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Oscillating Flow About Two and Three-Dimensional Bilge Keels

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The paper describes an experimental investigation of the damping provided by bilge keels in an oscillating flow (in a large U-shaped water tunnel). Rectangular and square flat plates were placed adjacent to a solid boundary (with no gap) in order to simulate flow about bilge keels. The single free edge of the rectangular plate and the three free edges of the square plates were beveled to 60-deg angles so as to form sharp edges with included angles of about 60 deg. For comparison, another sharp-edged rectangular plate was tested without the wall proximity effects. All plates were held normal to the direction of the ambient flow. The Fourier-averaged drag and inertia coefficients were then calculated as a function of a suitable Keulegan-Carpenter number through the use of the instantaneous force acting on each plate.

Introduction

The impetus for this investigation comes from the potential application of the results to the prediction of the roll damping provided to the ships by bilge keels. Such keels as flat plates are also used as baffles in rocket fuel tanks to minimize fuel sloshing. In either case, flat plates adjacent or attached to a wall are used in situations where some type of damping is required. From an analytical point of view, flow about a flat plate (normal to the ambient flow) is of considerable interest because the position of the separation points is fixed at the edge of the plate, unlike the circular cylinder.

Much of the past work on plates in oscillating flow dealt with nominally two-dimensional conditions. The oscillatory motion was created either by using a large pendulum or by placing the plate normal to the flow in a standing wave environment. Data for flat plates in oscillating flow were obtained by Keulegan and Carpenter (1958), Martin (1958), and Ridjanovic (1962). The use of this data in the design of bilge keels for ships is demonstrated by Miller et al. (1974), Cox and Lloyd (1977), and Himeno (1981).

The early work for ship role damping was pioneered by Froude. He first proposed the use of "bilge pieces" in 1865 and then went on to measure a coefficient for resistance for a flat plate in 1874 using a "heavy free-swinging" pendulum. Inspired by Froude's work, the investigation of bilge keels continued to the present time. Some of these include, but are not limited to, those by Bryan (1900), Serat (1933), Adams (1938), Gawn (1940), Tanaka (1961), Gersten (1969), Sasajima (1954), Kato (1966), Bolton (1972), and Lofft (1973). It is now realized that the damping of ships and barges are controlled primarily by the vortex formation, surface wave generation, and, to a much lesser extent, by the frictional resistance. The vortex generation or eddy-making by the bare hull accounts for significant damping, especially at larger roll angles. However, the addition of bilge keels increases significantly the said type of damping of most ships, particularly at low speeds where the roll is generally the worst and maximum damping is most needed, as shown experimentally by Gawn (1940), and many others since then. Ikeda et al. (1977) have shown that the "lift" effect on the hull increases with the Froude number, but the eddy-making components of their bilge keels show very little or no change with forward speed. The roll damping and barge-roll response are discussed by Robinson and Stoddart (1987) and Standing et al. (1992). The use of a vortex tracking method to the prediction of roll damping of a two-dimensional floating body is presented by Braathen (1987).

The investigations of damping of fuel sloshing in large storage tanks have produced some interesting results involving flat plates, especially through flow visualization (see, for example, Cole and Gambucci, 1961; Schwind et al., 1964; and Shih and Buchanan, 1971).

In spite of the extensive work done on damping, in general, and on bilge keels, in particular, there has not yet been a systematic investigation of the effect of wall proximity on fluid forces acting on rectangular and square plates immersed in a sinusoidally oscillating flow.

Experimental Facility and Procedures

The U-shaped water tunnel, used in the present investigation, has been described in detail a number of times (see, e.g., Sarpkaya 1976, 1977, 1986a, 1986b). Here, only a brief description, together with the modifications made to it for its adaptation to the present investigation, is presented. The water tunnel consists of a 0.91-m wide, 1.42-m high, and 10.67-m long horizontal section and two vertical legs of 0.91 m by 1.83 m by 7.62 m. The top of one of the legs of the tunnel is connected to an air supply system. A butterfly valve near the top of the tunnel oscillates with a frequency exactly equal to that of the natural frequency of the water in the tunnel. The flow amplitude may be maintained at the desired level indefinitely through the use of a fan and an electronic feedback control system. The characteristics of the planar oscillatory flow (velocity distribution and turbulence) have been described in Sarpkaya (1976).

As many as six different methods have been used to determine the characteristics of the ambient flow at the test section. These ranged from the visual observation of the maximum and minimum water levels in both legs of the tunnel to the use of an LDV system (Sarpkaya 1976, 1986a). These methods yielded the ambient-flow acceleration, velocity, and displacement with an error less than 0.05 percent. The signals generated by the capacitance wires and differential pressure transducers were entirely free from noise and were never filtered.

The test bodies consisted of 6-mm thick aluminum flat plates. Two square plates (76 mm by 76 mm and 102 mm by 102 mm) and a rectangular plate (91 cm by 102 mm) were used. The square plates were attached to a force transducer with a thin streamlined stinger arm, near their bases. The rectangular plate, which spanned the entire tunnel width, was attached to two force transducers at the ends. For wall proximity effects, the single free edge of the rectangular plate and the three free edges of the square plates were beveled to 60-deg angles so as to

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form sharp edges with included angles of about 60 deg. All plates were held normal to the direction of the ambient flow and to a solid bottom boundary. A small rectangular channel (slightly wider than 8 mm) was created in a false bottom, immediately below and along the length of the plate. The test plate was lowered (about 6 mm) into the groove and then calibrated. The width of the groove was such that the edges of the plate did not come into contact with the groove even for the largest force exerted on the plate. The purpose of the groove was to minimize the effect of the free space between the plate and the tunnel bottom. In any case the free space between the edges of the plate and the groove did not exceed about 1 mm in any direction. Another rectangular plate of the same size as the one just described was manufactured for a series of reference tests without the wall effect. It too spanned the entire width of the tunnel; however, both of its free edges were beveled to 60-deg angles so as to form sharp edges with included angles of about 60 deg.

The frequency parameter β (Sarpkaya, 1976) for the 91-cm by 102-mm rectangular plate and the 102-mm by 102-mm square plate was β (= Re/K) = 1845. For the 76-mm by 76-mm square plate, it was β = 1035. For the same ambient flow, the Keulegan-Carpenter number K = UmT/h for the larger plates (of height h = 102 mm) was 33 percent smaller and the Reynolds number was 33 percent larger than those for the smaller plate (h = 76 mm).

Presentation and Discussion of Results

The force exerted on a plate was expressed as usual in terms of the Morison approximation by

$$\frac{2F}{\rho h w U_m^2} = -C_d|\cos\theta|\cos\theta + C_m\frac{\pi^2}{\mathrm{K}}\sin\theta \qquad (1)$$

in which F represents the total force acting on plate; ρ, the density of fluid; h, the height of the plate; w, the width of the plate; U_m, the amplitude of the oscillating fluid characterized by $U = -U_m\cos\theta$; and C_d and C_m, the Fourier-averaged drag and inertia coefficients given by

$$C_d = -\frac{3}{4}\int_0^{2\pi}\frac{F_m\cos\theta}{\rho h w U_m^2}\,d\theta \qquad (2a)$$

$$C_m = \frac{2\mathrm{K}}{\pi^3}\int_0^{2\pi}\frac{F_m\sin\theta}{\rho h w U_m^2}\,d\theta \qquad (2b)$$

where F_m represents the measured force. The drag and inertia coefficients for a plate held normal to the flow depend on K = U_mT/h, Re = U_mh/ν (or β = Re/K), and h/w. For a rectangular plate spanning the test section, w becomes unity in the foregoing equations and F represents the force per unit length. Other coefficients, such as the rms and maximum values of the force in each cycle were calculated. These will not be reported here for sake of brevity.

The data obtained in the present investigation are compared with three sets of data obtained in an unbounded fluid (no appreciable wall-proximity effects). Keulegan and Carpenter (1958) used the midsection of a rectangular basin with standing waves. The plates (h = 40 mm to 76 mm) were fixed horizontally and below the water surface. The average values of the drag and inertia coefficients over a wave cycle were calculated through Fourier averaging. The Reynolds number ranged from 5200 to 14,200 and the Keulegan-Carpenter number ranged from 1.7 to 118.2; i.e., β varied significantly. Martin (1958) used 16-mm to 63-mm wide and 31-cm long rectangular plates in a nominally two-dimensional flow situation (bounded by side walls) and obtained mean drag coefficient data through the use of a pendulum, brought up to a predetermined amplitude of oscillation slowly by hand during a fairly large number of oscillations and then allowed to swing and decay freely. The visual observations were made of the number of oscillations required before an angular decay of a certain fraction of a degree occurred. The Keulegan-Carpenter number ranged from 1.7 to 26 (the Reynolds numbers were not specified). Ridjanovic (1962) used essentially the same apparatus and procedures to determine the effect of three-dimensional flow conditions on the mean drag coefficient. The w/h ratio of his test plates ranged from 19.4 to 1. The Reynolds number was not specified, but K ranged from 0.7 to 13. It is again emphasized that none of the three experiments described in the foregoing involved wall-proximity effects.

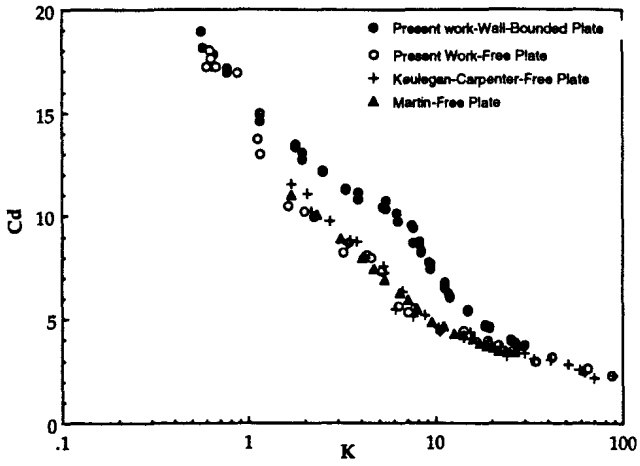

Fig. 1 Drag coefficient versus Keulegan-Carpenter number for wall-bounded and free rectangular plates

The drag coefficients obtained in the present investigation[1] for free and wall-bounded rectangular plates are shown in Fig. 1 as a function of K = U_mT/h, together with those obtained by Keulegan and Carpenter (1958) and by Martin (1958) with free plates. The drag coefficient for the wall-bounded plate exhibits a relatively sharp change in slope near K $\approx$ 8 and the effect of the wall is to increase C_d as much as 50 percent in the range 1 < K < 30 relative to the free-plate case. It is also noted that the Keulegan-Carpenter data, the Martin data, and those obtained in the present investigation with free plates are in good agreement.

The comparison of the inertia coefficients is somewhat more involved since C_m, unlike C_d, depends on the choice of the

[1] For sake of brevity, a plate simulating two-dimensional flow conditions (two side walls to eliminate the end effects) will be referred to hereafter as a "free plate." A plate which is resting on the wall (whether the two ends are bounded or free) will be referred to as "wall-bounded."

Nomenclature

A = amplitude of oscillation
C_d = Fourier-averaged drag coefficient
C_m = Fourier-averaged inertia coefficient
D = cylinder diameter
F = total fluid force
F_m = measured force
h = height of plate
K = $2\pi A/h = U_mT/h$, Keulegan-Carpenter no.
Re = Reynolds no., U_mh/ν
T = period of oscillation
t = time
U = velocity of ambient flow
U_m = maximum velocity in cycle
w = width (length) of plate
β = frequency parameter, Re/K
$\theta = 2\pi t/T$

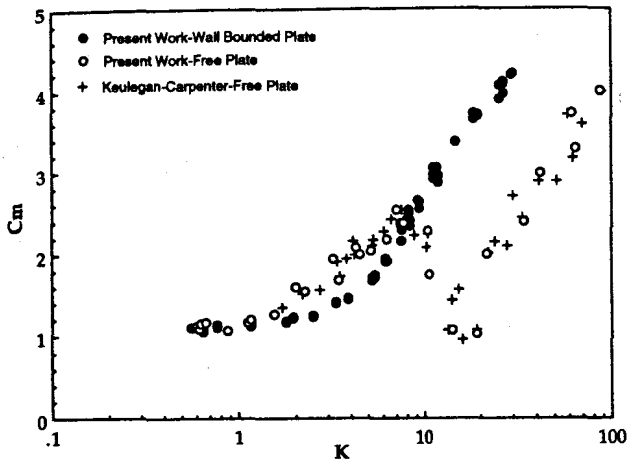

Fig. 2 Inertia coefficient versus Keulegan-Carpenter number for wall-bounded and free rectangular plates

reference volume. If the height h of the wall-bounded plate is taken as the reference diameter of the fictitious cylinder encircling the plate, the reference volume becomes $\pi h^2/4$ and this, in fact, is the volume used for all free plates. For a wall-bounded plate, one may regard the wall as a plane of symmetry and use h as the radius, rather than the diameter, of the cylinder encircling the real and the image plates. Then, one uses $\pi h^2/2$ (half of the cylindrical volume) in defining C_m for the wall-bounded plates, since the measured force is acting on the plate above the wall. Even though such a plot is relatively more meaningful, it was decided to present the data as they are obtained from Eqs. (2), i.e., using $\pi h^2/4$ as the reference volume whether the plate is free or wall-bounded. There is, of course, no difficulty in plotting the data in any other form.

Figure 2 shows the inertia coefficient for the wall-bounded and free plates as a function of the Keulegan-Carpenter number. Clearly, the free and the wall-bounded plates yield nearly identical C_m values for K smaller than about 8. For K > 8, C_m for the wall-bounded plate continues to increase due to the preservation of symmetry between the real and the image plates. In other words, the usual dip in C_m, often encountered in free-cylinder and free-plate experiments in the drag/inertia-dominated regime, is completely absent in C_m for wall-bounded plates. The C_m values for the free plate drop sharply for K larger than about 8 and then increase in a manner similar to that for the wall-bounded plate. The sharp drop in C_m in the range of 8 < K < 13 is often attributed to the onset of vortex-shedding asymmetry. Subsequently, the number of vortices shed during one-half-cycle increases and the flow becomes quasi-steady. It is apparent from the foregoing that the effect of the wall proximity is much more pronounced on the inertia coefficient than it is on the drag coefficient. There is no doubt that the behavior of C_d and C_m is accompanied by equally dramatic differences in the topology of the two flows, brought about by the presence or the absence of the wall.

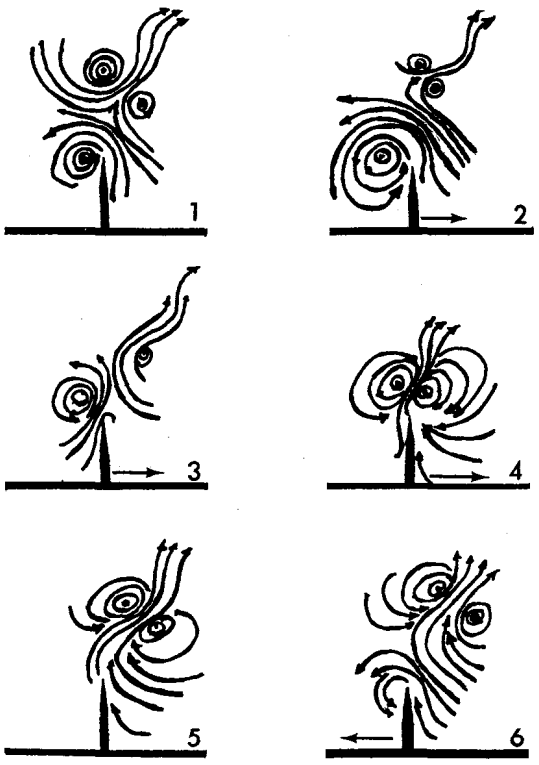

Fig. 3(*a*) Sketches of the flow topology about a rectangular plate for K < 3

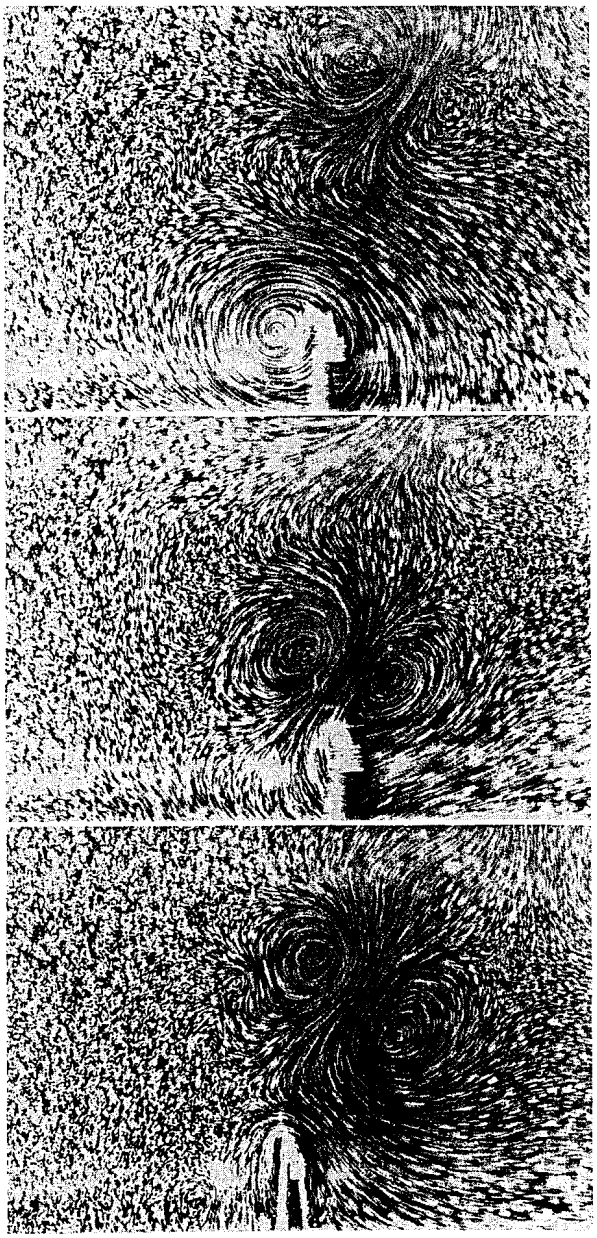

Fig. 3(*b*) Photographs of the flow pattern about a rectangular plate for K = 2.5. Frames 1, 4, and 6 in Fig. 3(*a*) correspond to the top, middle, and bottom pictures in Fig. 3(*b*).

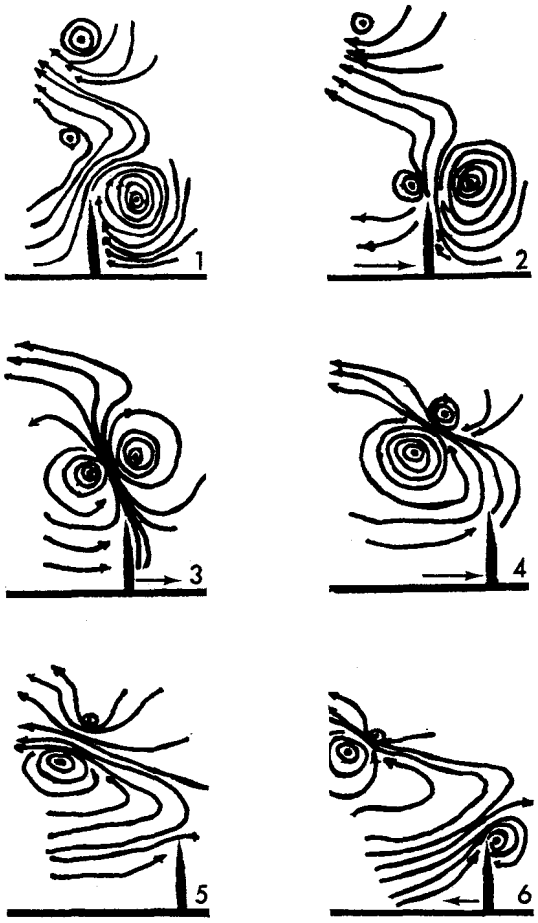

Fig. 4 Sketches of the flow topology about a rectangular plate for 3 < K < 8

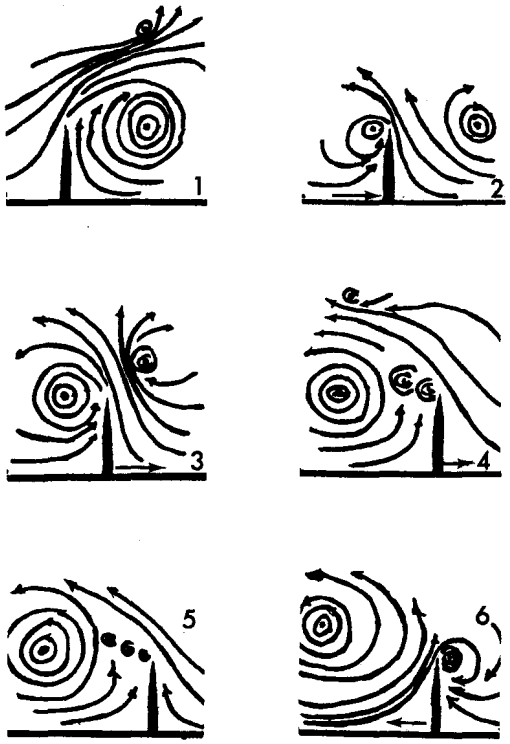

Fig. 5 Sketches of the flow topology about a rectangular plate for K > 8

Extensive flow visualization studies have been carried out to understand the relationship between the force coefficients and the change in flow topology. The results have shown that for K less than about 3 (see Figs. 3(*a*) and (*b*) where the photographs correspond to frames 1, 4, and 6 in Fig. 3(*a*), a newly created vortex forms a counter-rotating pair with the vortex shed in the next half-cycle and convects away in a diagonal direction, either to the upper left or to the upper right of the edge of the plate. The specific direction is random and dictated primarily by the starting conditions. However, once the direction is established, it remains in that direction for an indefinite period of time. The possibility of direction changes along the span has been investigated and it was found that, unlike the spanwise mode variations along a circular cylinder, the flow about the edge of the rectangular plate is very robust and nearly coherent. In the range $3 < K < 8$ (see Fig. 4), the vortex shedding pattern becomes more complex primarily because of the fact that the vortex shed in the previous half-cycle begins to decay (even faster with increasing K) by the time the new vortex in the next half-cycle reaches its full strength. Thus, instead of vortices convecting as a couple, as in the case of $K < 3$, the vortex of smaller strength begins to orbit about the newly formed stronger vortex. In short, a new vortex is formed in each half-cycle on opposite sides of the plate and the weaker and older of these vortices begins to orbit about the most recent one. This increases the size of the separation region near the plate, and, therefore, the flow does not start anew at each half-cycle as a potential flow as in the case of $K < 3$. It is apparent that the time for the shedding of a fully grown vortex and the time for the decay (controlled by the prevailing conditions) of an existing vortex, with which the newly shed vortex interacts, are of fundamental importance, not only in prescribing the particular topology of the flow, but also the character of the time-dependent force.

For K larger than about 8 (see, Fig. 5), not only a large vortex is formed at each half-cycle, but additional small vortices of the same sign emanate from the edge of the plate since the duration of flow in one direction is sufficiently long. Thus, one may regard $K \approx 8$ as the approximate beginning of a quasi-steady state where the number of vortices shed during one-half-cycle increases and the drag coefficient begins to decrease more rapidly with further increases in K.

Figures 6 and 7 show the drag and inertia coefficients for the wall-bounded square plates. Also shown in Fig. 6 is the data obtained by Ridjanovic (1962) using a 102-mm by 102-mm square free plate. It is apparent from our data that C_d does not depend on the size of the plate, within the range tested here, as

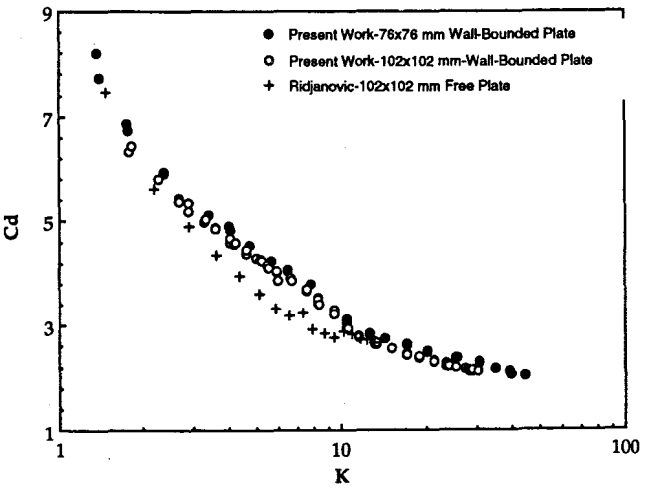

Fig. 6 Drag coefficient versus Keulegan-Carpenter number for wall-bounded and free square plates

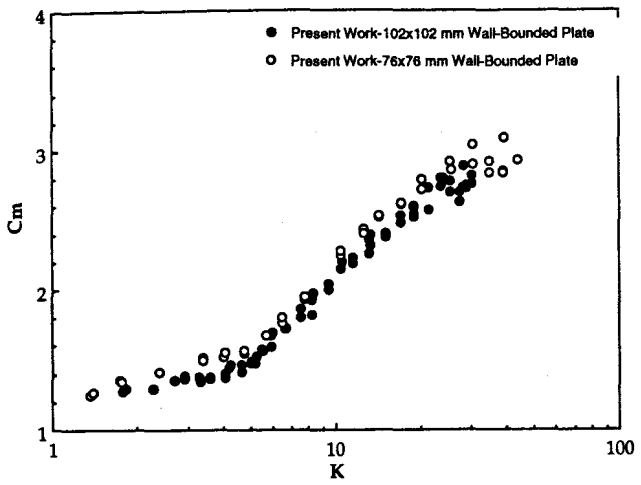

Fig. 7 Inertia coefficient versus Keulegan-Carpenter number for wall-bounded square plates

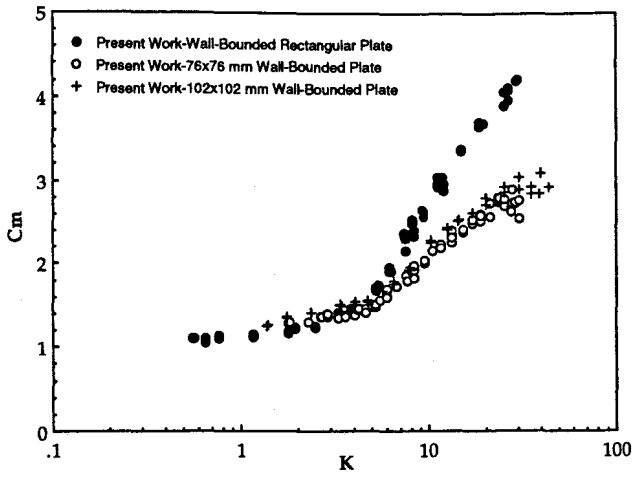

Fig. 9 Inertia coefficient versus Keulegan-Carpenter number for wall-bounded square and rectangular plates

far as the wall-bounded plates are concerned. Normally, one would not expect Reynolds number dependence for flows about sharp-edged plates, save for the effect of the boundary layer on the wall. The Stokes's boundary layer thickness was about 10 mm, i.e., δ_S/h was about 0.15 for the smaller plate and about 0.10 for the larger plate. Apparently, the effect of the wall is to increase C_d relative to the free-plate case. However, the three-dimensional nature of the flow has much stronger influence on the flow and force coefficients than the wall, as evidenced from a comparison of Figs. 1 and 6. The square plates have significantly smaller drag coefficients for a given K and there appears to be a relatively smoother decrease in C_d with K. A closer look reveals that for K smaller than about 2.5 there is a steep decrease in C_d. This is followed by an intermediate region bounded by K = 10. The final region shows a more gradual decrease of C_d towards its steady-state value.

The inertia coefficients shown in Fig. 7 again confirm the fact that the Reynolds number is not relevant here, except through the secondary influence of the Stokes boundary layer, as noted in the foregoing. Finally, Figs. 8 and 9 show a comparison of the corresponding drag and inertia coefficients for wall-bounded rectangular and square plates. As expected, the C_d for the rectangular plate is considerably larger than that for the square plate for all values of K. Thus, for bilge keels, it is preferable to use plates with as large w/h as possible to maximize damping. As far as the inertia coefficients are concerned, C_m for K > 5 grows at a higher rate. For K smaller than about 5, they yield nearly identical values. This is due to the fact that they approach nearly identical limiting values as K approaches zero: $C_m = 1$ for the rectangular plate and $C_m = 1.158$ for the square plate (Sarpkaya and Isaacson, 1981).

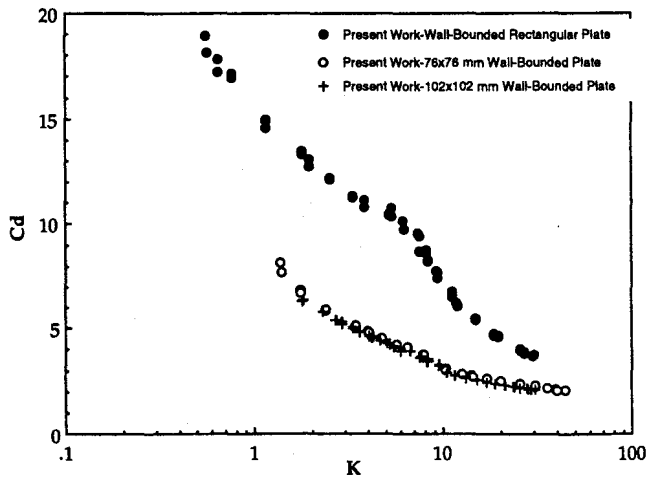

Fig. 8 Drag coefficient versus Keulegan-Carpenter number for wall-bounded square and rectangular plates

The experimental work has been followed by a discrete vortex analysis of the flow (Sarpkaya, 1994). These will not be discussed here in detail. Suffice it to note that the resulting vortex patterns were found to be in reasonable agreement with that observed in flow visualization, even though the forces calculated were about 15 percent larger than those measured.

Conclusions

The results presented herein have confirmed the fact that bilge keels do, in fact, provide a high degree of damping, particularly at small amplitudes of oscillation. This damping is a direct consequence of the vortices shedding from the edges of the plate. The use of the damping coefficients obtained from free-plate tests are not necessarily applicable to wall-bounded bilge keels. The effect of the wall proximity on the damping provided by a rectangular or square plate is quite significant relative to that of similar free plates, and depends on the aspect ratio of the plate and the shape of the wall to which it is attached. The wall-proximity strongly affects the inertia coefficient, particularly that of a rectangular plate with negligible end effects. The C_m values approach their limiting theoretical values as K $\rightarrow$ 0. The effect of the Reynolds number on the resistance of square and rectangular plates is negligible. The Stokes boundary layer provides only a secondary influence. For a ship at moderate speeds, the effect of the boundary layers may become more significant depending on the speed of the ship and the angle of attack about which the bilge keels oscillate.

It is highly desirable to carry out a detailed numerical analysis of the three-dimensional, unsteady, turbulent flow about bilge keels through the use of one of the existing finite-difference and/or finite-element codes. A relatively simple analysis of the flow about a two-dimensional wall-bounded rectangular plate through the use of the discrete vortex method is capable of simulating the flow topology, at least during the early stages of the motion. However, the forces calculated are about 15 percent larger than those measured. Suffice it to note that complexities of unsteady nonequilibrium turbulence with or without separation under a variety of time-dependent flow conditions (periodic flows, transverse or bidirectional dynamic response of bodies) are far from understood. It appears that the prediction of the evolution of the wake, after the spontaneous separation of the

viscous layer, is possible only for relatively small Reynolds numbers. For Reynolds numbers of any practical importance, the use of approximate turbulence models and experiments still appear to be the only recourse. A practical need exists for developing more general techniques capable of predicting the behavior of both laminar and turbulent unsteady boundary layers.

Acknowledgments

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Errata

In the November 1995 issue, the following papers were mistakenly credited to Subrata Chakrabarti as the Associate Technical Editor:

"Hydrodynamic Damping, Flow-Induced Oscillations, and Biharmonic Responses," by T. Sarpkaya, pp. 232–238.

"Ship Hull Bottom Slamming," by P. Rassinot and A. E. Mansour, pp. 252–259.

They should have been properly credited to Dr. Cengiz Ertekin as the Associate Technical Editor. The Technical Editor regrets this mix-up.