TECHNICAL NOTE

HYDRODYNAMICS INDUCED VIBRATION TO TRASH-RACKS

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Abstract In conventional power plants trash-racks are provided at the intakes to protect the turbines. In pumped storage plants, the draft tube or tailrace must also have trash-racks to protect the units while pumping. Because the loads believed to cause many failures of trash-racks are dynamic in nature, it is important to understand the dynamic characteristics of trash-rack structures in general and a single rack in particular. The classical added-mass solution structure-fluid dynamic interaction is known as an approximate solution procedure. An accurate added-mass approach mixed with implementation in finite element framework is proposed. In this proposal, experimental conclusions, supported by theory, led to presentation of more accurate results in vibration of trash-racks. This numerical solution as a powerful method to solve such a complex problem can be employed to carry out dynamic characteristics of these structures while vibrating in water.

Key Words Trash-racks, Natural Frequency, Numerical Solution

چکیده دریچه صافی در نیروگاه‌ها، ورودیهای آب برای حفاظت توربین بکارگرفته می‌شود. در تلمبه خانه‌ها و مجاری پایین دست نیز برای حفاظت از تلمبه‌ها نیز از دریچه صافی استفاده می‌شود. به علت طبیعت دینامیکی نیروهای وارد بر دریچه‌های صافی که منجر به شکست آن می‌شود، بررسی ویژگی‌های دینامیکی عمومی دریچه صافی و همچنین یک تیغه منفرد آن مورد توجه و حائز اهمیت است. روش عمومی جرم افزوده در اندرکنش دینامیکی آب و سازه به عنوان یک راه حل عمومی شناخته شده‌است. در این قالب یک روش دقیقتر به گونه ترکیب در چهارچوب روش اجزا، محدود ارائه شده‌است. براساس این پیشنهاد و استفاده از تجربیات مبتنی بر نظریه مربوطه، نتیجه منجر به ارائه نتایج دقیق‌تری برای حل ارتعاش دریچه صافی شده‌است. این روش حل عددی بصورت یک روش حل قوی برای تحلیل این گونه مسائل پیچیده قابلیت کاربرد داشته و از آن میتوان در تعیین ویژگی‌های دینامیکی این گونه سازه‌ها در شرایط ارتعاش در آب بهره جست.

1. INTRODUCTION

Trash-racks are classified by location, intake or tailrace, and also as fixed or removable. Intake trash-racks are located between the pump-turbine and the upper reservoir while tailrace trash-racks are located between the pump turbine and the lower reservoir. However, existing of any type of trash-rack either against intake flow or in tailrace results in loss of revenue is due to the added head loss. Therefore, the most severe flow conditions for intake trash-racks typically occur during pumping and for tailrace trash-racks, the most severe flow conditions usually exist in the generating mode.

Trash-racks generally consist of flat steel bars with supporting truss or beams, arranged to form a panel for ease of construction and maintenance. These racks are primarily installed at pump or turbine intakes to protect the machinery from the damaging effects of the debris. Therefore, of primary concern is large debris that can damage the rotating parts. The bar spacing is set by requirements of the hydraulic machinery. Spacing of 140 mm is very common for protection against floating debris. However, in special cases while gravel or fish entry must be prevented smaller spacing may be used in the lower panel (in the case of gravel) of the rack. Furthermore, for impulse wheels, careful screening is necessary to prevent debris from choking the nozzles.

Bars are normally, made of different shape such as rectangular, airfoil, and circular. The selection of bar shape and size depends on the structural requirements for static and dynamic loads, climatic conditions and head loss.

The racks because of their importance must be designed to withstand static and dynamic loading encompassing various modes of operation and environmental conditions.

The following factors should be considered in the proper design of trash-racks:

-Velocity distribution
-Head loss caused by the racks
-Dynamic response of the rack to the flow
-Cleaning frequency
-Total weight and cost
-Maintainability
-Blockage by frizzles ice and anchor ice formation

Design of hydraulic structures generally requires a favorable velocity distribution to provide the best performance. A non-uniform flow distribution may cause unbalanced dynamic loading or present a circulation that could lead to the formation of air-entering vortices or localized high velocities that may create excessive vibration of the rack. Any of the stated possibilities may also lead to excessive turbulence that could create not only unacceptable head losses but also damage to trash-rack itself. Sometime, an excessive turbulence may also be provided by frizzle ice formation in cold regions.

To obtain a well uniformly distribution of velocity of flow, the geometry and orientation of racks and also, the type of profile sections used in construction of beams must be carefully investigated. The approach should be designed to have gradual transitions free from sudden change in direction or shape. However, avoiding any irregularity in streamlines, which may be provided through lake geometry, intake position, of culvert bend etc. must be investigated.

For the dynamic design of a structure such as trash-rack, mainly, the dynamic properties of vibrating structure, in contact with fluid, must be considered. The dynamic response of submerged structure due to the dynamic interaction between the structure and the fluid simulated as simple dynamic behavior of a single plate of a simple frame element sustaining a higher mass named as added-mass. Westergaard [1] made the first use of the added-mass concept in vibration of structure. He calculated the hydrodynamics pressure exerted by reservoir on a rigid vibrating single-degree of freedom flat vertical plate. As an alternative to use this hydrodynamics force on the surface as an external load, he suggested adding an equivalent mass of water to the mass of structure. Accordingly, the effects of these forces would produce an inertia force equivalent to the real forces. Jacobson [2] and Housner [3] extended the Westergaard approach to calculate the hydrodynamics pressure and the equivalent add-mass in rigid cylindrical structure.

With the advent of the finite element method and with the help of powerful computers accurate solutions of hydrodynamics effects using the Eulerian and Lagrangian approaches became possible. However, the added-mass solution remained popular as the easiest and fast solution. Niwa & Chopra [4] used Eulerian solution to show that the added-mass obtained from a rigid motion is not an exact presentation of the hydrodynamics effects. Haroun and Housner and Lee etal [5,6] carried out Eulerian-based analytical studies of flexible storage tanks in which they demonstrated that the hydrodynamics pressure distribution for flexible structures is different from that of the same structure when considered rigid. Some experimental investigations carried out by Maheri [7,8] proved these results.

2. FORCES ON TRASHRACKS

The possible forces, which are normally considered to be applied on racks, are due to trash or ice accumulation, drag and head loss or dynamic nature. In the most cases, there is pressure transducers installed at both upstream and downstream of trash-rack position measuring water pressures. As soon as the pressure difference is more than 6m of water, closing butterfly valve and the trash automatically will shut down the turbine or ice must be cleaned from the front of trash-rack.

However, to check the performance of the system, all resisting forces on trash-racks must be estimated and considered in both head loss and design of trash-rack.

While flow passing bluff body, eddies form alternatively on either side of the body and pass into the wake. This fluctuation in the flow pattern induces dynamic forces on the rack. The frequency of vortex shedding by a stationary body is given as Strouhal number as follows:

$$f_f = \frac{SV_0}{d} \tag{1}$$

Where, f_f is forcing frequency, S Strouhal number, V_0 approach velocity in flow path, and d is thickness of body perpendicular to flow. Strouhal number depends on section geometry of rack and is given for a few shapes by Levin [9]. Accordingly, a hydrodynamics cyclic force, which changes upon the stated frequency, is applied on a rack.

In most design cases, the natural frequency of a single rack is compared with the one obtained through Strouhal number avoiding a resonance condition.

However, it is known that loads cause failures of trash-racks are dynamic in nature; it is of importance to investigate the dynamic characteristics of trash-rack structures in general and the racks in particular. For a simple rack, mathematical procedures are used to determine the dynamic response. However, for more complex systems, numerical analyses can easily be employed to solve this vibration problem.

To understand the dynamic characteristics, structures may be idealized as a system of springs, masses and dashpots. Among these aspects, damping is a unknown phenomenon that extracts energy from the vibrating system. Biggs [12] gives an excellent discussion of structural damping and its effects on the dynamic response of structures. A modal analysis will provide structural frequencies and mode shapes, but it will not predict structural damping.

Structural damping is usually assumed to be linearly proportional to velocity and opposite in direction. However, the effective damping is usually expressed as a certain percentage of critical damping, $C_{crit.}$. The critical damping is defined as the amount of damping required eliminating vibration and for a simple one degree of freedom rack is given as follows:

$$C_{crit.} = 2\sqrt{KM} \tag{2}$$

K and M are stiffness and mass of the system. The equation of motion is as follows:

$$M\frac{d^2y}{dt^2} + C\frac{dy}{dt} + ky = F_l \sin\Omega t \tag{3}$$

The solution takes the form

$$y = e^{-\beta t}[c_1 \sin \Omega dt + c_2 \cos \omega dt) + \frac{\frac{F_l}{K}[(1-\frac{\Omega^2}{\omega^2})^2 \sin \omega t - 2(\frac{\vartheta\Omega}{\omega^2}) \cos \Omega t]}{(1-\frac{\Omega^2}{\omega^2})^2 + 4(\frac{\vartheta\Omega}{\omega^2})} \tag{4}$$

where, $\vartheta = C/2M$ and $\omega_d = \sqrt{\omega^2 - \vartheta^2}$ is the natural frequency of the damped system. In the first statement, the first term has the same frequency as the applied force and is called the forced part. The second term has the same frequency as the structural natural frequency and is called the free part. In this case only the forced part that is the second term in Equation 4 is interested. The answer of this portion is written as follows:

$$Y = \frac{\frac{F_l}{K}[(1-\frac{\Omega^2}{\omega^2})^2 \sin\omega t - 2(\frac{\vartheta\Omega}{\omega^2})\cos\Omega t]^{1/2} \sin(\Omega t + \theta)}{(1-\frac{\Omega^2}{\omega^2})^2 + 4(\frac{\vartheta\Omega}{\omega^2})^2} \tag{5}$$

This expression reaches a maximum while $\sin(\Omega t + \theta) = 1.0$, therefore, the dynamic load factor as the ratio of the dynamic deflection to the static deflection that would have resulted if

F_l had been applied as the static load, is obtained as follows:

$$(DLF)_{\max} = \frac{1}{(1-\frac{\Omega^2}{\omega^2})^2 + 4(\frac{\vartheta\Omega}{\omega^2})^2} \tag{6}$$

The value of *(DLF)* versus (ϑ/ω) that is the ratio of actual to critical damping gives the effect of damping in reducing vibrations, particularly at resonance.

3. DYNAMIC RESPONSE OF RACK SUBMERGED IN WATER

While a flat shape structure is to vibrate and submerged in water, fluid structure interaction is known to affect as a damper, though, the natural frequency is less than the free vibration in air. An alternative approach is to evaluate the added mass term instead of considering the damping effects through fluid-structure interaction. Many authors like Nguen [10] and Sell [11] recommended different added mass factors to adjust the vibration of a flat structure in water.

In this research a Lagrangian finite element solution is employed to solve the effect of fluid-structure interaction in vibration of a single rack. To provide numerical model of a single rack and surrounding water the behavior of both rack an water is assumed as linear elastic with corresponding K (bulk modulus) and G (shear modulus). These parameter values for water need some investigation, although, G is not to be taken as zero and ν (Poisson ratio) not to be greater or equal to 0.5. Furthermore, natural frequencies obtained through different G values are severely different!

Three type composition of elements such as steel beam elements in solid waters (type 1), steel shells in solid waters (type 2), and steel solid waters (type 3) may be investigated.

For a plate vibration submerged in water, the general dynamic equation is as follows:

$$(M+A')\overset{\circ\circ}{\delta_i}+(C+B')\overset{\circ}{\delta_i}+(K+C')\delta_i = F_i(t) \tag{7}$$

Where, *M*, *C*, and *K* are mass, damping, and stiffness matrices respectively. *A*, *B*, and *C* are similar to *M*, *C*, and *K*, respectively, but as the effects of submerging the system in water $\overset{\circ\circ}{\delta_i}$, $\overset{\circ}{\delta_i}$, and δ_i are acceleration, velocity and displacement respectively. $F_i(t)$ stands for hydrodynamics force applied to rack *i* which is naturally function of *t* (time).

Accordingly, the natural frequency of a rack in the case of submerging in the water (f_{nw}) is calculated as follows:

$$f_{nw} = \frac{1}{2\pi}\sqrt{\frac{K+C'}{M+A'}} \tag{8}$$

Therefore, assuming f_n as natural frequency of rack in air, the ratio of the two frequencies is calculated as follows:

$$\frac{f_{nw}}{f_n} = \sqrt{\frac{1}{1+(A'_x/m_x)}} \tag{9}$$

where, A_x' and m are added mass because of submergence and mass itself for unit length of rack at nth mode, respectively.

$F_i(t)$ is comprised of a vertex-shedding-induced hydrodynamics force, and a variable Struhal force, which its frequency stated in Equation 1. The drag force can be calculated as follows:

$$F_{d,s} = C_{d,s}\,\ell\,D\left\{\frac{\sin\alpha}{\sin\beta}\right\}^{1/2}\rho\frac{V^2}{2g} \tag{10}$$

where, $C_{d,s}$ depends on the geometry of rack, α and β are flow angle and rack angle respectively. ρ and *V* stand for density and flow velocity, respectively. *D* is thickness of body perpendicular to flow and *l* is the length of body in the flow.

4. TRASHRACK VIBRATION

In general case, flow velocity in between the racks are high, therefore, because of turbulent flow in the

TABLE 1 Natural Frequencies of a Unit Rack in Air.

Element type	First Mode Hertz	Second Mode Hertz	Third Mode Hertz	Forth Mode Hertz
Shell	129.58	203.79	296.58	658.72
Frame	128.87	200.84	290.53	624.06
Solid	131.16	206.85	301.88	332.80

TABLE 2. Natural Frequencies of a Unit Rack in Air.

Method	Major axis (f_w) Hertz	Weak axis (f_w) Hertz
Levin	1464	293
Continues Mass	1451	290
Frame Element	1464	295
Solid Element	1337	304

vicinity of racks, there will be irregular distribution of vortex effects and some contact effects which globally may induce vibration of trash-rack. However, while the frequency of induces vibration matches with the natural frequency of trash-rack structure, there will be a resonance, which leads to severe vibration and thereafter, fatigue effects and failure of trash-rack. As a result, the main characteristic for a trash-rack vibration is known as the relation between flow induced and structural natural frequency of trash-rack. Therefore, to provide a safe condition of trash-rack, these two frequencies must be checked and controlled somehow not to provide resonance.

To control the two named frequencies, mostly, the flow-induced frequency is out of our control. However, engineers in a simple way carry out the natural frequency of a single rack span and compare with that of flow induced one. This method, though is simple and easy to grasp, however, is weak to see the vibration characteristics of whole structural body of trash-rack including boundary conditions.

5. NATURAL FREQUENCIES BY NUMERICAL METHOD

To investigate the main vibration characteristic of a rack as natural frequency, different methods of adopting meshes are mapped. Three types of three dimensional meshes for twice of rack span as frame elements surrounded by solid water elements, shell elements for rack surrounded by solid water elements, and solid elements for both rack and water are considered. The shear modulus of solid elements stand for water is negligible in all cases. To apply the correct boundary conditions, the analyzed rack unit length is comprised of one full rack span extended by half of span at both sides to consider a proper mid span boundary conditions at both ends. Some numerical trial and errors showed that the parameters affecting the natural frequency of taken piece of rack are mainly, mechanical behavior of surrounding water elements and number of taken modes in vibration.

To obtain a base to investigate the change of frequencies, the natural frequencies of a unit rack in air, 1200mm long with 100×20mm cross section is considered. There are two supports at both sides makes a continuous span of 600mm in middle, and proper boundary conditions at both ends. Upon three types of chosen elements, and number of modes, the natural frequencies are computed as stated in Table 1.

Although, the structure of unit rack and boundary conditions are same, the obtained frequencies are slightly different. The main reason

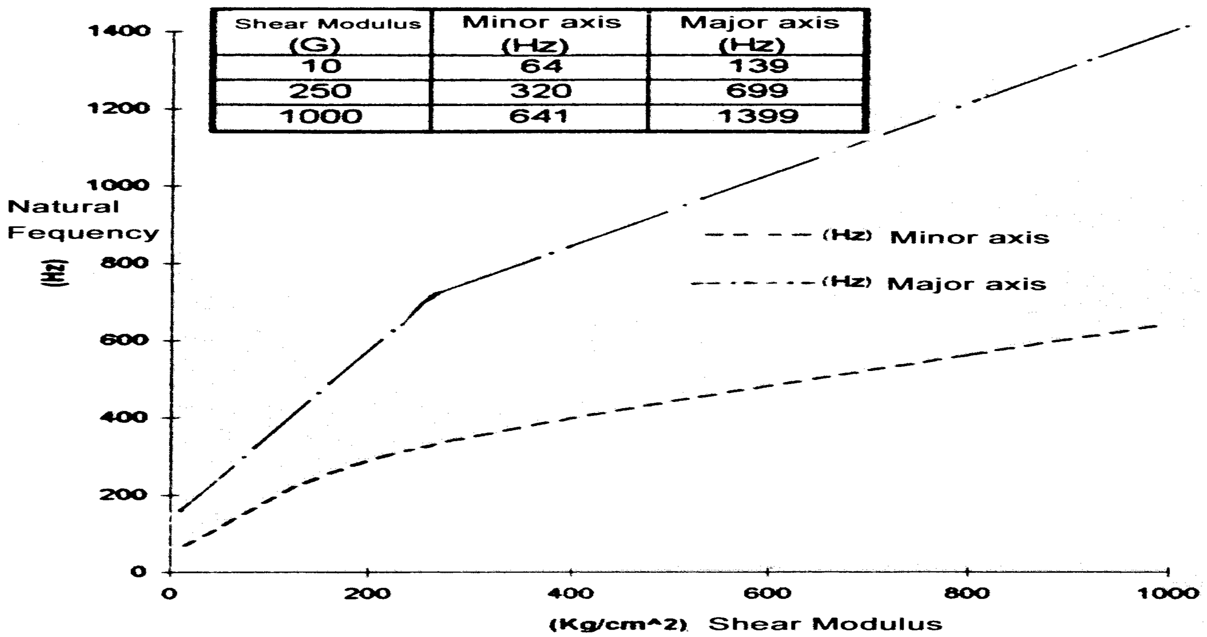


Figure 1. Rack natural frequency in water.

TABLE 3. Natural Frequencies of a Unit Rack in Water.

Effective Sec	**Major axis (f_w) Hertz**	**Weak axis (f_w) Hertz**	**Coefficient Add. Mass**
Full Depth	92.47	462.50	6.25
0.7 Depth	126.10	631.	4.37
0.55 Depth	2140.85	704.5	3.44

is the differences in type of degrees of freedom considered at each node of different elements. However, the modal shapes obtained at all cases are quite similar.

Table 2 shows the results obtained through Levin [9], continues mass, frame element, and three-dimensional solid elements.

To calibrate the results through the small shear modulus of water, the obtained frequencies are compared with one empirical method, and analytical method assuming a continuous mass through the unit.

The natural frequencies of taken unit rack, based of experimental methods, which generally, used in engineering, upon effective depth of cross section are calculated in Table 3. The coefficient of additional mass conforms to submerging the unit in water also presented.

To obtain the values of similar frequency of units in water, the surrounding water elements with negligible shear modulus were included to the existing rack elements. The result of obtaining same natural unit rack frequency in water upon weaker axis of bar section, through all stated methods shows that shear modulus of water must be equal to 8.5 Kg/cm^2. Upon major cross section axis of bar, shear modulus value must be equal to 1000 Kg/cm^2. The natural frequency of rack in water versus change of shear modulus upon two weaker and stronger cross section axes is shown in Figure 1.

TABLE 4. Natural Frequencies of a unit rack in water (Numerical)

Coefficient of Add. Mass	Major axis (f_w) Hertz	Weak axis (f_w) Hertz
3.4	139	624.7

According to Equation 4 the obtained value of additional mass for submerging a rack in water is calculated as 3.3. However, the overall obtained value of this coefficient numerically, is given in Table 4.

6. NUMERICAL RESULTS

To investigate the natural frequency of one complete segment of trash-rack, comparing with the obtained result, a segment of trash-rack, 5800mm wide, 3600mm height, and 700mm depth modeled by three types. The first model is comprised of 9 transversal trusses and the equally distanced racks modeled as frame elements. This model includes 432 nodes and 1263 frame elements which additional mass considered to consider water effects. The truss members are pipe cross sections, 160mm diameter and 13.5mm thickness. Figure 2 shows the mesh of this model. The second model comprised of three transversal trusses to sustain the racks. This model includes

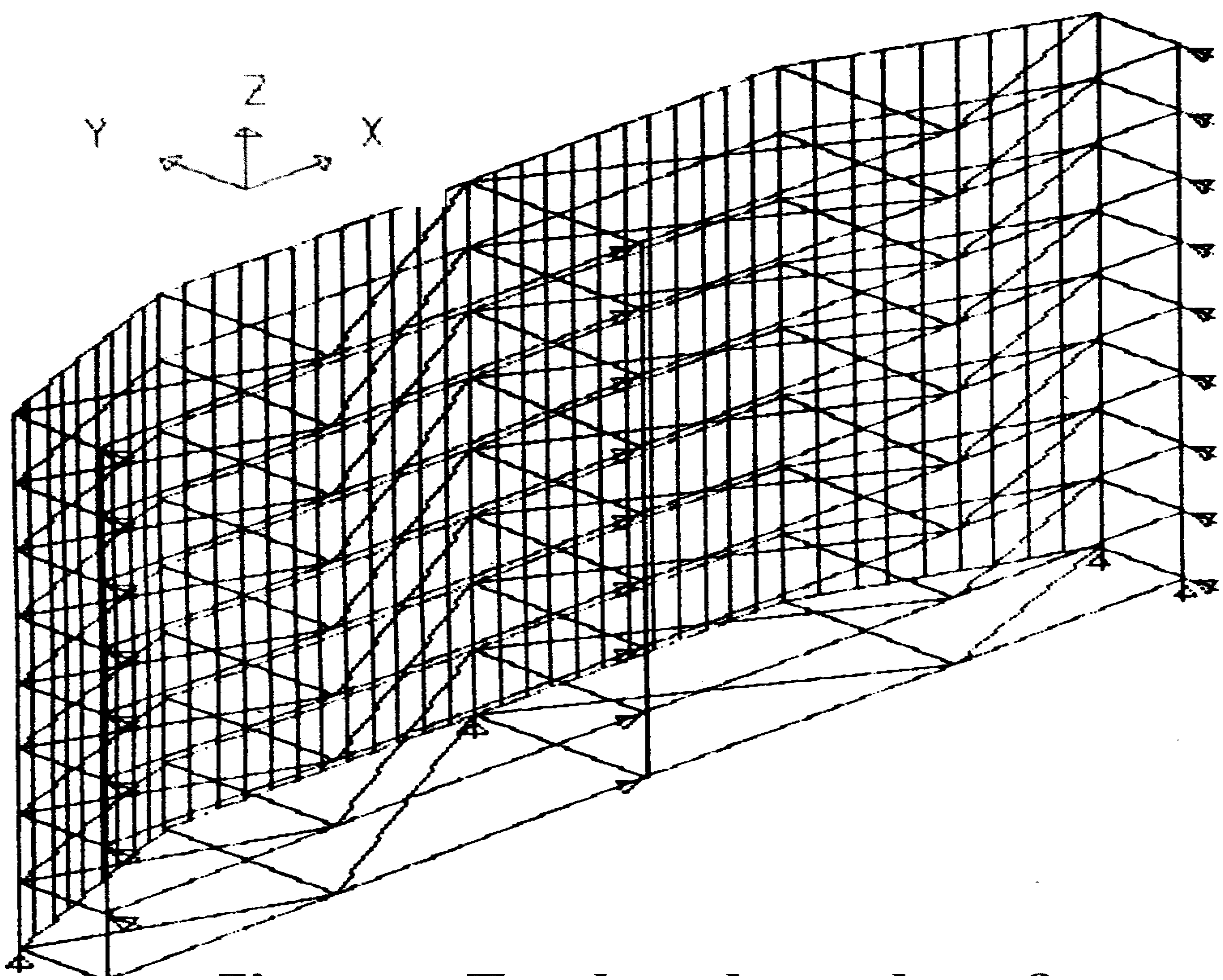


Figure 2. Trashrack mesh as frames (434 nodes, 863 Frame elements).

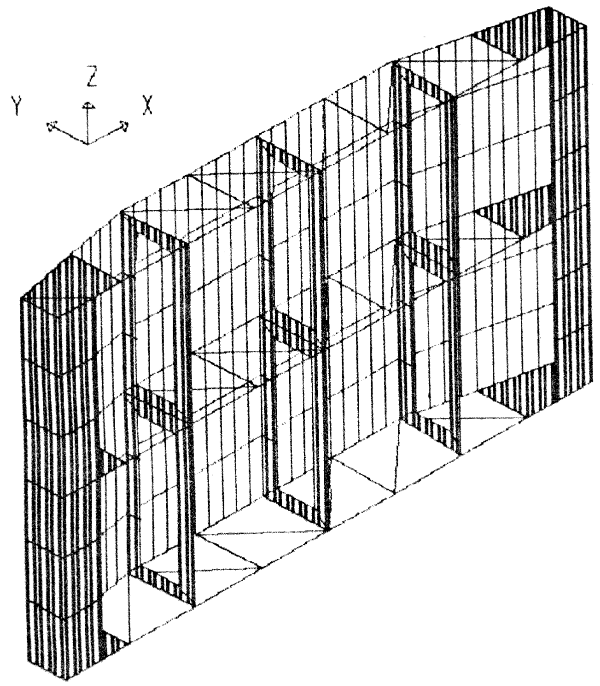


Figure 3. Trashrack Mesh as Frame & Shell Elements (388 Nodes, 613 Frame Elements, 78 Shell Elements, W=6350 kg).

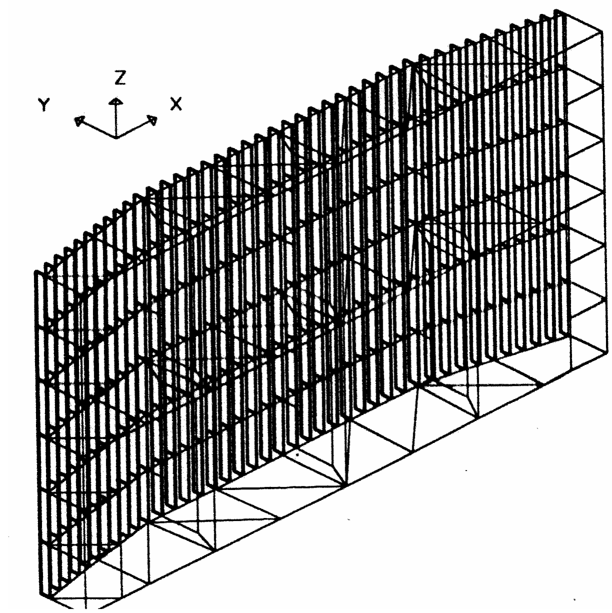


Figure 4. Trashrack Mesh as Solid, Shell, Frame El. (1532 Nodes, 246 Solid, 78 Shell, 654 Frame Element).

388 nodes, 613 frame elements, 78 shell elements that as a whole lead to the solution of 2228 equations. Figure 3 shows the mesh of this model. The third model is similar to the first one, but the racks are modeled by three-dimensional solid elements. This model also includes 1532 nodes, 246 solid elements, 78 shell elements, and 654 frame elements. Figure 4 shows the mesh of this model.

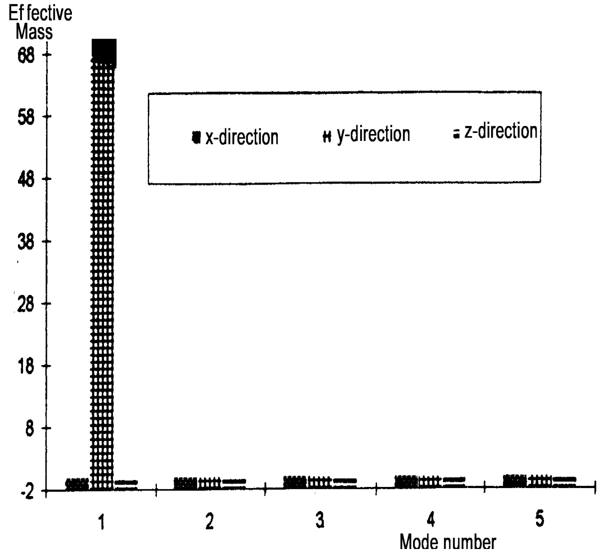
For all three models the additional masses are considered. To conform the boundary conditions of three models with the existing conditions, the bottom nodal supports are restrained in Z and Y directions. To conserve symmetry shape of trash-rack, the nodes on symmetry axis are to be restrained in X direction, therefore, the transversal displacements are distributed symmetrically.

The computed natural frequencies of whole trash-rack segment, in three cases of introduced modeling upon considering first, second, and third mode of vibration, presented in Table 5.

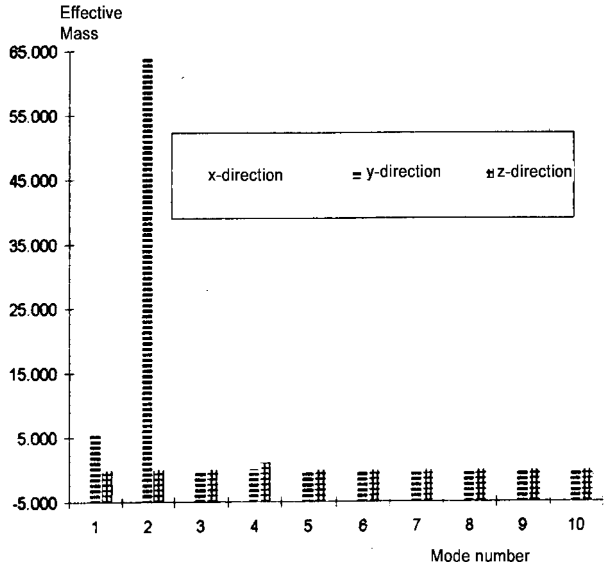
7. THE EFFECTS OF ADDES MASS COEFFICIENT OF DIFFERENT MODES

In the case of using three-dimensional solid elements for racks (i.e. third model) in structure, are calculated for first ten modes and stated in Table 6. The comparison of the similar frequencies was obtained through fist and second models with third, represents that when solid elements are used the stiffness of rack is affected at a higher level. Therefore, the obtained frequency values are more. Accordingly, upon the use of third model the percentage of added mass considered for each mode in three directions are presented in Table 7. Furthermore, it is numerically found that the frequencies are affected while a single frame is under vibration or two or three frames are standing over each other's and vibrate.

Figure 5-a shows the percentage of added mass in three directions for five first modes. These effects while two and three frames are standing over each other's are shown in Figures 5-b and 5-c, respectively.



(a)



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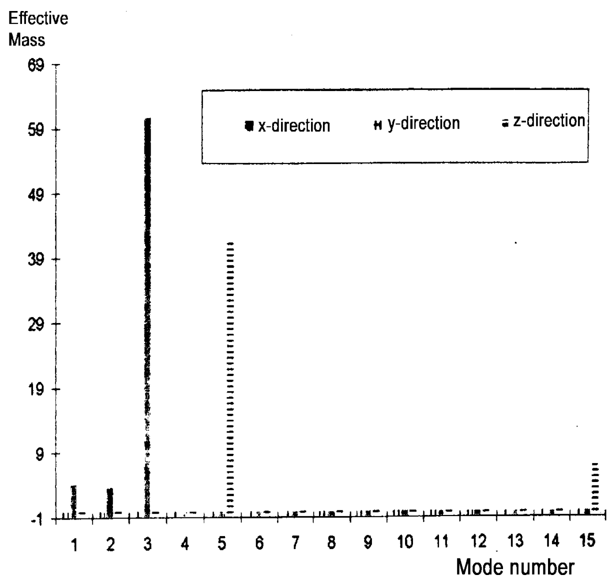

(c)

Figure 5. Effective mass in x, y and z d: (a) one, two and (c) three frame.

8. CONCLUSION

The presented method as a numerical solution can be employed to carry out the intensity of vibration of submerged structure such as trash-rack based on added-mass solution. The comparison of numerical frequencies and Levin method shows the general accuracy of model results.

The proposed method is also an extension of the simple, approximate, traditional added-mass solution for which any general-purpose structural analysis can be utilized. This method was employed to determine the natural frequencies of racks and whole structure of trash-rack to produce a more accurate result.

It is proposed that instead of checking and control of the natural frequency of a single rack with lots ambiguous and dissimilarities, specifically not considering proper boundary conditions, it is simply possible to investigate the whole trash-rack structural body natural frequency. In this way, the effects of dynamic characteristics of other parts of trash-rack are seen in vibration.

The main dynamic characteristic of such a structure as frequency is investigated and the effects of parameters as the type of used elements with different nodal degrees of freedom rack effective length, fluid shear modulus, and over burden pressure by other frames are presented.

To investigate a realistic hydrodynamics result, it is also recommended to apply some multiplying time functions to the applied loads to trash-rack after measuring pulse effects by transducers.

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