し御激勵を頂いた平賀先生や研究中に種々の御援助を受けた近藤記念海事財團及び其の役員の方々に厚く御禮を申述べる。

昭和 12 年 8 月 15 日

Appendix. The Details of the Problem of A Hydrodynamic Method of Damping the Vibrations of a Ship.

1. *Introduction.*

In the previous paper[1] we studied the probable causes of the natural damping of the vibrations of a ship, the conclusion at which we arrived being such that although the generation of the surface waves as well as the effective structural damping are likely to participate greatly in the natural damping of the same vibrations, it is still impossible for these causes to minimize the ship's vibration to any satisfactory extent.

In the present paper we shall give a suggestion to a hydrodynamic method of minimizing the vibration. The damper to be used is of such blade type as immersed in water like Motora's stabilizer[2] but connected elastically to the ship's hull. Since the surface of the blade is to be parallel to the ship's length, the additional resistance of the ship is very small. The mass of the blade, nevertheless, is not too small for the reason that the same blade acts as a dynamic damper like Inglis's case.[3]

In the present method there is no damping resistance imposed between the damper mass and the ship's structure but the vibration of the same mass is subjected to a high resisting force due to the dynamic action of the water, which results from the oscillatory motion of the damper mass relative to the ship's movement.

2. *General Arrangements and Preliminary Experiments.*

As a simplest model of a ship in vibration we took a free-free beam of nearly

(1) K. Sezawa and W. Watanabe, "Damping Forces in Vibrations of a Ship," *Journ. Soc. Nav. Arch., Japan*, 59 (1936), 99-120.

(2) S. Motora, *Journ. Soc. Nav. Arch., Japan*, 32 (1923), 75-84; 36 (1925), 109-117.

(3) C. E. Inglis, *T. I. N. A.*, 75 (1933), 252-267

uniform cross section and length $2l=162{\cdot}4$ cm, the concentrated masses $M_1=647{\cdot}5$ gr, $M_2=760$ gr, $M_2'=760$ gr being placed at three points in the beam, namely at its middle point as well as at its both free ends. Even should the masses be concentrated at such special points, the vibrational nature of the problem would not much differ from that of the actual condition. From the vibration experiment of a cantilever beam with the formula $M_2l^3p^2/EI=3$ together, the stiffness of the beam EI was found to be $3{\cdot}99.10^{10}$ C. G. S.

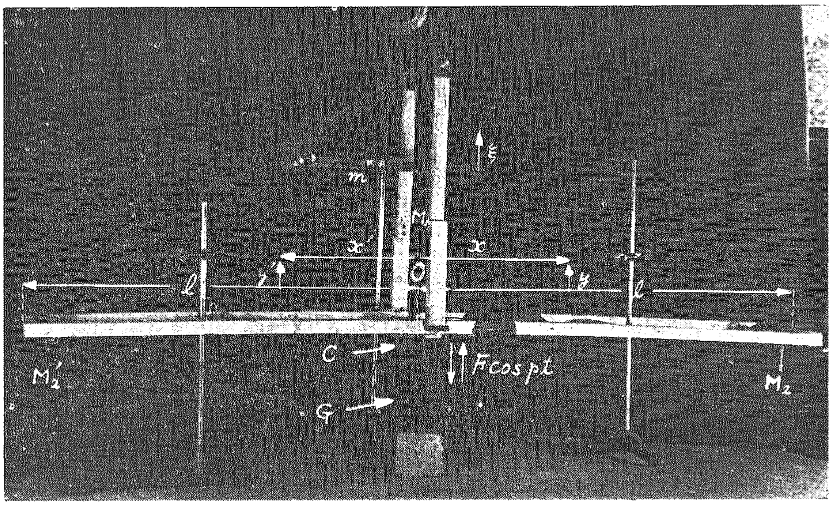

Fig. 1.

The effective mass of the blade damper was $m=135{\cdot}4$ gr, the ratio of $m/(M_1+M_2+M_2')$ being thus about 1/15.

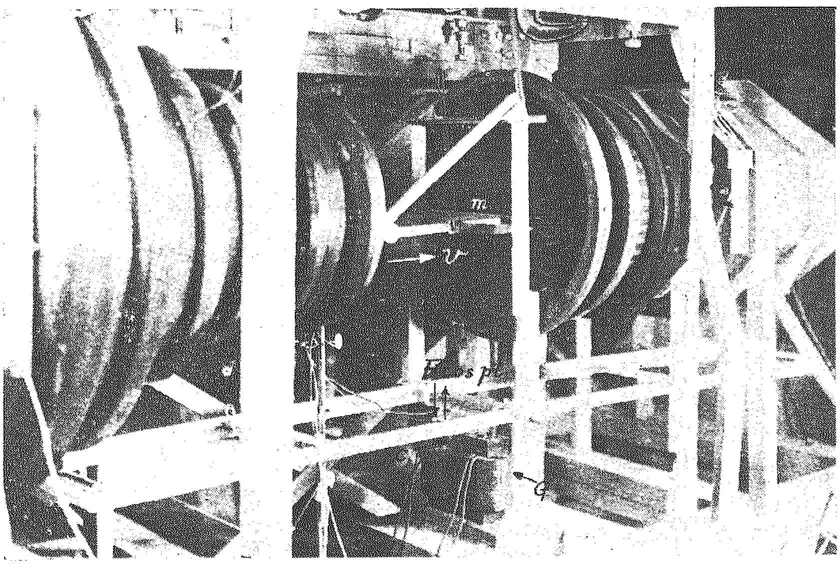

Fig. 2.

Since it was a difficult matter for us to use an experimental tank, a wind tunnel[4] of 30 cm air nozzle was temporarily availed of in our case, the tunnel under consideration and the model being shown in Fig. 2. The restitutive and damping forces on the damper at the fluid speed v are generally

$$c=c_0+\frac{\partial C_L}{\partial \xi}\frac{\rho A}{2}v^2, \quad\ldots\ldots\ldots\ldots\ldots\ldots(1)$$

$$\mu=\mu_0+\frac{\partial C_L}{\partial \xi}\frac{\rho AL}{2}v \quad\ldots\ldots\ldots\ldots\ldots\ldots(2)$$

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(4) A wind tunnel in the Aeronautical Research Institute, Tokyo Imperial University. We used the tunnel for the present experiments in the hope that a similar kind of experiments for devising the vibration damper of a fuselage would be made forthwith in the same tunnel.

respectively, where c_0, μ_0 are the restitutive force (due to spring) and damping force at zero speed; ρ, A, L are fluid density, blade area, and length between the blade spring (in our case) and its hinge. $C_L(\rho A/2)v^2$, which is the effective transverse force on the blade end at the displacement ξ, was observed in non-vibrational state, with the final result that

$$\frac{\partial C_L}{\partial \xi}\frac{\rho A}{2}=0{\cdot}00556 \text{ gr. mass/cm}^2, \quad\quad (3)$$

in the case of air stream, v being taken in cm/sec. Since L was 32 cm, (2) and (3) gave

$$\mu=\mu_0+0{\cdot}178\, v \text{ gr. mass. sec/cm}^2; \quad\quad (4)$$

together with

$$c=9{\cdot}8.10^4+0{\cdot}00556\, v^2 \text{ gr. mass/cm}^2. \quad\quad (5)$$

On the other hand, the free vibration test of the damper only, using the formula

$$\xi=Ae^{-\frac{\mu}{2m}t}\cos\left(\sqrt{\frac{c}{m}-\left(\frac{\mu}{2m}\right)^2}\,t+\varepsilon\right), \quad\quad (6)$$

gave rise to the result shown in Table I, from which we obtained $\mu_0/2\,m=0{\cdot}289$ (C. G. S.) and together with (4) we got $m=135{\cdot}4$ gr, the value of μ thus being, for example, $\mu=81{\cdot}0$ (C. G. S.) for $v=0$, and $\mu=516$ (C. G. S.) for $v=2450$ cm/s.

Table I.

v (m/s)	0	5·0	10·0	15	19·9	24·5
$\mu/2\,m$	0·289	0·740	1·082	1·175	1·803	1·904

3. *Forced Vibration Experiments and their Results.*

A moving coil C affixed to the middle point of the beam was faced to an electromagnet G on the room floor. While a direct current of 100 volt and 0·3 amp. was transmitted through the coil in the magnet G, alternating currents of various cycles ranging from 1/sec to 10/sec that were generated from a specially designed motor generator, were made to pass through the moving coil C, the voltage (10 volts) and amperes (0·25 amp.) of the alternating currents being kept almost constant. The reason why we put the moving coil at the middle point of the beam is such that in the case of a given exciting force the appropriate position of the same force is the loop of the vibrational mode of the beam.

We conducted two series of experimenting cases, that is, the series in which the damper is fixed to the main beam and the series in which the damper is free to oscillate relative to the main beam, the whole cases being shown in Table II. (the chord of the damper being taken as length dimension in Reynolds's number).

Table II. Reynolds's number R (and air velocity v m/s).

Damper condition	Case		1	2	3	4	5
Fixed to the beam	A	$R/10^4=$	0	4·74	9·98	15·4	23·3
		v (m/s)=	0	4·75	10·0	15·5	23·4
Oscillate relative to the beam	B	$R/10^4=$	0	4·81	9·98	—	24·4
		v (m/s)=	0	4·82	10·0	—	24·5

In the case of forced vibrations, the displacements of the middle point of the beam, namely the point at which the exciting force was applied, were recorded without magnification (in the original records) on smoked papers of a drum. The data recorded on the smoked papers were generally of the type shown in Figs. 3, 4, 5, 6 which exactly correspond to Cases A1, A5, B1, B5 respectively, N being the vibrational frequency per sec of the exciting force. (As will be shown later, Case A1 is an almost undamped condition and Case B5 an extremely damped condition.)

All the amplitudes in every case were read off from records of the type shown in Figs. 3–6 and plotted in Figs. 7, 8, which, in the nature of things, represent the resonance curves. In these figures Cases A1, B1 were specially indicated by thick broken lines and Cases A5, B5 by thick full lines, whereas Cases A2, A3, A4, B2, B3 were shown by thin full lines.

Since in Cases A's the damper is in a fixed condition, there is only one peak in every resonance curve. In Cases B's, on the other hand, owing to the possible relative movement of the damper, two peaks exist in the respective resonance curves. As to the resonance conditions of higher frequencies, the problem lies outside the scope of our experiments.

It will be seen that when the damper is in a fixed condition and the wind velocity is not too high, the peaks in the resonance conditions are generally pronounced. Even should the damper be in a fixed condition, the vibration amplitudes of the beam would diminish to a certain extent with increase in fluid speed; the maximum amplitude in the condition A5, namely $R=23{\cdot}3.10^4$ ($v=23{\cdot}4$ m/s) was about 1/4 that of the condition A1, namely $R=0$ ($v=0$); see Fig. 7.

In the active condition of the damper, that is, the condition that the damper was subjected to its spring and hydrodynamic forces and rendered vibratory, the

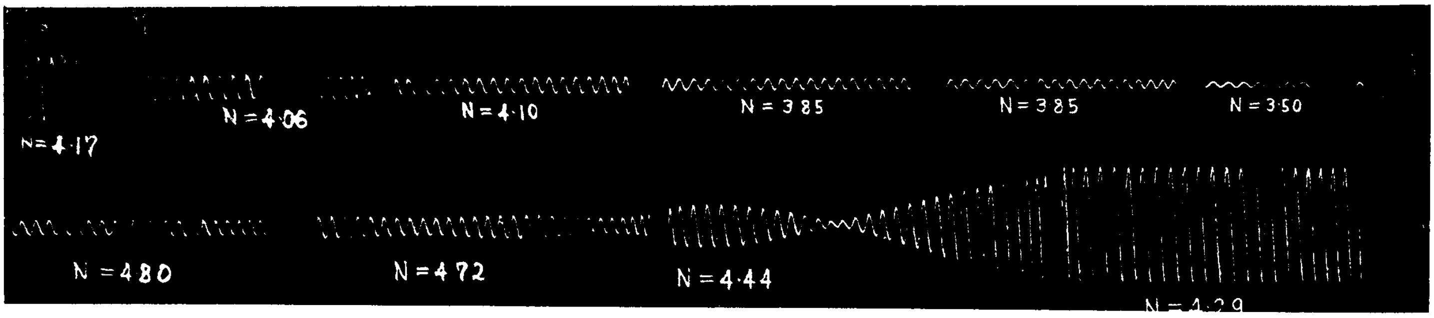

Fig. 3. Case A1. Damper in a Fixed Condition ; $v=0$. (Scale contraction = 1/3)

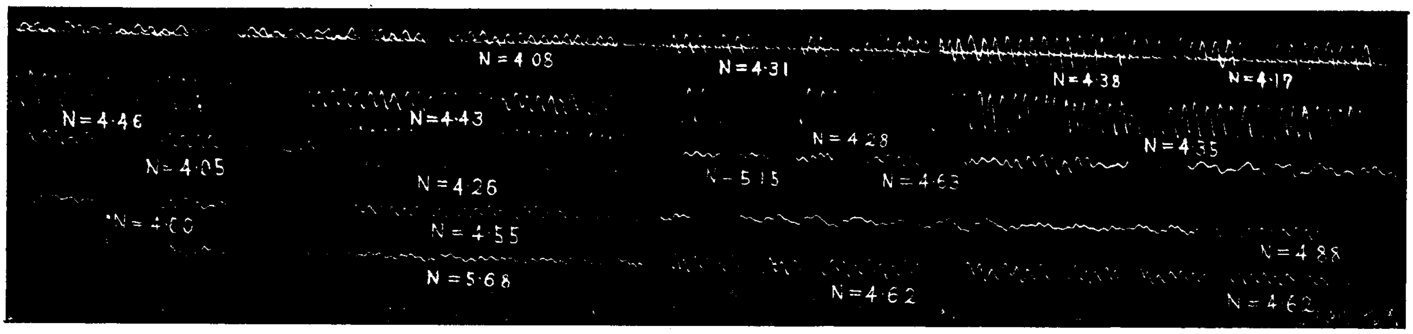

Fig. 4. Case A5. Damper in a Fixed Condition ; $v=23.4$ m/s. (Scale contraction = 1/3)

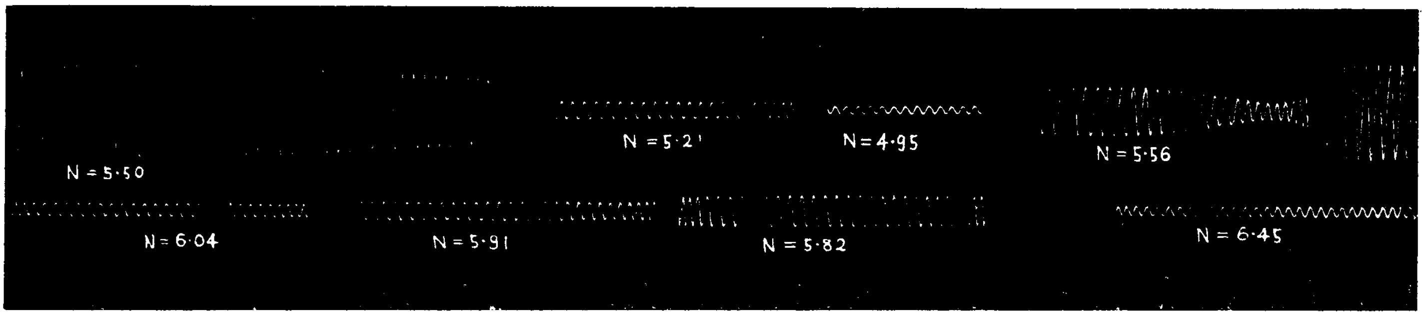

Fig. 5. Case B1. Damper in an Active Condition ; $v=0$. (Scale contraction = 1/3)

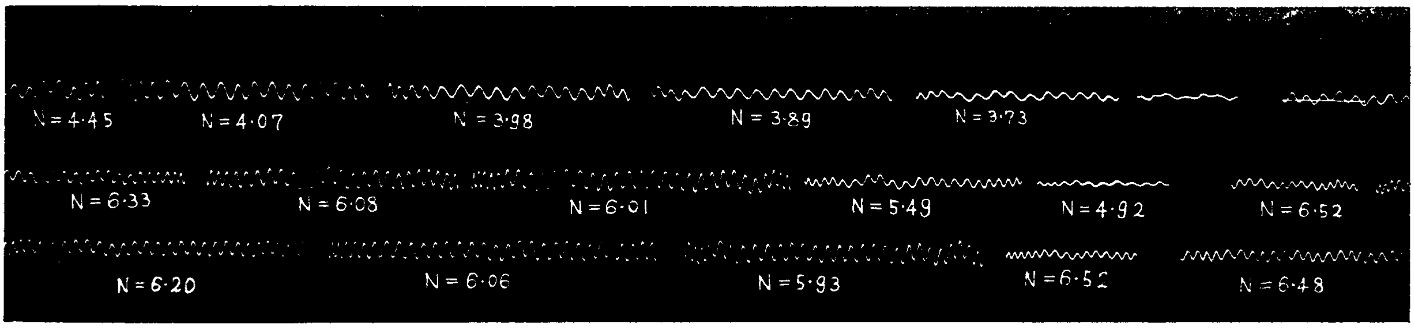

Fig. 6. Case B5. Damper in an Active Condition ; $v=24.5$ m/s. (Scale contraction = 1/3)

vibration amplitudes never exceeded a certain critical. It appears that in the best condition of the damper the ordinates of both the peaks under consideration are of nearly equal heights. In Case B5, namely $R=24\cdot4.10^4$ ($v=24\cdot5$ m/s), the amplitudes of both resonance conditions were 1/2 cm, which was merely 1/6 that of Case B1 or even 1/9 that of Case A1; see Fig. 8. It will also be seen that notwithstanding that every condition excepting the one corresponding to the relative movement of the damper, was the same for B5 and A5, the maximum amplitude in Case B5 was only 1/3 that in Case A5.

It seems that such an efficient condition of

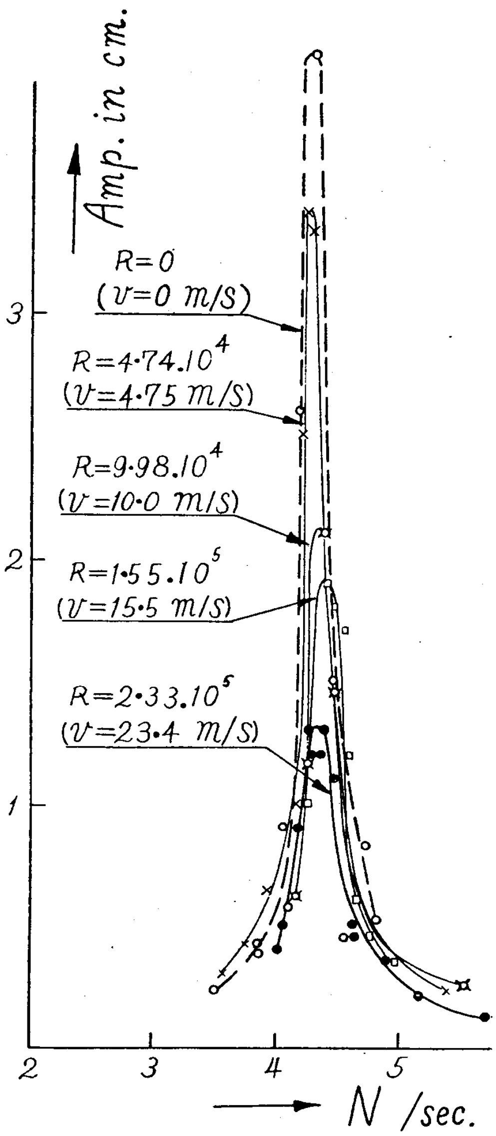

Fig. 7. Resonance Curves in Cases A. Experimental.

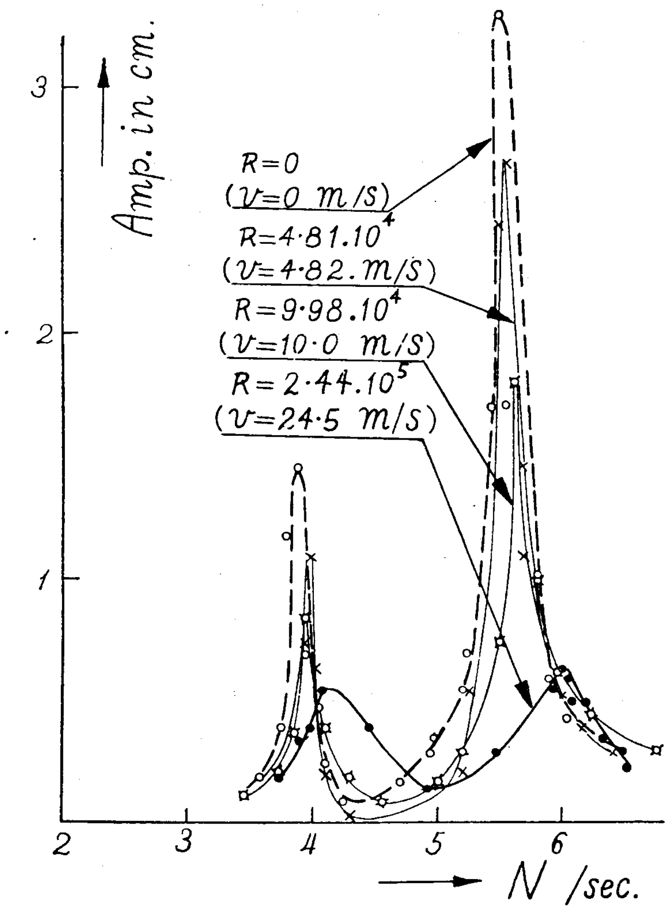

Fig. 8. Resonance Curves in Cases B. Experimental.

the damper as in Case B5 is possible to exist provided the natural period of the damper alone is nearly equal to that of the beam and also the hydrodynamic damping force assumes a moderate value. Although no vibration experiment was made for a case of any higher speed, it appears improbable to expect a much more damped condition of the beam even at such a higher speed. One of the specialities of the damper of the present type lies in the condition that the vibration amplitudes of the beam never exceed a certain extent for any vibrational frequency including the resonance conditions. Thus, in Case B5 the ratio of the maximum amplitude (under resonance) to the minimum amplitude (in non-resonance condition) is less than 4.

Although the state of the second or higher resonance may be another problem to be discussed, since other kinds of resistances,[5] say, structural damping, wave-making resistance, water viscosity, and etc., then take the large part in the vibration damping, there is no need for further comment.

4. *Mathematical Theory.*

Let the length of the beam, namely the ship, be $2l$ and take the origin of x at its middle point. If the mass of the beam be concentrated at certain points, the equations of the deflectional vibration of that beam assumes the form

$$\partial^4 y/\partial x^4=0, \qquad \ldots\ldots (6)$$

the solution of which is

$$y=(A+Bx+Cx^2+Dx^3)e^{ipt}. \qquad \ldots\ldots (7)$$

The boundary conditions are such that

$$x=l_1\,;\quad \frac{\partial^2 y}{\partial x^2}=0,\quad -EI\frac{\partial^3 y}{\partial x^3}+M_2\frac{\partial^2 y}{\partial t^2}=0, \qquad \ldots\ldots (8),\ (9)$$

$$x=0\,;\quad \frac{\partial y}{\partial x}=0,\quad m\frac{\partial^2}{\partial t^2}(y+\xi)+c\xi+\mu\frac{\partial}{\partial t}(y+\xi)=0, \qquad \ldots\ldots (10),\ (11)$$

$$2EI\frac{\partial^3 y}{\partial x^3}+M_1\frac{\partial^2 y}{\partial t^2}-c\xi-\mu\frac{\partial}{\partial t}(y+\xi)=Fe^{ipt}, \qquad \ldots\ldots (12)$$

where M_1, M_2, M_2 are concentrated masses at the middle point and ends of the beam, EI being the stiffness of the same beam, whereas m, c, μ, ξ are mass, stiffness, damping resistance, and the deflection (relative to the beam) of the damper. Fe^{ipt} is the periodic exciting force at the middle point of the beam.

Substituting (7) in (8)～(12) and taking real parts, we finally get

$$y=F\sqrt{\frac{X^2+Y^2}{\Gamma_1^2+\Gamma_2^2}}\cos\left(pt+\tan^{-1}\frac{\Gamma_2}{\Gamma_1}-\tan^{-1}\frac{Y}{X}\right), \qquad \ldots\ldots (13)$$

$$\xi=F\sqrt{\frac{S_1^2+S_2^2}{\Gamma_1^2+\Gamma_2^2}}\cos\left(pt+\tan^{-1}\frac{\Gamma_2}{\Gamma_1}-\tan^{-1}\frac{Y}{X}\right), \qquad \ldots\ldots (14)$$

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[5] K. Sezawa and W. Watanabe, *loc. cit.* [1].

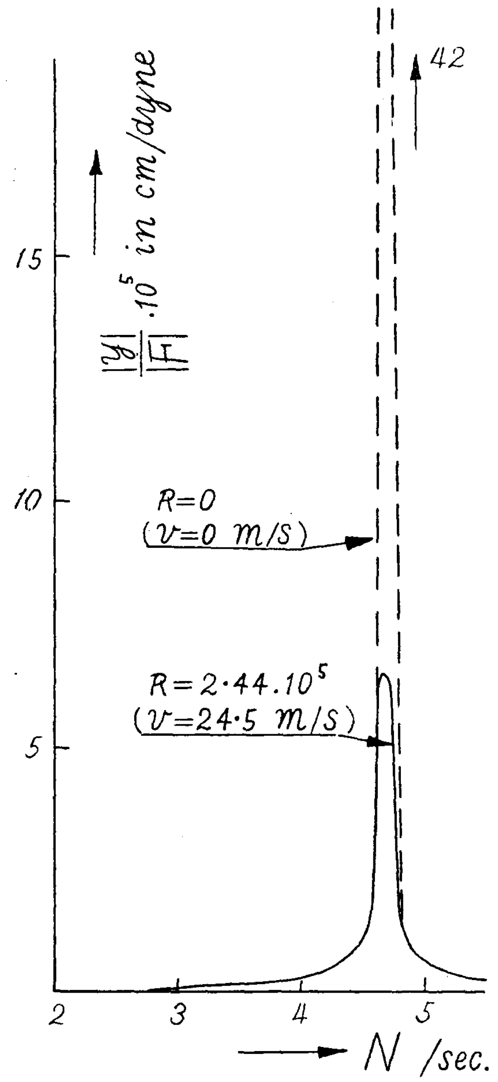

Fig. 9. Resonance Curves in Cases A Mathematical.

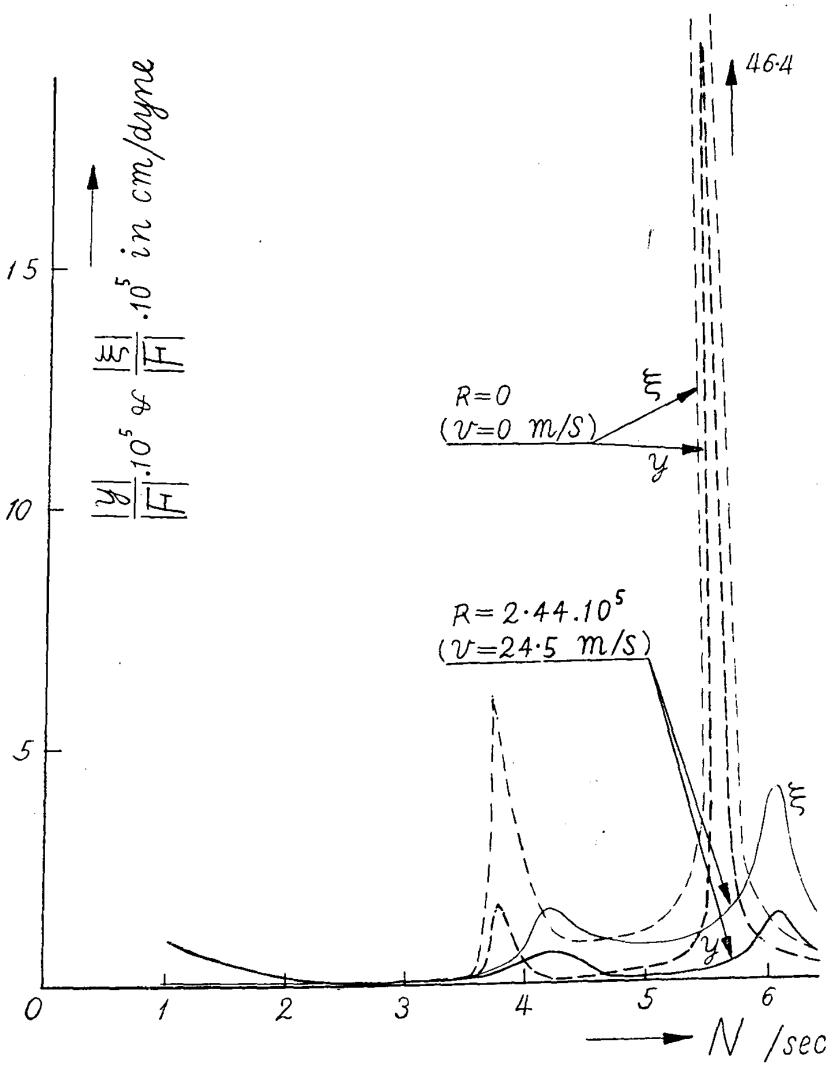

Fig. 10. Resonance Curves in Cases B. Mathematical. Thick lines represent $|y|/|F|$ and thin lines $|\xi|/|F|$.

where

$$\left.\begin{aligned} X&=P_1+Q_1x^2+R_1x^3,\\ Y&=P_2+Q_2x^2+R_2x^3, \end{aligned}\right\} \cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots (15)$$

$$\left.\begin{aligned} \Gamma_1&=-6EI(M_1+2M_2)mp^4+2M_1M_2ml^3p^6\\ &\quad+\{6EI(M_1+2M_2+m)p^2-2(M_1+m)M_2l^3p^4\}c,\\ \Gamma_2&=p\mu\{-6EI(M_1+2M_2)p^2+2M_1M_2l^3p^4\\ &\quad+(6EI-2M_2l^3p^2)c\}, \end{aligned}\right\} \cdots\cdots\cdots\cdots (16)$$

$$\left.\begin{aligned} &P_1=(mp^2-c)(6EI-2M_2l^3p^2),\quad P_2=p\mu(6EI-2M_2l^3p^2),\\ &Q_1=3M_2p^2l(mp^2-c),\quad Q_2=3p\mu M_2p^2l,\\ &R_1=-M_2p^2(mp^2-c),\quad R_2=-p\mu M_2p^2, \end{aligned}\right\} \cdots\cdots (17)$$

$$S_1 = -mp^2(6EI - 2Ml^3p^2), \quad S_2 = -p\mu(6EI - 2M_2l^3p^2),$$

the exciting force being

$$F\cos pt \qquad \text{.......(18)}$$

Using the equations just obtained for such special cases as correspond to A1, A5, B1, B5 in the experiments whose numerical data were given in Section 2, we obtained the resonance curves as shown in Figs. 9, 10. $|y|/|F|$ (in cm/dyne), namely the maximum amplitude of the beam per unit exciting force, for Cases A1, A5, are shown by full and broken lines respectively in Fig. 9, whereas $|y|/|F|$ and $|\xi|/|F|$ (in cm/dyne) in Case B1 as well as Case B5 are shown by thick and thin full lines as well as thick and thin broken lines respectively in Fig. 10.

Comparing the thick full line in Fig. 10 with that in Fig. 8, it will be seen that both the results, mathematical and experimental, fairly agree in every vibrational frequency. In Cases A1 and B1, in which the vibrations are not well damped, the amplitudes obtained mathematically are enormously large. Since there are some damping resistances still in the models arising from other causes, perfect agreement would be rather unreasonable.

Finally, it appears from Fig. 10 that the amplitudes of the damper itself are not too large even in its active condition, its application to the actual case being therefore rather possible.

5. *Application of the Mathematical and Experimental Results to a Full-scale Ship; and Concluding Remarks.*

It is obvious that any mathematical result should apply to an actual case provided the values of all the numerical constants for both sides are the same. The question may arise as to whether or not the results of model experiments may be used in an actual case. A well established law of similitude would eliminate any difficulty in the problem.

From the mathematical discussion already shown and also from the hydrodynamical condition, the parameters for defining the results are such that

$$\text{(i)}\ \frac{Ml^3p^2}{EI}, \quad \text{(ii)}\ \frac{m}{\sum M}, \quad \text{(iii)}\ \frac{EI}{Ml^3}\cdot\frac{m}{c}, \quad \text{(iv)}\ \frac{\mu p}{c}, \quad \text{(v)}\ R,$$

R being the Reynolds's number, shall be the same for the model and the full-scale ship. The condition in (i) is, at all events, satisfied if we discuss the problem with reference to the resonance condition. Although we obtained the ratio of $m/\sum M$ as 1/15 in the present case, the ratio under consideration would be much more reduced were other conditions changed suitably. Regarding to the condition (iii), $\alpha EI/Ml^3$ (α being a number) should be nearly equal to c/m, namely, the vibrational period of the damper and that of the ship are nearly equal. It would also be rendered possible to get the condition (iv), namely that $\mu p/c$ is of an assigned value, provided the area of the damper were properly selected. The treatment of the problem in obedience to (v) the Reynolds's number, is somewhat difficult. Since in model experiments the Reynolds's number is liable to be rather too small, there may remain some doubt on the immediate application of such results of the model experiments as ours to the actual case. In the vibration problem, nevertheless, the important part is $\partial C_L/\partial\xi$, which may readily be adjusted by merely changing the stream line form of the damper; the Reynolds's number itself has no place in any important part of the problem.

The word "a free-free beam" or simply "a beam" used in experimental as well as mathematical parts should now be meant by a ship floating on water in an idealized sense. While, furthermore, we studied such a particular case that a blade damper is placed at a certain level above the beam, the damper which might be actually designed, on the other hand, should be directed downwards or obliquely into the water like Motora's stabilizer.

In the present experiments, for convenience, we used a 30 cm wind tunnel in the Aeronautical Research Institute. We have, nevertheless, the intention to extend the present investigation to dampening of the vibration of an aeroplane fuselage whose vibrational nature resembles that of the ship in some way.

In conclusion we wish to express our thanks to Professor Hiraga for his encouragement and also to the Council of Kondô Memorial Marine Foundation for aid received for the present investigation.

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討　　論

○座長(平賀　讓君) 何方か御質問の方は御座いませんか。

○井口常雄君 風洞に用ひた model の圖が分らないのですが　物が出て居りますか。

○妹澤克惟君 それには Fig. 1 を御覽下さい。

○井口常雄君　すると、それを船としますと何か水の中に出す譯ですね。

○妹澤克惟君　此の儘實船と比べるのは大膽すぎますが、兎に角、水の中に出す譯です。

○井口常雄君　それが振動するのですか。

○妹澤克惟君　それは船が振動すると自然に振動して其の振動は船のものより大きいのです。然し大して大きくなる譯ではありません。

○井口常雄君　weight が相當要るのではないですか。又そのものゝ進行に對する抵抗が出てくるんではないですか。

○妹澤克惟君　其の抵抗が船全體の resistance を增加し、其の振動を止める爲の HP が夫れだけ餘計にかゝる譯です。夫れが餘り大きくなると困ります。尤も damper の振動性のために抵抗が增すとすれば、damper のない普通の狀態の船でも振動で抵抗がふえてると申せます。

○井口常雄君　量的の關係が分りませんか。重さや HP を增さずに旨くやる事が出來るとよいのですが。

○妹澤克惟君　mass が小さくては困るのです。船の rudder や bilge keel, horizontal rudder 等が apply 出來たらよいと思ひます。forced oscillation が何れの振動周期に對しても小さく damp するのです。船の振動は irregular のやうに云はれてゐるかも知れませんが、engine から起るものですから、そんなに irregular でないでせう。そのやうな場合の振動では mass が小さくても有效になります。anti-rolling tank, Inglis の vibration damper の場合も同樣です。又、走る時の damping の係數は大きくなりますが、飛行機の翼や rudder 等でも hydrodynamic damping force が當然大きくなります。

○廣田守道君　飛行機にも適用されるのですか。

○妹澤克惟君　之は船としてやつたものですが、航空研究所の風洞でやつたものですから飛行機の方面にも extend してもよいでせう。胴體には應用してよいでせうが、翼には利用出來かねると思ひます。

○松山武秀君　船體と damper との connection は現今貴方がやつてゐる樣な工作方法で納まりますか。

○妹澤克惟君　之は damper の附根を pivot して、先の方に helical spring でも附けたら簡單でせう。寧ろ元良さんの樣に shaft を出さないで independent に出したらよいでせう。

○座長(平賀　讓君)　これは船全體の設計と云ふ點から見ますると mass が negligible でなければ態々大きな mass をつけてまで振動を減らす必要はないのではないでせうか。それほど振動で困つてゐる譯ではないと思ひますが。

○妹澤克惟君　御尤もの考であります。而も之は船でなくてもさうでせう。これは damper だけ

の mass としてだけではなく、船の形を今とすつかり變へて damper の中に cargo を積む様にでもせねばならぬでせう。今の魚の様な形を思ひ切り違ふものにして、damper をうまく搖らせる様にしたらよいと思ひます。何にしても此の研究は船に關する物の性質を草分け的に考へただけであつて直ぐ役に立つ研究の積りでは御座いません。

○井口常雄君 その damper が船より餘計に搖れるのですか。

○妹澤克惟君 さうです。それは damper がきく狀態では damper も搖れなくなります。しかし船よりは幾らか多くうまく行けばまあ、2〜3 倍位も搖れますが、それは船自體の殆ど零の振幅に比して 2〜3 倍といふ意味です。

○井口常雄君 水中に出てゐる抵抗が少くとも、振動すると進行に對する resistance が大きくなるのではありませんか。

○妹澤克惟君 その進行の resistance は船全體の resistance の增加となり engine の HP の增加となつて來る譯です。之は前述の如く普通の狀態で船自身が振動すれば HP が增加してるかといふ議論と同じになります。

○座長(平賀　讓君) damper によつて船に新らしく vibration が出てると云ふ懸念はないでせうか。

○妹澤克惟君 それは力學的制振器が有效な狀態で取付けてあれば實際問題として絕對に大丈夫です。

○座長(平賀　讓君) 都合よくゆらせるつもりでも damper 自身にも新しい vibration が出て來ませんか。

○妹澤克惟君 Fig. 8 を御覽下さい。resonance にあたるのは、abscissa の 4〜5 位の處で Fig. 7 の高くなつてゐる處が夫れで御座います。damper を附けた爲に新たに vibration を起さぬかと云ふ事に對しては Fig. 8 の abscissa 4 及 6 の處の ordinates が少しく高くなつてゐる。之れは附けない時には Fig. 7 の相當低くなつてゐる處 (resonance でない所) であつて夫れだけ犧牲として增したもの、つまり 4, 6 の處が犧牲になり damper のない時の 2 倍の vibration になつてをります。しかし之は實際問題として negligible (Fig. 7 の peak に比し) で御座います。凡ゆる vibration の周期に於て、振幅は Fig. 8 の實線以上にはならない事を申して置きます。又このやうな狀態では强制振動中に誘起され得る自己振動もやはりよく damp されるものです (之は私の他の研究から明瞭)。

○井口常雄君 飛行機では補助翼がブラブラするとフラッターを起す。そんな風にはなりませんか。

○妹澤克惟君 飛行機のフラッターでも翼が捩れる丈の時は補助翼の多少のブラブラは theoretical にも實驗的にも限界速度を高めるに反て良いのですが、實際には torsion だけでなく bending もあ

るものですから vibration が大きくなります。しかし只今の制振器はフラッターの問題とは全く關係のないものであることを御注意致します。又、關係のないことですがフラッターの限界速度に就ても種々の條件をよく御吟味下さる事を希望致します。

○松山武秀君　試驗水槽で水中突出物を曳きます時にもやはりバタバタします。取付の處のブラブラさが貴方の場合と違ふかも知れませんが、yield する程相當バタバタします。

○妹澤克惟君　周期が違ふと宜しくありません。自己振動の周期が大體船のと同じなら大變宜しいのです。その外 damper の質量や減衰力が適當でなければなりません。恐らく場合が大いに違ふものと御比較なさつてると想像致します。

○吉識雅夫君　先程の松山さんへの御答へになつた事ですが、舵の pivot の様にしたらよいと云ふ御話でしたが、舵を vibrate させて其の spring force を船體に持つて來て、2 degree of freedom の dynamical vibration absorber として absorb さすのでせうが、それでは spring force が出ないのではありませんか。

○妹澤克惟君　それは pivot は單に pivot であり、ばねに直接力が働く譯です。 pivot からばねまでの距離は一定以上の必要があります。

○吉識雅夫君　rudder は flexural vibration をさすばかりでなく、回轉振動もさす譯ですか。

○妹澤克惟君　さうです。此の實驗では實驗上やむを得ず幾何學的には回轉にも linear にもなりさうですが、實際にきくのは主としてばねの直接力のやうにしてあります。制振器に對する御了解を多謝。

○座長（平賀　讓君）　外に何方か御座いませんか。御座いませんでしたら妹澤、渡邊兩君の有益な paper に御禮を申上げたいと思ひます。先に横山君のは振動の原因を無くする事に就いて、今のは船體に出來た振動を相殺せんとする考へを理論的及び實驗的に試みられ、振動防止に關して誠に有益にして新しい考案であります。我々に取つては新しい考案の實行は中々困難でありますが、將來發達しまして良き實際方法となる場合にも、先刻申しました通り、小さな mass でなければ實行は可なり困難であらうかとも思はれます。尙實行方法はどうか妹澤博士に於かれまして、更に御研究の上御發表願ひたいと思ひます。此の極めて有益な paper の御發表に對して皆樣と共に拍手を以て感謝の意を表したいと思ひます。（一同拍手）