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ORIGINAL ARTICLE

VIBRATION INDUCED SPEED DEVIATION OF A CARRIAGE ON TOWING TANK IN SEVERAL RAIL CHAIR DISTANCES

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ABSTRACT: The paper presents an analytical approach to the problem of Vibration Induced Speed Deviation of a Carriage on Towing Tank. Starting from early studies based on a simply supported beam interacting with a lumped mass moving at constant speed, in recent years researchers have improved the models of both the rail and the vehicle. On this basis, the rail is analyzed by FEM to gain rail deflection in some rail chair distances. The analytical method shows that deviation of rail in static and dynamic deflection has little effect in vibration of rail and it can be neglected.
KEYWORDS: Speed Deviation, Carriage, Towing Tank, Rail Chair.

INTRODUCTION

It is well known that a rolling load produces on a bridge or in a girder a greater deflection and greater stresses than the same load acting statically. Such an impact effect of live loads on bridges is of great practical impotance and many engineers have worked on the solution of this problem (Huang, 1976; Matsumoto *et al.,* 1992; Xinfeng *et al.,* 2010; Saiidi *et al.,* 1994; Au *et al.,* 2001; Anagnostopoulos and Gerrard, 1976; Wang *et al.,* 2003). Railway bridges may be intensively excited due to high speed trains. Therefore, a simple theoretical model of a bridge was subjected to a row of axle forces moving along it by Fryba, (2002).

Based on the Lagrangian approach, Cheung considered the vibration of a multi-span non-uniform bridge subjected to a moving vehicle is analyzed by using modified beam vibration functions as the assumed modes (Cheung *et al.,* 1999). Cheng showed that the effect of the bridge structure on the dynamic response of the track structure is considerable (Cheng *et al.,* 2001). Marchesiello presented an analytical approach to the problem of vehicle–bridge dynamic interaction (Chatterjee *et al.,* 1994).

A towing tank is primarily used for model testing of ship (Stefano *et al.,* 1999; De Wilde *et al.,* 2004). One set of model testing concerns the steady test in which the speed of the model must kept constant. Otherwise, the model has accelerations and the accelerations will introduce additional forces such as added mass and the test result is no longer reliable. The non-smoothness of the carriage speed may be initiated from different sources such as electrical system including shane-system, servomotor etc., control system and vibration.

The deviation of speed during the steady movement where the measurement happens is investable. However, the deviation must be kept under the certain limit. The criterion of speed deviation is different where many towing tank are using the 0.2% of the fastest speed as the maximum allowable speed deviation.

One source of speed deviation is the vibration caused by rail. When the carriage wheels pass the rail the combination of static and dynamic forces cause the rail deflection. On this basis, the rail is analyzed by FEM to gain rail deflection in some rail chair distances. The analytical method shows that deviation of rail in static and dynamic deflection has little effect in vibration of rail and it can be neglected.

BASIC MODELING

A rail is sitting on rail chair at certain span, s, supporting carriage weight and dynamic forces. This analysis supposed the material of rail as JIS S45C steel and their properties represented in table 1.

Table 1: Mechanical properties of JIS S45C

Density	7,850 kg/m^3
Yield Strength	343 MPa
Ultimate Strength	569 MPa
Modulus of Elasticity	205 GPa

The model of disc and rail has a periodic motion and it can determine by a cyclic motion as shown in figure 2 and it's amplitude is equal to $\frac{\delta}{2}$.

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$$T = \frac{\lambda}{V} \quad (1)$$

$$\omega = \frac{2\pi}{T} \quad (2)$$

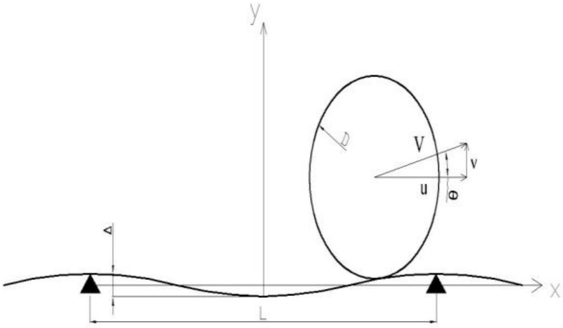

Figure 1: Modeling of rail and wheel

By considering that the speed in horizontal direction must be constant and its deviation from initial value can't be more than 0.2 %, we can achieve to horizontal velocity according to:

$$y = -\delta\cos(\omega os \quad (3)$$

$$\dot{y} = \omega\delta\sin(\omega in \quad (4)$$

$$\ddot{y} = \omega^2\delta\cos(\omega os \quad (5)$$

Also by using θ as a deviation angle we can obtain horizontal velocity as shown in Eq.(6):

$$y = -\delta\cos(kx) \quad (6)$$

$$\theta = \frac{dy}{dx} = k\delta\sin(kx) = \bar{\theta}\sin(kx) \quad (7)$$

$$u = V\cos\theta = V\cos(\bar{\theta}\sin(kx)) \quad (8)$$

Calculations:

$$\lambda = 800\,mm$$

$$\delta = \frac{\Delta}{2} = \frac{0.1033}{2} = 0.05165mm$$

$$V = 19\,\frac{m}{s}$$

$$T = \frac{\lambda}{V} = \frac{0.8}{19} = 0.0421\sec$$

$$\omega = \frac{2\pi}{T} = 149.2256\ \text{rad/sec}$$

$$k = \frac{2\pi}{\lambda} = \frac{2\pi}{0.8} = 7.854\,\text{rad/m}$$

$$\dot{y} = \omega\delta\sin(\omega in = \bar{v}\sin(\omega in$$

$$\bar{v} = 149.2256 \times 0.05165 \times 10^{-3} = 7.7075 \times 10^{-3}\ \frac{m}{s}$$

Then horizontal velocity can be obtained as follow:

$$\theta = k\delta\sin(kx) = \bar{\theta}\sin(kx) \quad (9)$$

$$\bar{\theta} = k\delta = 7.854 \times 0.05165 \times 10^{-3} = 4.0565 \times 10^{-4}\ \text{rad} = 0.002324\,\text{deg}$$

$$u = V\cos\theta = V\cos(\bar{\theta}\sin(kx)) = 19 \times \cos(0.002324 \times \sin(kx))$$

$$\frac{du}{dx} = 0 \Rightarrow Vk\bar{\theta}\cos(kx) \times \sin(\bar{\theta}\sin(kx)) = 0 \Rightarrow kx = \frac{\pi}{2} \Rightarrow x = \frac{\pi}{2k} = 0.2$$

$$u_{min} = u_{x=0.2} = 19 \times \cos(0.002324 \times \sin(\frac{\pi}{2})) = 18.999998\,\frac{m}{s}$$

$$\frac{\Delta u}{u}\% = 8.228 \times 10^{-6}$$

THE CASE STUDY

The current model is a carriage with 4 wheels that moving on a rail. Deflection of rail considers as a primary aim to obtain speed variety. Deflection is based on FEM calculation by commercial software.

3.1. Static Deflection

This investigation supposes that all contact regions with rail as "No Separation" condition. And contact region between rail chair and rail support part as "Bonded" condition.

Carriage weight is 15,000kgf and it is supported by 4 wheels. So, this report suppose that each wheel supports about 3,750kgf of weight, and its load is applied to rail. Standard gravity is applied to all parts, and fixed condition applied at rail chair's bolting hole as shown in figure (2).

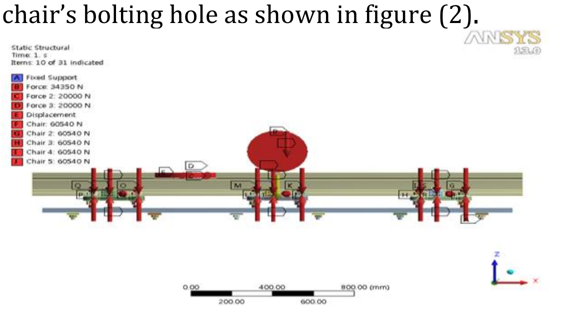

Figure 2: Rail conditions

After analyzing the model with FEM, deflection of rail obtains as shown in figure (3). This deflection has analyzed in static situation.

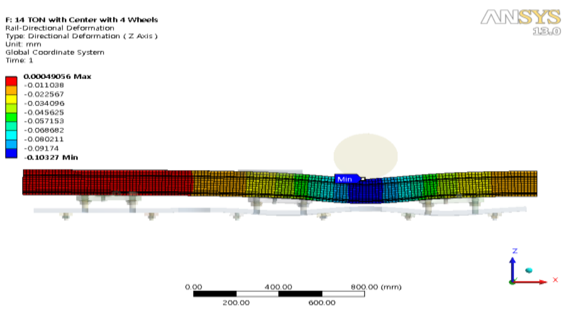

Figure 3: Deflection of rail due to 15kgf

3.2. Dynamic Deflection

By considering that the defection of rail effects on speed of model, dynamic analysis can be performed. When there is a centrifugal acceleration, we can summarize this acceleration with acceleration of gravity and obtain the normal force as follow:

$$\ddot{y} = \omega^2 \delta \cos(\omega os$$

$$\ddot{y}_{max} = 149.225^2 \times 0.05165 = 1.150\ {}^{m}\!/_{s^2}$$

$$F = m(g + \ddot{y}) = 3750 \times (9.81 + 1.15) = 41100\ \text{N}$$

So, Deflection of rail can be obtained by considering corrected force. FEM analysis has been performed and corrected deflection is equal to 0.1232mm. This value define as a Δ'.

By referring to the previous step of calculations and interchanging Δ' instead of Δ, corrected force will be obtained:

$$\ddot{y}_{max} = \omega^2 \delta' = 198.87^2 \times 0.06164 \times 10^{-3} = 2.438\ {}^{m}\!/_{s^2}$$

$$F = m(g + \ddot{y}) = 3750\,(9.81 + 2.438) = 45930\ \text{N}$$

Then we can obtain a final corrected deflection according to corrected force 45930N by FEM analysis. This value is equal to 0.1272mm. Also we can determine speed variety accordance to dynamic deflection as follow:

$$\theta = k\delta \sin(kx) = \bar{\theta} \sin(kx)$$

$$\bar{\theta} = k\delta = 10.47 \times 0.0636 \times 10^{-3} = 6.65892 \times 10^{-4}\ \text{rad} = 0.03815\ \text{deg}$$

$$u = V\cos\theta = V\cos(\bar{\theta}\sin(kx)) = 19 \times \cos(0.03815 \times \sin(kx))$$

$$\frac{du}{dx} = 0 \Rightarrow Vk\bar{\theta}\cos(kx) \times \sin(\bar{\theta}\sin(kx)) = 0 \Rightarrow kx = \frac{\pi}{2} \Rightarrow x = \frac{\pi}{2k} = 0.15$$

$$u_{min} = u_{x=0.15} = 19 \times \cos(0.03815 \times \sin(\frac{\pi}{2})) = 18.999996\,\frac{m}{s}$$

$$\frac{\Delta u}{u}\% = 2.1 \times 10^{-5}$$

DISCUSSIONS AND CONCLUSIONS

In the present study, we deal with speed variety of a carriage having 4 wheels that move on a rail. In this study, two situations were investigated as static and dynamic situations and we reached to appropriate value of speed variety. Also we can propose span of rail chair as a desire variable. So we can show the effect of this item on speed variety. Then we consider a carriage which its weight is 12,000kgf and supported by 6 wheels. So, this case suppose that each wheel supports about 2,000kgf of weight, and its load is applied to rail. Standard gravity is applied to all parts, and fixed condition applied at rail chair's bolting hole. By considering all contact regions with rail as "No Separation" condition and contact region between rail chair and rail support part as "Bonded" condition we can analyze represented model by FEM.

In this section we analyzed 4 cases of rail chair distance. Rail chair distances are 600mm, 800mm, 1,000mm and 1,200mm. Results of FEM analysis represented in figures 4-7.

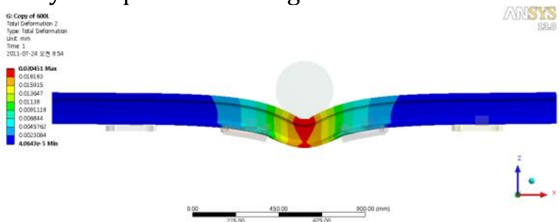

Figure 4: Deflection of rail by span 600mm

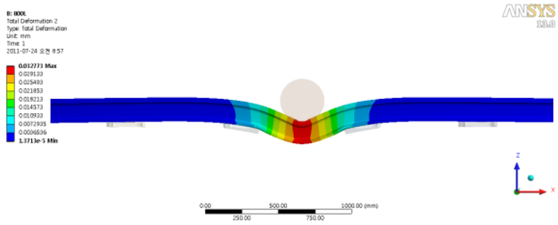

Figure 5: Deflection of rail by span 800mm

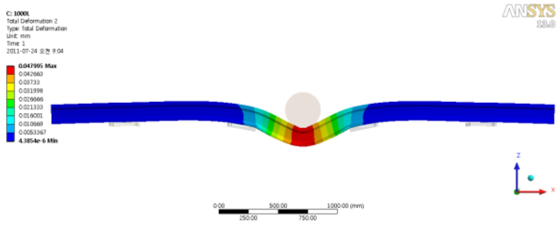

Figure 6: Deflection of rail by span 1000mm

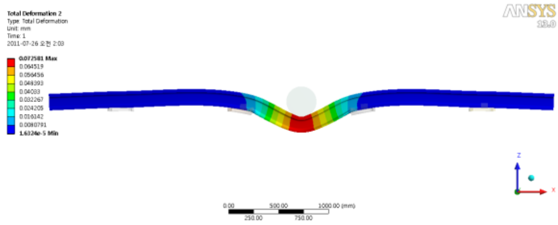

Figure 7: Deflection of rail by span 1200mm

Table 2: Deflection of rail and percent of speed variety

CASE	Max. Deflection(mm)	Δu	$\frac{\Delta u}{u}\% \times 10^{-6}$
1. 600mm	0.0204	18.99999956	2.3
3. 800mm	0.0328	18.9999988	6.3
5. 1,000mm	0.0480	18.9999976	12.6
7. 1,200mm	0.0726	18.9999945	28.9

Results show that deflection increase by increasing in rail chair distance as shown in table (2). Also these deflections effect on horizontal speed that the percent of $\frac{\Delta u}{u}$ represents in table (2).

Also, deviation from acceptable range of variety is so little and we can neglect this variety and consider that direction of velocity only is horizontal.

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