ANALYTICAL STUDY ON THE VIBRATION RESPONSE OF CURVED TRACK SUBJECTED TO MOVING LOAD

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Abstract: A periodical solution on the out-of-plane vibration response of curved track, modelled as periodically supported curved Timoshenko beam, subjected to moving load is determined here. Firstly, the general dynamic response induced by moving load along curved path on an elastic semi-infinite space is obtained on the basis of Duhamel Integral and Dynamic Reciprocity Theorem. Then, in the case of periodic curved track structure, the general dynamic response equation in the frequency domain is simplified in a form of summation within the track sleeper spacing instead of integral. The transfer function of curved track is settled using transfer matrix approach. To verify the validity of the analytical model, the vibration of simple supported curved beam under moving load is obtained and compared with existing reference. Besides, the vibration of curved track of different radii is obtained and compared, indicating that: the response of curved track decreases with the increase of the track radius; the vibration spectrum is abundant and closely related to the load speed.
Keywords: analytical solution, vibration response, curved track, moving load, transfer function

1 Introduction

The planar curved beams, arches and rings have been widely used in machines and structures, such as bridges, aircraft structures and turbo machinery blades, because of their potential applications. The curved track is simplified as periodically supported uniform curved beam here to analysis the vibration of curved track, with the super elevation neglected.

Both the analytical method and the Finite Element Method have been employed in the pioneering study on the out-of-plane vibration of curved beam (Love A. E. H., 1927; Bickford W. B., 1975; Kawakami M., 1995; Yang Y. B., 2001); however, few works have been conducted for the vibration response of curved track subjected to moving load.

A periodical solution on the out-of-plane response of curved track, modeled as periodically supported curved Timoshenko beam, subjected to moving load is considered here. Firstly, the general dynamic response induced by moving load along curved path on an elastic semi-infinite space is obtained on the basis of Duhamel Integral and Dynamic Reciprocity Theorem. In the case of periodic track structure, the general dynamic response equation in the frequency domain is simplified in a form of summation within the track sleeper period instead of integral. The transfer function of curved beam is solved using the transfer matrix approach.

2 Moving Load on the Semi-Infinite Space

Consider the elastic semi-infinite space, shown in Figure 1, subjected to vertical load $g(t)$ moving along a curved path with the radius of R, the initial position of θ_0, the angular speed of c. The vertical time-domain dynamic displacement of receiver ξ can be obtained on the basis of Duhamel integral (Jia Y. X., 2009):

$$u(\xi,t)=\int_{-\infty}^{+\infty} g(\tau)h_z(\xi,\theta(\tau),t-\tau)\mathrm{d}\tau \quad (1)$$

Herein: $u(\xi,t)$ is vertical vibration displacement of receiver ξ, and right hand side of Equation (1) represents a convolution integral of the time history of the moving load $g(t)$ and the vertical transfer function $h_z(\xi,\theta(\tau),t-\tau)$ between the time-dependent load position $\theta(\tau)$ and receiver ξ. Besides, $t-\tau<0, h_z(\xi,\theta(\tau),t-\tau)\equiv 0$.

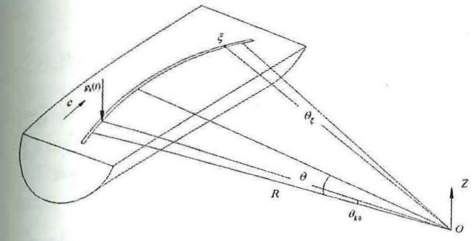

Figure 1 Semi-infinite space subjected to moving load

With the Dynamic Reciprocal Theorem and the Forward Fourier Transform of the time t to the circular frequency ω, the response displacement in the frequency domain can be expressed as:

$$\hat{u}(\xi,\omega)=\int_{-\infty}^{+\infty} g(\tau)\hat{h}_z(\theta_0+c\tau,\xi,\omega)\exp(-i\omega\tau)\mathrm{d}\tau \quad (2)$$

Herein:

$$\hat{h}_z(\theta_0+c\tau,\xi,\omega)=\int_{-\infty}^{+\infty} h_z(\theta_0+c\tau,\xi,t-\tau)\exp(-i\omega(t-\tau))\mathrm{d}t$$

is the transfer function in the frequency domain. And "^" is defined to be the expression in the frequency domain, similarly hereinafter.

3 Moving Load on Track Structure

Consider the periodically supported curved track, half of which is only taken into account, shown in Figure 2, subdivided into an infinite number of track cells with the length of θ_{cell}, which is the sleeper spacing. The track structure is traversed by a vertical load $g(t)$, with the angle speed of c. ξ is the receiver on the rail.

According to the relativity of motion, the load moving forward passing over one cell, equivalents to that the load does not move, while the observation point moves in the opposite direction passing over a cell. Then the dynamic response in frequency domain can be simplified in a form of summation within the track sleeper spacing instead of integral, by transferring the moving of the load on the rail to the moving of pick-up point moving within a specific sleeper spacing, which has been proved by the Floquet Transformation (Jia Y. X., 2009).

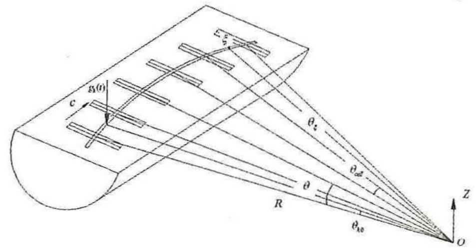

Figure 2 Curved track subjected to moving load

At t, θ is the load position in the global coordinate system: $\theta=\theta_0+ct$, herein, θ_0 is the initial position of the load.

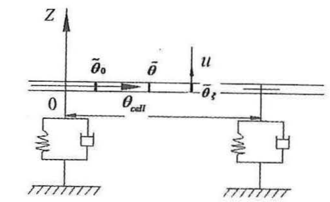

Figure 3 Local coordinate system

The local coordinate system $\tilde{\theta}$ is set up in track basic cell, shown in Figure 3. The relationship between the global coordinate system and the local coordinate system can be expressed as follows:

$$\tilde{\theta}=\theta-n_\theta\theta_{cell},\ \tilde{\theta}_\xi=\theta_\xi-n_\xi\theta_{cell},\ \tilde{\theta}_0=\theta_0-n_0\theta_{cell}$$

Herein: "~" is defined to be the expression in the local coordinate system, similarly hereinafter. n_θ, n_ξ, n_0 are respectively the numbers of basic track cells θ_{cell} between the origin and the load position θ, between the origin and the pick-up point ξ, between the origin and the initial load position θ_0 in the global coordinates. Then:

$$t=(\theta-\theta_0)/c=\tilde{\tau}+(n_\theta-n_0)\theta_{cell}/c \quad (3)$$

$$\tilde{\tau}=(\tilde{\theta}-\tilde{\theta}_0)/c \quad (4)$$

According to Equation (4), when the load moving in one sleeper spacing on the track, the vibration response at receiver ξ can be expressed as:

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$$\hat{u}(\xi,\omega)_{\theta_{cell}} = \int_{-\infty}^{+\infty} g(\tau)\hat{h}_z(\theta_0 + c\tau,\xi,\omega)\exp(-i\omega\tau)\mathrm{d}\tau$$
$$= \int_0^{\frac{\theta_{cell}}{c}} g\left[\tilde{\tau} + \frac{(n_\theta - n_0)\theta_{cell}}{c}\right]\hat{h}_z(\theta_0 + c\tilde{\tau} + (n_\theta - n_0)$$
$$\theta_{cell},\xi,\omega)\exp\left[-i\omega\left(\tilde{\tau} + \frac{(n_\theta - n_0)\theta_{cell}}{c}\right)\right]\mathrm{d}\tilde{\tau} \tag{5}$$

In fact, n_θ changes once when a sleeper spacing is passed over, and n_θ changes time and time again when the load keeps moving on the rail. However, the curve is not infinite long, the angle of the curve is n_t, n_θ would change from n_0 to $n_0 + \frac{\theta_t}{\theta_{cell}}$.

With the help of Equation (4), the expression of the time can be transformation for the expression of space, and then one can get:

$$\hat{u}(\xi,\omega)$$
$$= \sum_{n_\theta = n_0}^{n_0+\theta_t/\theta_{cell}} \frac{1}{c}\int_{\tilde{\theta}_0}^{\tilde{\theta}_0+\theta_{cell}} g\left[\frac{(\tilde{\theta}-\tilde{\theta}_0)+(n_\theta - n_0)\theta_{cell}}{c}\right]\hat{h}_z(\tilde{\theta}+$$
$$n_\theta\theta_{cell},\xi,\omega)\exp\left[-i\omega\left(\frac{(\tilde{\theta}-\tilde{\theta}_0)+(n_\theta - n_0)\theta_{cell}}{c}\right)\right]\mathrm{d}\tilde{\theta} \tag{6}$$

Equation (6) is the dynamic response of the track structure under vertical moving load in the frequency domain.

4 Transfer Function of Curved Track

As referred in reference (Jia Y. X., 2009), the transfer function $\hat{h}_z(\tilde{\theta},\tilde{\theta}_t + (n_{\theta_t} - n_0)\theta_{cell},\omega)$ can be solved as the product of the state variables $S(\tilde{\theta},\omega)$ of the load excitation point and the transfer function of the periodically supported beam, which can be divided into several basic track cells θ_{cell} Besides, the transfer function of basic track cell θ_{cell} can be solved as the product of the transfer function of the curved beam and the support under the curved beam, using transfer matrix approach (Sun J. P., 2009) as follows.

4.1 The transfer matrix of the curved beam

The curved track is simulated as periodically supported planar curved Timoshenko beam; the support under rail is modeled as mass-spring-damper element. For an infinitesimal element of curved beam, shown in Figure 4, with the length measured along the neutral axis of the curved beam denoted by s, the x, y and z axes are taken in tangential directions, radial and transverse directions respectively. The origin of the coordinates moves along the neutral axis of the beam. u is transverse deflection, the slope α due to pure bending an d the angle of torsion φ, the radius is R, θ is the central angle corresponds to the curve element. The cross-section properties and material properties are constant along the beam. The shearing force Q_z, bending moment M_y and the torsion moment M_x are all shown in the Figure 4.

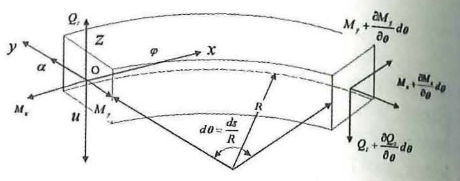


Figure 4 The coordinates of the curved beam element

For the analysis of an infinitesimal element ds in the curved beam, the shear deformation is taken into account, one can get:

$$\alpha = v + \frac{\partial u}{\partial x} - \frac{\varphi x}{R} \tag{7}$$

Herein: v is the transverse shear angle. Twisting angle γ:

$$\gamma = \frac{\partial\varphi}{\partial x} + \frac{1}{R}\frac{\partial u}{\partial x} \tag{8}$$

The force-displacement relationship of the curve beam can be obtained as follows:

$$Q_x = KGAv \tag{9}$$

$$M_y = -EI_y\left(\frac{\partial\alpha}{\partial x} - \frac{\varphi}{R}\right) = EI_y\left(\frac{\varphi}{R} - \frac{\partial\alpha}{\partial x}\right) \tag{10}$$

$$M_x = -EI_s\frac{\partial^2\gamma}{\partial^2 x} + GI_d\left(\frac{\partial\varphi}{\partial x} + \frac{\alpha}{R}\right) \tag{11}$$

$$B_i = EI_s\frac{\partial\gamma}{\partial x} \tag{12}$$

Herein: E is the Young's modulus; G is the shear modulus; K is the shear correction factor; I_y is the vertical bending moment of inertia; I_d is free torsion moment of inertia; I_s is polar moment of the cross-section; A is the sectional area.

Considering the homogeneous beam with infinite degrees of freedom, the dynamic equilibrium equations of the infinitesimal element of curved beam can be obtained, according to its equilibrium condition.

$$\frac{\partial Q_z}{\partial x} = \rho A\frac{\partial^2 u}{\partial t^2} \tag{13}$$

$$\frac{\partial M_y}{\partial x} = \rho I_y\frac{\partial^2\alpha}{\partial t^2} + Q_x - \frac{M_x}{R} \tag{14}$$

$$\frac{\partial M_x}{\partial x} = \rho I_s\frac{\partial^2\varphi}{\partial t^2} + \frac{M_y}{R} \tag{15}$$

$$\frac{\partial M_x}{\partial x} = \rho I_s\frac{\partial^2\varphi}{\partial t^2} + \frac{M_y}{R} \tag{16}$$

Herein: The shearing force is Q_z, bending moment is M_y and torsional moment is M_x, Double warping moment is B_i, warping angle is γ, ρ is the mass per unit volume.

The state vector of any point in the curve beam can be expressed as:

$$S = \{Q_z, M_y, M_x, B_i, u, \alpha, \varphi, \gamma\}^{\mathrm{T}}$$

Equations (7~16) can be expressed using Matrixes:

$$\frac{\partial S}{\partial x} = AS \tag{17}$$

Herein:

$$A = \begin{bmatrix} 0 & 0 & 0 & 0 & -\rho A\omega^2 & 0 & 0 & 0 \\ 1 & 0 & -\frac{1}{R} & 0 & 0 & -\rho I_y\omega^2 & 0 & 0 \\ 0 & \frac{1}{R} & 0 & 0 & 0 & 0 & -\rho I_s\omega^2 & 0 \\ 0 & 0 & -1 & 0 & 0 & 0 & 0 & GI_d \\ -\frac{1}{KGA} & 0 & 0 & 0 & 0 & 1 & \frac{x}{R} & 0 \\ 0 & -\frac{1}{EI_y} & 0 & 0 & 0 & 0 & \frac{1}{R} & 0 \\ \frac{1}{RKGA} & 0 & 0 & 0 & 0 & -\frac{1}{R} & -\frac{x}{R^2} & 1 \\ 0 & 0 & 0 & \frac{1}{EI_s} & 0 & 0 & 0 & 0 \end{bmatrix}$$

The general solution of Equation (17) can be settled as:

$$S(x) = \mathrm{e}^{Ax}S_0 \tag{18}$$

Herein: S_0 is a constant matrix in the solution. The curved beam can be divided into many infinitesimal elements, with the length of Δx, and then one can get:

$$x_k = k\Delta x \quad (k = 1,2,3...) \tag{19}$$

$$x_{k+1} = x_k + \Delta x \tag{20}$$

Then:

$$S(x_{k+1}) = T_i(\Delta x)S(x_k) \tag{21}$$

Herein: $T_i(\Delta x) = \mathrm{e}^{A\Delta x}$.

Based on the precise integration method of the exponential matrix (Sun J. P., 2009):

$$T_i(\Delta x) = \mathrm{e}^{A\Delta x} = \left(\mathrm{e}^{\frac{A\Delta x}{2^N}}\right)^{2^N} = (\mathrm{e}^{A\tau})^{2^N} \tag{22}$$

Herein: $\tau = \Delta x / 2^N$, N=20.

4.2 The transfer matrix of the support

For the periodically supported track structure, the periodic support is simulated as double-layer mass-spring-damper system, in which rail pad and sleeper pad are both modeled as spring-damper element, the sleeper is modeled as concentrate mass between the rail pad and sleeper pad. The double-layer support is calculated as a spring-damper element, as shown in Figure 5, of which the composite stiffness k_v can be expressed as:

$$k_v = \frac{ck_r\cdot(ck_{sb} - M_s\omega^2)}{ck_r + (ck_{sb} - M_s\omega^2)} \tag{23}$$

Herein: k_r, k_s, k_b are respectively the stiffness of rail pad, sleeper pad and subgrade; c_r, c_s, c_b, are respectively the damping of rail pad, sleeper pad, and subgrade , M_s is the sleeper mass.

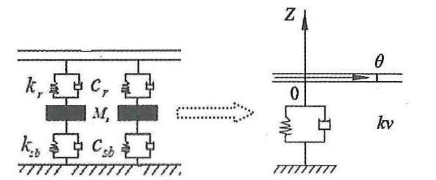


Figure 5 The spring-damper element under the curved beam

Consider an infinitesimal element on the support of the curved beam, shown in Figure 5, the state vectors of the two sides of the infinitesimal element are defined as follows:

The left side:

$$S_i^L = \{Q_{iz}^L, M_{iy}^L, M_{ix}^L, B_i^L, u_i^L, \alpha_i^L, \varphi_i^L, \gamma_i^L\}^{\mathrm{T}}$$

The left side:

$$S_i^R = \{Q_{iz}^R, M_{iy}^R, M_{ix}^R, B_i^R, u_i^R, \alpha_i^R, \varphi_i^R, \gamma_i^R\}^{\mathrm{T}}$$

Then one can get:

$$u_i^R = u_i^L, \alpha_i^R = \alpha_i^L, \varphi_i^R = \varphi_i^L, \gamma_i^R = \gamma_i^L$$
$$Q_{iz}^R = Q_{iz}^L - k_{sleeper}(\omega)u_i, M_{iy}^R = M_{iy}^L, M_{iz}^R \tag{24}$$
$$= M_{iz}^L, B_i^R = B_i^L$$

$k_{sleeper}$ is the composite stiffness of the sleeper, which is simplified as spring-damper element.

Equation (24) can be expressed as:

$$S_i^R = T_{\sup p} S_i^L \tag{25}$$

Herein:

$$T_{\sup p} = \begin{bmatrix} 1 & 0 & 0 & 0 & -k_{\text{sleeper}}(\omega) & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 1 \end{bmatrix}$$

4.3 Initial state vector of the curved beam under unit load

(1) Unit load between two sleepers

The state vectors of the double sides of the curve beam element are defined to be S^L, S^R for the left side and for the right side, as shown in Figure6.

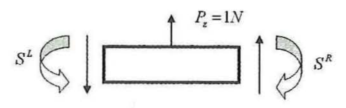


Figure 6 Mechanical analysis of the beam element

According to the transfer matrix, one can get:

$$S^R = T_i(\Delta x)S^L \tag{26}$$

$$S^R - S^L = P \tag{27}$$

(2) Unit load on the sleeper

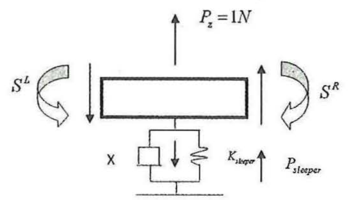


Figure 7 Mechanical analysis of the beam element with support

$$-Q_z^L + Q_z^R + 1 = P_{sleeper} = X \cdot k_{sleeper}(\omega) \tag{28}$$

$$S^R = T_i(\Delta x)S^L \tag{29}$$

Herein: $T_i(\Delta x) = T_i(\Delta x/2)^L T_{\sup p} T_i(\Delta x/2)^R$

Then the state vector can be settled.

With the initial state variables and the transfer function of the curved beam settled, the dynamic response of the periodically supported curved track structure under the moving load could be solved.

5 Calculation Examples

According to the theories above, the calculation program is formed.

5.1 Model validation

As a special case of the analytical model presented above, Yang Y. B. (2001) has examined the vibration of simple supported curved beam subjected to moving load, as shown in Figure 8. To verify the validity of the model in this paper, the example is recalculated here. The curved beam was simply supported, the given data is: a=5 m, b=1.8 m, $\bar{\alpha}=30°=\pi/6$, R=45.84 m, length L=24 m, E=32.3×10^9 N/m^2, ν=0.2, G=E/[2(1+ν)], $k'=0.833$, $I_x = ab^3/12 = 2.43$ m^4, $I_y = ba^3/12 = 18.75$ m^4, $J_\theta = I_x + I_y = 21.18$ m^4, $A = ab = 9$ m^2, $V_p = 40$ m/s, $P = 9.8\times 29.9\times 10^3$ N, and damping $\zeta_d = 0$.

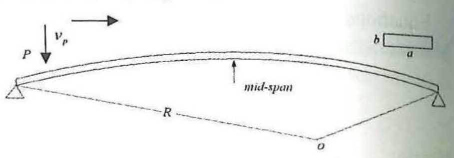


Figure 8 Simple supported curved beam

The mid-span vibration displacement of the simple supported curved beam under moving load is obtained, shown in Figure 9.

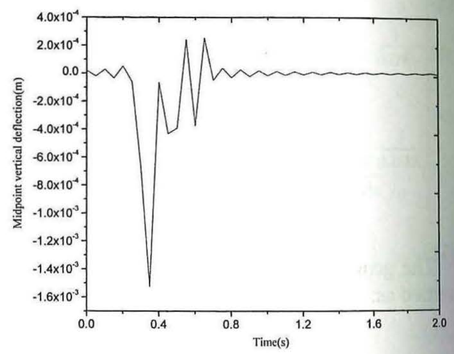


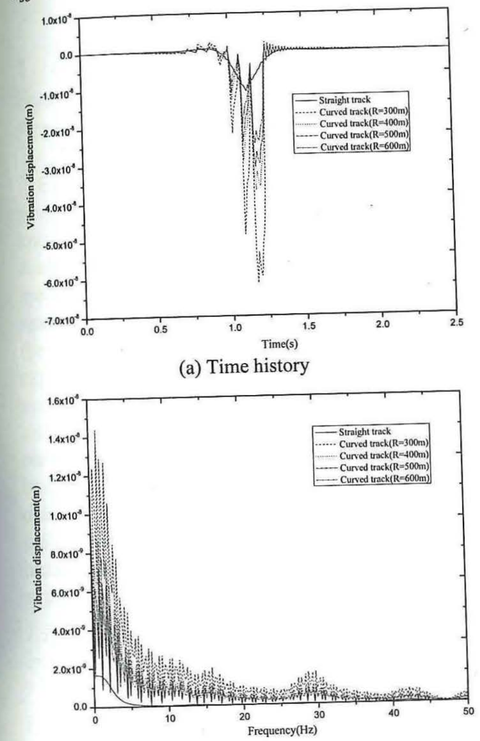
Figure 9 Displacement response of the simple supported beam

The calculated displacement response here coincides well with the example given by Yang Y. B. (2001), which confirms the reliability of the presented theories.

5.2 Vibration response of curved track

Consider the curved track structure, subjected to moving force $g(t) = 1$ N, with constant speed of v = 300 km/h, the vibration receiver ξ is located at 9.3m away from the initial position of the moving load, shown in Figure 2. The vibration of curved track of different radii (R=300 m, 400 m, 500 m, 600 m, ∞) under moving load is obtained and compared, as shown in Figure 10, and the vibration in 0~50 Hz is only taken into account.

The parameters are as follows: Rail mass per unit length: m_r=60 kg/m, Elastic Modulus: E=210 GPa, Cross Section Area: A=7.60×10^{-3} m^2, Cross section inertia moments: I=3.04 ×10^{-5} m^4, Damping Ratio: ξ_r=0.01, sleeper mass per unit length: m_s =50 kg/m, sleeper spacing L_{cell}= 0.60 m, bed mass per unit length: m_b=260 kg/m, and the mass of sleeper and bed are taken into account together. The fastener employed here is DTVI$_2$ fastener, of which the stiffness and damping parameters are k_r=78 MN/m, c_r=5.0×10^4 N·s/m respectively. The stiffness and damping parameters of the sleeper pad are k_{sb}=100 MN/m, c_{sb}=5.0×10^4 N·s/m respectively.



(a) Time history

(b) Frequency spectrum

Figure 10 The vibration displacement of the receiver

With the comparison above, we can see that:

Under the same moving load, the vibration of curved track is bigger than that of straight track; the vibration spectrum of curved track is more abundant. The response of curved track decreases with the increase of the track radius.

The peak values of the vibration spectrum appear around 14 Hz, 28 Hz and 42 Hz. And the time of the load traveling in the sleeper spacing is t = 0.072 s, and f = 1/t = 13.9 Hz, which coincides with the frequency point of the peak spectrum. The vibration spectrum is closely related to the load speed.

6 Conclusions

A periodical solution on the out-of-plane vibration response of curved track, modelled as periodically supported curved Timoshenko beam, subjected to moving load is determined here.

The vibration of simple supported curved beam under moving load was obtained and compared with existing results to verify the validity of the presented theories.

Under the same moving load, the response of the track decreases with the increase of the track radius. The vibration spectrum is closely related to the load speeds, besides the response spectrum of curved track is more abundant.

Acknowledgement

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APPLICATION OF EXPERT SYSTEM OF KARST HAZARD EVALUATION PREDICTION ON MOUNTAIN TUNNEL CONSTRUCTION

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Abstract: This paper analyzed the acquisition of hazard evaluation prediction knowledge in the expert system of karst hazard evaluation prediction as well as the conditions and computation in obtaining rule parameters, and carried out a hazard evaluation prediction to a tunnel karst hazard case in Yiwan Railway by applying the expert system, concluding the advantages and disadvantages of the expert system in predicting karst hazard evaluation, which can provide reference for other similar engineering hazard evaluation.
Keywords: tunnel, karst hazard, hazard evaluation, expert system

1 Introduction to the Expert System of Karst Hazard Evaluation Prediction

Expert system refers to an intelligent programming system equipped with specialized knowledge and experience, which can simulate the thinking process of experts with the experiences and specialized knowledge accumulated by experts for many years to solve complicated problems within the field that can only be solved by experts. The expert system of karst hazard evaluation prediction consists of five parts, including the inference machine for karst hazard evaluation prediction (consulting device, interpreter), knowledge base of karst hazard evaluation prediction, integrated database of the system, back-stage management of the expert system as well as the user interface of hazard evaluation prediction. Figure 1 is the structure of the expert system.

1.1 Knowledge acquisition of karst hazard evaluation prediction

The knowledge acquisition of karst hazard evaluation prediction includes the following aspects:

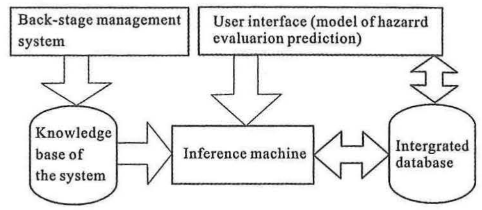

Figure 1 Structure of the expert system of karst hazard evaluation prediction

(1) Single stratum

Based on the lithology, it is divided into strong karstified limestone of pure nature, middle karstified dolomite, weak karstified mud dolomite and marlstone, and petrofabric composed of dissolubility rock and non-dissolubility rock (based on the circumstances).

(2) Geologic structure

According to the fold morphology, it is divided into syncline composed of single lithostrome, anticline composed of single lithostrome, syncline composed of petrofabric, and anticline composed of petrofabric; longitudinal fault, cross fault and oblique fault are divided according to the relationship between fault strike and fold-axis as well as topographical divide.

(3) Topography and geomorphology

In view of the combination amount of calcipit, trough valley and ponor and funnel on the

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