Stress Analysis of a Piping System Under High-Intensity Shock Loads

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SUMMARY

This paper summarises a methodology for the structural analysis of piping systems under high-impact shock loads, such as those predicted in naval codes. An equivalent spring-mass system is determined in such a way that, under the action of fictitious forces, its response reproduces well the naval specifications. Numerical analysis are performed using one FEM code. The stress analysis of an hypothetical piping system is then performed.

INTRODUCTION

The mechanical design of naval structures shall ensure that its machinery and equipments can withstand the shock effects of non-contact underwater explosions. Design shock loads are related to grades of shock severity, specified by standars/Naval Codes/ and are characterized by their high intensities and short durations. Even when the ship is not severely damaged, it is important to ensure that its internal systems, such as pressure vessels, pipings, machinery and equipments remain operational during and after the event of the explosion. It is therefore necessary to verify that the stresses due to the shock loading are kept under suitable stress limits thus a transient response of the structure under the dynamic excitation have to be performed.

This paper presents a methodology for the design of ship components under shock loads. As it will be seen, the method consists basically in substituting the excitation acting in the boundaries of the system by loads applied to a fictitious structure, which response reproduces well the shock parameters predicted by the naval codes.

Section 4 illustrates the application of the method to a piping system designed according to ASME Code, Section III, /ASME/. The piping system is assumed to have linear-elastic behaviour, and the shock excitation is simulated by means of a short duration, half-sine force pulses. A finite element code/ANSYS/ is employed on solving the problem.

THE SHOCK ENVIRONMENT

After the event of an underwater explosion, pressure waves propagate in the water at the speed of sound, reaching intensities up to 10 MPa in less than 1 ms. The direct waves, together with additional waves reflected from the water surface and the bottom of the sea, reach the ship provoking large local displacements of the hull structure and the systems attached to it. Depending on the way the piping systems are connected to he hull, they may be subjected to acceleration up to hundreds of times the gravity.

The shock parameters, which are obtained experimentally, are thus related to the virtual mass of the equipments under design. The virtual mass is a concept created to take into account the fact that a flexibly supported item will initially lag behind respect to the hull displacement, then subsequently move, eventually faster than the hull itself. It all depends on the real masses of the components and on the stiffness of their supports. Fig. 1, obtained from the naval codes, shows the accelerations, velocities, fundamental periods, pulse duration, displacements and time to reach the maximum velocity varing as a function of the virtual mass.

The codes state that the data obtained from Fig. 1 shall be expressed in terms of velocity transients at different points of the hull structure.

The dynamic response of the internal systems during and after the event of the shock pulse may be evaluated. A typical curve for the velocity pulse is presented in Fig. 2.

METHOD OF ANALYSIS

An initial difficulty which may arise in the analysis of a system under a transient load as shown in Fig. 2 is that not every computer code disposes of routines ables to perform the dynamic analysis of such a velocity excitation. This is, actually, the case of the ANSYS code. As long as ANSYS allows to define a displacement transient, one could integrate the velocity pulse and impose the resulting displacement transient to the system, hence determining its response. However, it would require to define very refined displacement time histories, for the whole period assumed for the integration of the motion equations.

A second option could be to determine the spectrum of velocities from Fig. 2 and then perform a spectrum response analysis. But, as long as ANSYS does not allow multiple spectrum excitation, a large conservatism should be assumed.

A third method, which will be described in the following, substitutes the velocity pulse by a half-sine force pulse, allowing an accurate reproduction of the shock parameters, and a good understanding of the system response physics.

Thus, consider the velocity pulse shown in Fig. 2, neglecting the damping "δ" and the rigid body velocity B.

Derivating then the pulse, one determines accelerations as shown in Fig. 3.

Note that this acceleration transient is the response of a single degree of freedom (SDOF) mass-spring (M-K) system, excited by a half-sine force pulse, of short-duration (that is, $T2 = \pi.\sqrt{(M/K)} \rangle\rangle T1$), shown in Fig. 4. Now, if the mass M is arbitrarily chosen, it is possible to determine the stiffness K and the force amplitude F0 that will reproduce the accelerations of Fig. 3 and, hence, the shock parameters stated by the codes, Fig. 1.

In short, with this artifice one may generate the hull motions which are going to excite the internal systems, by applying a fictitious force transient to a spring-mass system with the same natural frequency of the hull structure. To ensure that this frequency is not influenced by the mass m of the internal system, a value $M \rangle\rangle m$ must be chosen.

The parameters M, K and F_o on Fig. 4 are obtained writing down the response of the fictitious SDOF M-K system, as follows/Clough, 1975/:

$$\frac{a(t)}{F_o/K} = -\frac{\beta\, w^2}{1-\beta^2}\left(\beta \text{ sen } \bar{w}t - \text{sen } \frac{\bar{w}t}{\beta}\right), \quad 0 \langle t \langle T1 \quad \text{phase I} \quad (1)$$

$$\frac{a(t)}{F_o/K} = -\frac{v(T1)}{F_o/K}\, w \text{ sen } w\,(t-T1) - \frac{x(T1)}{F_o/K}\, w^2 \cos w\,(t-T1), \quad t \rangle T1$$

$$\text{phase II} \quad (2)$$

where a(t) is the acceleration at time t, w is the system natural frequency, w' is the applied load frequency, $\beta = w'/w$. v(T1) and x(T1) are respectively the velocity and displacement of the mass point M at the end of the force pulse, T1.

Values of β and Fo/K are numerically determined. β is determined by trial and error and Fo/K is obtained from the maximum value the expression a(t)/(Fo/K) assumes. β is chosen in such a way that the ratio between the maximum and minimum values of a(t)/(Fo/K) respectively in phase I and II is the same as the ratio of the maximum and minimum acceleration values taken from the standards/Naval Codes/. Once β is known, the natural frequency of the equivalent SDOF M-K system is obtained from the following expression:

$$w = \frac{\bar{w}}{\beta} = \frac{\pi}{\beta T1} \qquad (3)$$

Then the stiffness K is evaluated, choosing afterwards a suitable value of M, and hence the force pulse intensity Fo may be obtained from the value of a(t)/(Fo/K) obtained for that β. The motion parameters of the M-K system determined in this way are the same as those predicted by the codes.

Fig. 5 shows schematically how the method developed above may be employed to determine the response of a piping system subjected to a shock excitation. It is seen in the figure that all piping supports and anchors are subjected to the same base movements, thus there is only one equivalent M-K system. In the general case, when the virtual masses of equipments and support structures are not the same, several M-K systems would be required in order to simulate the different displacements of the several parts of the hull.

RESULTS

In order to verify the accuracy of the method described above, for a particular shock level prescribed in the codes, the equivalent M-K system was modelled by means of the ANSYS code. A piping system, which total mass m was much smaller than M, was also attached to it.

The ratios between displacements, velocities and accelerations predicted by the codes and those determined with the proposed model are shown in Tab. 1 It is seen that a good accuracy is achieved.

Table-I: Comparison of the Results

Ratio: Naval Code/Proposed Method		
Displacement	Velocity	Acceleration
1.060	1.046	1.012

Subsequently, the method was applied to the design of piping systems under several different shock excitations. A typical example, shown in Fig. 6, considers a heavy equipment attached to the hull by a very stiff structure (the virtual mass of the system in this region is large) and another equipment connected to the hull by resilient supports (small virtual mass of the system in this region). Yet another virtual mass is associated to the hull's region to which the base of the piping supports are connected. Therefore, on presence of different virtual masses, the hull will impart different shock transients to the piping system. The piping must be stiff enough to resist the enertia forces without exceeding primary stress limits but, on the other hand, enough flexibility must be provided to allow the different base movements without exceeding the secondary stress limits.

The model represented in Fig. 6 had a discretization refined enough to capture the bending mode shapes with frequencies up to three times the frequency associated to the maximum amplification of a SDOF system, when subjected to the force pulses here defined. The FE code recomendations for the determination of the time-step was taken into account, but several tests were also undertaken in order to determine its appropriate length for the integration of the equation of motion equations, as the response is very sensible to the time step variations. Finally, the ANSYS code was employed to determine the resulting stresses. The system was qualified acording to ASME Code, subsection NB, the maximum stress intensity reaching about 60% of the allowable stress.

CONCLUSION

A methodology for the design of piping systems under high-impact shock loads was presented. The method was developed assuming the use of computer codes wich do not allow excitations defined in terms of velocity transients or multiple response spectra. An worth characteristic of the method is that it avoids a large volume of data input and gives in addition a good insight of the shock phenomenon physics.

The results obtained suggest also that the presented formulation is well suited for the design of naval piping systems, as shown in section 3.

REFERENCES

/1/ Code for the design of Naval Systems, classified documents.

/2/ ASME B&PV Code Section III, Subsection NC, 1986.

/3/ ANSYS User's Manual, Version 4.2, 1985.

/4/ Dynamics of Structures, Clough and Penzien, McGraw Hill, 1975.

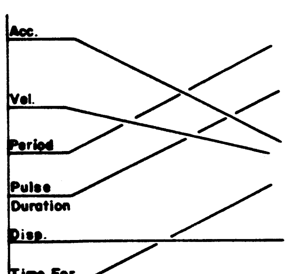

FIG.1 - SHOCK PARAMETERS

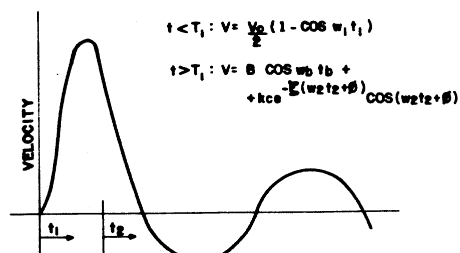

FIG.2 - TYPICAL VELOCITY PULSE

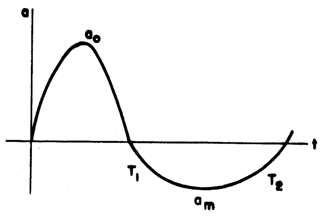

FIG. 3 - ACCELERATION HISTORY

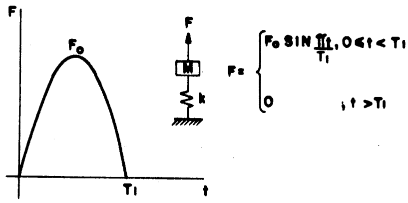

FIG. 4 - SINE PULSE FORCE

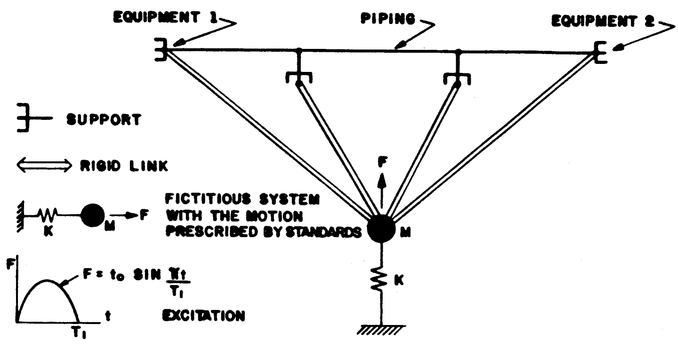

FIG. 5 - MODEL FOR THE PIPING ANALYSIS

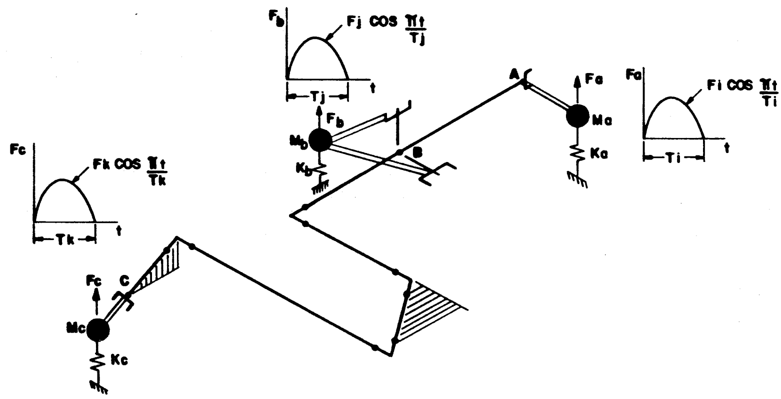

FIG. 6 - MODEL FOR THE SHOCK ANALYSIS