DREDGING DYNAMICS AND VIBRATION MEASURES

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ABSTRACT

The demands for dredging have found a profound increase in recent times. The reason for this could be attributed to growing demand for land and waterway transportation. A typical example is the Salt Lake Kolkata developed by land reclamation from maintenance dredging carried out at Hooghly River. The demand for dredging will be increasing further due to rapid urbanization and economic viability for waterway transportation. Therefore, there is growing demand for understanding the operational characteristics during dredging. As dredging is a process of excavating material from under water surface it undergoes severe vibration due to the interaction between soil-structure and water wave-structure. For that reason, there is major issue of damaging the part and decreased productivity. The same case happens for cutter suction dredger (CSD) what we are considering for recent study. In CSD the cutter damage is a major issue and the design of drive system for cutter is a challenging problem for the researchers. During the cutting operation huge amount of force and power are required. It is very important for the designers to have sufficient knowledge about the environment to operate the cutter drive system safely and economically. The present study considers the dynamic forces acting on the cutter drive during the operation. A mathematical model of the cutter suction dredger was derived using Finite element method. Two shaft having different orientations connected using a universal joint were modelled as spring elements. The shaft element consisted of five degrees of freedom per node. The effect of wave loading was calculated based on Morison equation. The inertia force contribution from the Morison equation contributes to the added mass effect on the cylindrical shaft hence influences the free vibration. The natural frequency and mode shape for a straight and shafts inclined at a specific configuration were analysed. A parametric study was carried out by varying the spring stiffness and damping, and its influence on the natural frequency and mode shape was determined.

KEY WORDS

Dredging, Cutter Suction Dredger, Universal joint, Morison Equation

NOMENCLATURE

ρ	Density (kg/m^3)
a	Amplitude of wave (m)
A	Area of the cylindrical shaft (m^2)
C_d	Drag coefficient
C_m	Inertia coefficient
D	Diameter of the shaft (m)
E	Modulus of Elasticity (N/m^2)
F	Wave force (N)
g	Acceleration due to gravity (m/s^2)
G	Modulus of Rigidity (N/m^2)
h	Depth of water (m)
I	Moment of inertia (m^4)
J	Polar moment of inertia (m^4)
K	Elemental stiffness matrix
$[K]$	Global stiffness matrix
k	Wave number
L	Length (m)
M	Elemental mass matrix
$[M]$	Global mass matrix
M_a	Added mass
U	Water particle velocity (m/s)
ω	Water wave frequency (Hz)

1. INTRODUCTION

It is very important to maintain or increase the depth of navigation channels, anchorages, or berthing areas for safe passage of boats and ships. So dredging operation is frequently carried out at the port to maintain the draft condition for the entering vessel. This involves in the removal of sediments and debris from the bottom of lakes, rivers, harbours, and the other water bodies. One of the equipment used for the dredging operation is the cutter suction dredger. The layout of the cutter suction dredger system is shown in the figure below.

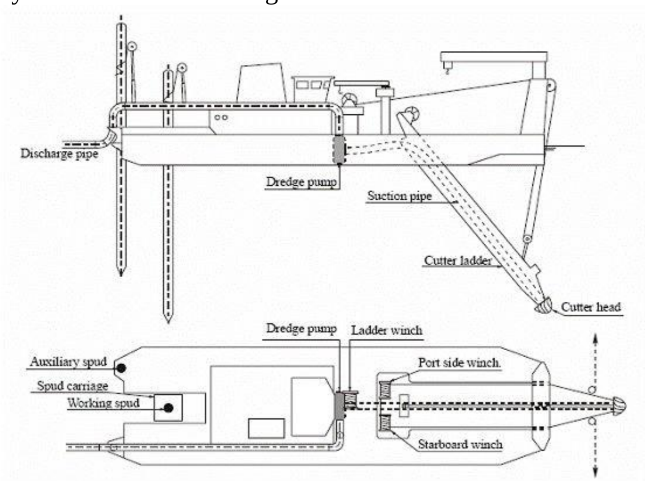

Fig 1: the top view and front view of the cutter suction dredger. [6]

An estimation of response due to wave loading can be helpful for quantifying the overall contribution to the vibrational effect from response to impulsive load. This paper is primarily focussed on the influence over the linear vibration of the cutter suction dredger subjected to wave loading.

Recent work has focussed on the development of the mathematical modelling of the differently oriented cutter drive using finite element method. Morison equation helps to predict the effect of wave loading and the added mass effect due to the inertia force contribution.

For smooth and economical operation of a dredger, it is important to understand the vibration characteristics. The major parameters responsible for the performance and workability are the forces exerted on the cutter, different soil characteristics, responses due to various wave conditions etc. which can affect its working condition. Many researchers have been working on these areas to increase the efficiency of the dredger.

The force calculation on the cutter is the primary factor for dredging which is responsible for dredging performance. The dredging process during excavating material from sea bed surface using cutter suction dredger, it is very important to know about the forces exerted on the cutter. Young et al. [1] developed an experimental set up to calculate the forces in horizontal, vertical and axial direction. To know these forces a static analysis was done on carriage by the application of static loads to the cutter head in horizontal vertical and axial directions. The static equations were validated by the static finite element analysis through SolidWorks, a modeling software. The experimental and theoretical calculations were compared over the dredge cutter head. This is also important to make the optimal design of the cutter to reduce the undesired forces resulting less damage to the cutter and increase the performance which can ultimately reduce the cost of the dredging project. Ma Yasheng et al. [2] explored the 3-d cutting forces calculation of the cutter head. The emphasis was given on blade cutting forces with the small cutting angle and calculated for the saturated sand. The pressure distribution also calculated using MATLAB to ensure the similarity of the results with their experimental work.

The performance of the dredge is also affected by different wave condition. Forces due to different waves also affect the workability resulting reduction on efficiency. Wave height and frequencies are important parameters to be considered for the dredging operation. Koning et al. [3] experimented in laboratory for the soil-cutter head interaction under wave condition. It was found that the wave frequency influences the interaction between soil and cutter head. It was observed that the load on the cutter head can be reduced by low swing velocities and higher cutter revolution. Keuning et al. [4] worked in irregular waves to develop mathematical model for computation of behavior of cutter suction dredger. The work shows the importance of the soil behavior in the mathematical model. The behavior of dredge is heavily influenced by nonlinear characteristics of soil. This dynamic behavior depends on the type of soil. Hence it is important to get the knowledge about the soil reaction forces on the cutter.

The paper focuses on the dynamic behaviour and response of the cutter suction dredger shaft. The shaft is subjected to wave loading. The soil structure interaction is initiated by understanding the effect of an impulse is provided at the free end. This is to emulate the condition of a cutter in a linear manner.

2. MODELING

The cutter suction dredger consists of two shafts connected using a universal joint. The purpose of the universal joint is to transmit the torque from the straight shaft connected to the motor. The straight shaft which is connected to the prime mover is longer in comparison to the inclined shaft. A schematic sketch of the system is shown in the figure below.

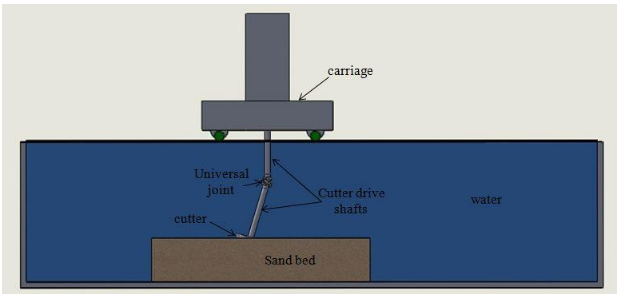

Fig 2: Equivalent model for the dredging operation

2.1 FINITE ELEMENT MODEL

A theoretical model of the cutter suction dredger shaft was modelled using finite element analysis. The shaft was considered as an Euler-Bernoulli beam. The effect of bending was considered in two planes Each node had five dof. The extra dof incorporated the effect of axial dof. The schematic diagram of the vibration model of the system is shown in figure.

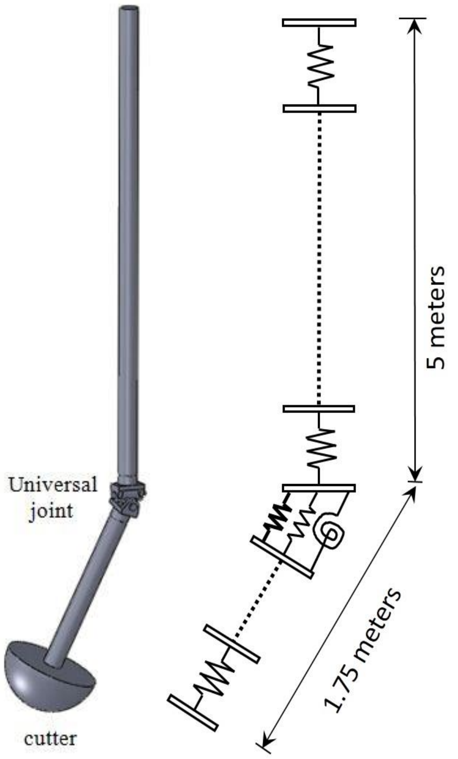

Fig 3: Diagram of cutter drive with cutter, its spring equivalent system

The inclined portion of the shaft included a transformation matrix. The universal joint was modelled using a spring element of high stiffness. The numerical values of the spring constants, moment of inertias were calculated from the shaft material properties and shaft dimension information.

The elemental mass matrix of the system is [5]

$$M = \rho AL/420\begin{bmatrix} 156 & 0 & 0 & 0 & 22L & 54 & 0 & 0 & 0 & -13L \\ 0 & 156 & 0 & -22L & 0 & 0 & 54 & 0 & 13L & 0 \\ 0 & 0 & 140J/A & 0 & 0 & 0 & 0 & -70J/A & 0 & 0 \\ 0 & -22L & 0 & 4L^2 & 0 & 0 & -13L & 0 & -3L^2 & 0 \\ 22L & 0 & 0 & 0 & 4L^2 & 13L & 0 & 0 & 0 & -3L^2 \\ 54 & 0 & 0 & 0 & 13L & 156 & 0 & 0 & 0 & -22L \\ 0 & 54 & 0 & -13L & 0 & 0 & 156 & 0 & 22L & 0 \\ 0 & 0 & -70J/A & 0 & 0 & 0 & 0 & 140J/A & 0 & 0 \\ 0 & 13L & 0 & -3L^2 & 0 & 0 & 22L & 0 & 4L^2 & 0 \\ -13L & 0 & 0 & 0 & -3L^2 & -22L & 0 & 0 & 0 & 4L^2 \end{bmatrix}$$

The elemental stiffness matrix of the system is [5]

$$K = \begin{bmatrix} 12EI/L^3 & 0 & 0 & 0 & 6EI/L^2 & -12EI/L^3 & 0 & 0 & 0 & 6EI/L^2 \\ 0 & 12EI/L^3 & 0 & -6EI/L^2 & 0 & 0 & -12EI/L^3 & 0 & -6EI/L^2 & 0 \\ 0 & 0 & GJ/L & 0 & 0 & 0 & 0 & -GJ/L & 0 & 0 \\ 0 & -6EI/L^2 & 0 & 4EI/L & 0 & 0 & 6EI/L^2 & 0 & 2EI/L & 0 \\ 6EI/L^2 & 0 & 0 & 0 & 4EI/L & -6EI/L^2 & 0 & 0 & 0 & 2EI/L \\ -12EI/L^3 & 0 & 0 & 0 & -6EI/L^2 & 12EI/L^3 & 0 & 0 & 0 & -6EI/L^2 \\ 0 & -12EI/L^3 & 0 & 6EI/L^2 & 0 & 0 & 12EI/L^3 & 0 & 6EI/L^2 & 0 \\ 0 & 0 & -GJ/L & 0 & 0 & 0 & 0 & GJ/L & 0 & 0 \\ 0 & -6EI/L^2 & 0 & 2EI/L & 0 & 0 & 6EI/L^2 & 0 & 4EI/L & 0 \\ 6EI/L^2 & 0 & 0 & 0 & 2EI/L & -6EI/L^2 & 0 & 0 & 0 & 4EI/L \end{bmatrix}$$

The assembled global equation of motion of the system can be obtained as:

$$[M]\ddot{X}(t) + [C]\dot{X}(t) + [K]X(t) = 0 \quad (1)$$

C is the damping matrix obtained by using Rayleigh damping assumption.

The mode shape without considering the effect of universal joint for a straight shaft would be as shown in figure below.

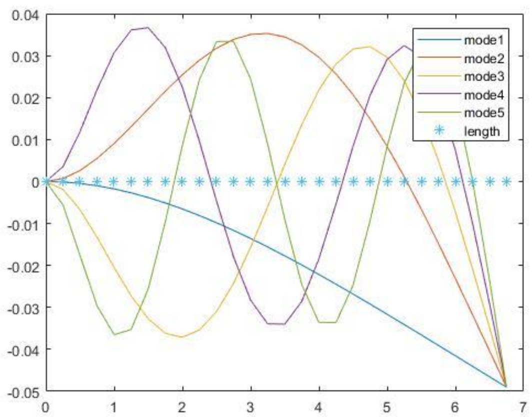

Fig 4: Modeshapes of the system

The addition of the universal joint would constraint the mode at the location of the hooke joint. This is observed in the mode shapes obtained for the shaft system by considering high stiffness at the connection point.

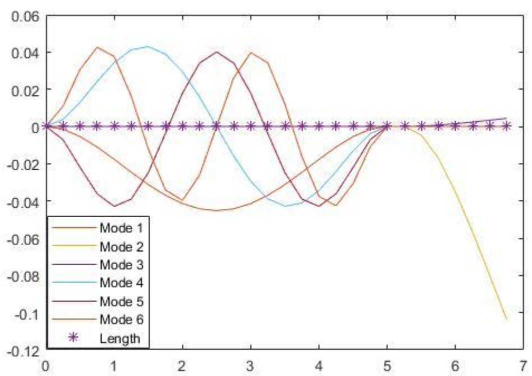

Fig 5: Modeshapes of the system with high stiffness at the joint

2.2 WAVE MODEL

For the present study it is assumed that the drive system moves through the deep water for the cutting operation. The water wave associated with the system acts as the external force acting on it. These forces are derived from the Morison's equation.

The Morison's equation is

$$F(\mathrm{t}) = \tfrac{1}{2}\rho C_d DLU^2 + \rho C_m AL(\dot{U} - \dot{V}) \tag{2}$$

Where the water particle velocity is

$$U = \frac{agk}{\omega}\frac{\cosh k(L+h)}{\cosh kL}\cos(kx - \omega t) \tag{3}$$

And the water particle acceleration is

$$\dot{U} = agk\frac{\cosh k(L+h)}{\cosh kL}\sin(kx - \omega t) \tag{4}$$

The first term of the Morison's equation (2) is called as drag force and the second term is called as inertia force. Thus the equation of motion of the system can be derived as

$$[M]\ddot{X}(\mathrm{t}) + [C]\dot{X}(\mathrm{t}) + [K]X(\mathrm{t}) = F(\mathrm{t}) \tag{5}$$

In case of unsteady flow around the structure an additional effect must be considered resulting from the fluid acting on the system which is called added mass effect when formulating the system equation of motion. This effect is contributed by inertia force contribution from the Morison equation. Therefore the equation of motion can be changed as follows

$$[M]\ddot{X}(\mathrm{t}) + [C]\dot{X}(\mathrm{t}) + [K]X(\mathrm{t}) = F(\mathrm{t}) - [M_a]\ddot{X}(\mathrm{t}) \tag{6}$$

Or $[M + M_a]\ddot{X}(\mathrm{t}) + [C]\dot{X}(\mathrm{t}) + [K]X(\mathrm{t}) = F(\mathrm{t})$ (7)

Where $M_a = \rho\pi D^2 L/4$ (8)

The new global mass matrix and the stiffness matrix is obtained due to the added mass effect of the wave force from the Morison's equation. The new mode shapes obtained for the shaft system due to added mass effect is shown below.

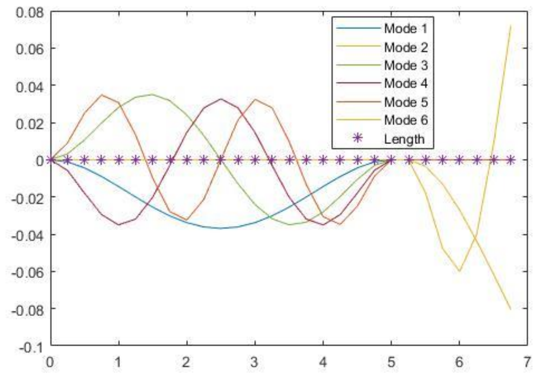

Fig 6: Modeshapes of the system due to added mass effect with high stiffness at the joint

3. RESULT AND DISCUSSION

The length of the longer shaft is 5m and the length of the shorter shaft is 1.75m. The diameter of each of the shaft is 0.2m and the shaft material is taken as mild steel. The rotation of the cutter over the bed surface is considered to be producing the impulsive force which is applied at the end of the cutter drive.

The variation of force due to the wave profile along the depth of the sea is shown below.

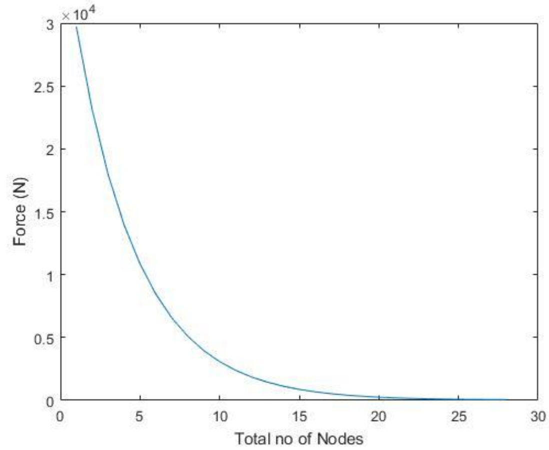

Fig 7: Variation of force along the depth of the sea

This figure gives the information about the variation of the forces along the depth of the sea and it shows that the force is decreasing along the depth. It is maximum at the water level and minimum at the bottom. The velocity profile along the depth of the sea is shown below.

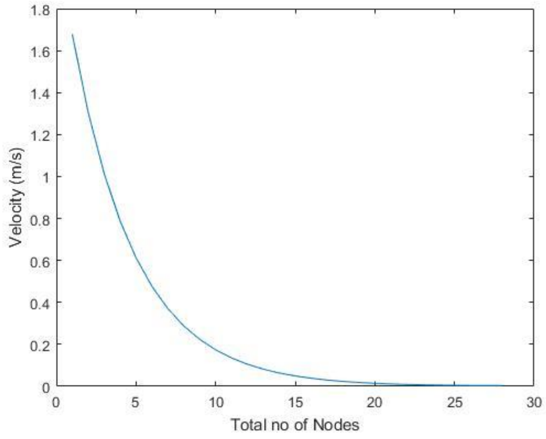

Fig 8: Variation of velocity along the depth of the sea

From this figure it is clear that the velocity also decreasing along the depth of the sea. As the system is fixed at the one end and free at the other end the whole system is considered to be the fixed-free beam type or cantilever type. The initial condition is provided as an initial displacement and initial velocity at the free end.

The time series response without considering the force, over the entire length of the system is calculated. It has been found that due to the high stiffness associated with the universal joint at the connection point of two shafts is highly influencing the response. The time series responses have been plotted along the length over the time and shown below.

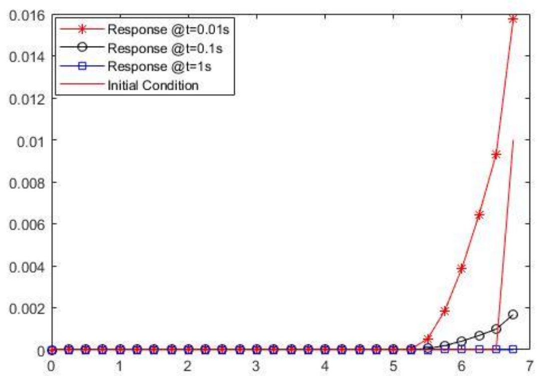

Fig 9: Time series response atdifferent time at t=0.01 sec, t=0.1 sec and t=1 sec

From the above figure it has been found that due to the impulsive force, the response is higher at the end position of the shaft but over the time, the response is getting reduced. The high stiffness of the universal joint reduces the vibration produced due to the impulse at the end of the shaft to be transmitted to the entire length of the shaft.

4. CONCLUSIONS

The dynamics of the shaft system of the cutter drive is studied and designed as two shafts connected by means of a universal joint. The dynamic equation is generated by considering the system as two noded finite element beam element with five degrees of freedom per each node. The wave forces have been calculated over the beam segment and the mode shapes have been plotted for the added mass effects. The time series responses have been determined over the entire length.

This work is dealing with the vibrational effect due to the impact load produced at the extreme end of the shaft system. The result shows that by considering the universal joint as a highly stiff material, it can be capable of reducing the vibrational effect produced at the bottom of the shaft system to the entire length.

5. ACKNOWLEDGEMENTS

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6. REFERENCES

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