重载机车车轮不圆演变规律及动态影响

刘鹏飞^{1*}, 王天龙¹, 吕凯凯², 曹云强¹, 张凯龙¹, 宋婧婧¹

1. 石家庄铁道大学, 省部共建交通工程结构力学行为与系统安全国家重点实验室, 石家庄 050043;

2. 西南交通大学, 牵引动力国家重点实验室, 成都 610031

* 联系人, E-mail: lpfswjtu@163.com

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摘要 针对六轴重载机车存在的车轮不圆顺伤损问题, 开展了一个镟轮周期内的不圆顺现场跟踪测试, 分析了车轮不圆顺随运行里程的演变规律及谐波分布特征. 建立了重载机车-轨道纵垂耦合动力学模型, 以不圆顺作为激励输入, 研究了不圆顺演变对机车轮轨动态相互作用的影响. 结果表明, 三轴转向架端部轮对不圆顺发展速率比中间轮对更快, 运营至 17×10^4 km时, 车轮径向偏差幅值最大接近0.9 mm, 1~3, 6~8阶谐波是车轮不圆顺的主要构成. 不圆顺极易引起较大的轮轨垂向冲击振动和轮重减载现象, 诱发轮轨高应力接触状态, 不利于机车牵引性能的稳定发挥, 尤以7阶谐波影响最为显著. 运营里程超过 10×10^4 km时, 建议进行车轮不圆顺检修.

关键词 重载机车, 车轮不圆顺, 耦合动力学, 轮轨接触

重载铁路是铁路货物运输的重要组成部分, 其中, 大功率重载电力机车是当前我国重载列车的主要牵引力量. 相较于普通车辆, 重载机车轮轨系统服役环境更为恶劣, 轮轨界面除发挥蠕滑导向作用外, 还需在有限的接触面积上传递较大的牵引力及电制动力. 车轮踏面在长期运行过程中无疑会产生不均匀磨耗及剥离掉块等问题, 而车轮不圆顺即是踏面圆周向不均匀磨耗的主要表现. 上述问题的存在严重缩短了镟轮周期, 如HXD1型重载机车一度将镟修里程降至 7.6×10^4 km, 最长也不超过 13×10^4 km^[1], 而神朔铁路SS4G型机车镟轮周期在 $10 \times 10^4 \sim 18 \times 10^4$ km, 对于轮箍运行 10×10^4 km到限的机车, 磨耗极易引起异常振动^[2]. 车轮不圆顺是铁路车轮伤损的一种普遍形式, 国内外科研人员针对这一现象开展了大量研究. Ferrara等人^[3]提出了一种考虑轨道缺陷和车轮不圆顺的预测列车引起振动的数值模型, 分析了轮轨之间的动态相互作用. Johansson和An-

dersson^[4]采用列车-轨道相互作用模型, 研究了车轮不圆顺的长期演变行为. 李大地和戴焕云^[5]基于线路动力学试验数据, 分析了具有高阶规则车轮不圆顺磨耗车辆转向架各部件振动水平及振动传递关系. 陈伟等人^[6]通过数值仿真研究列车高速运营状态下车轮不圆顺化的波深、谐波阶数对车辆动力学性能的影响. 陈光雄等人^[7]通过轮对-钢轨-轨枕弹簧有限元振动模型研究发现, 当正确选择轨枕支撑弹簧刚度或当控制轮轨摩擦系数不大于0.27时, 可抑制甚至消除车轮不圆顺磨耗. 宋志坤等人^[8]研究车轮不圆顺对车辆振动及轮轨力的影响并提出不同阶次的车轮不圆顺限值. 刘道远^[9]研究了地铁车轮不圆顺激励下的轮轨耦合振动问题. 从现阶段研究成果看, 研究对象主要集中在高速列车和地铁车辆, 而针对重载机车车轮不圆顺演化规律和动态影响问题的研究尚未深入, 特别是综合考虑机车的轮轴牵引力. 基于当前正常服役中的重载机车,

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开展车轮不圆顺的现场测试及其振动影响研究,对于未来发展更大轴重机车、合理制定机车车轮养护维修规程具有重要意义。

为此,本文针对25 t轴重六轴大功率重载机车,通过现场测试,获取不同运营里程下的车轮不圆顺数据,研究不圆顺的谐波组成及演变规律。在此基础上,建立机车-轨道纵垂耦合动力学模型,分析车轮不圆顺激励下的轮轨动态相互作用力及法向、切向接触应力,从而探明车轮不圆顺演化对重载机车轮轨动态相互作用的影响。

1 车轮不圆顺现场测试及几何特征分析

1.1 现场测试

课题组于2017年先后3次前往现场进行数据测量,测试现场照片如图1所示。测试对象为25 t轴重重载电力机车,轴式C0-C0。测试仪器采用m|wheel型车轮粗糙度测量仪,仪器采样间距1 mm,机车车轮直径1.25 m。不圆顺测点位于车轮名义滚动圆处,即距轮背70 mm处。

为便于比较,本文针对同一台机车进行跟踪测量,测试了一个镟修周期内不同运营里程下的车轮不圆顺发展情况,测得的各车轮径向偏差幅值(即车轮实际半径和车轮名义滚动半径之差)统计结果如图2所示。图中*i*L, *i*R分别表示第*i*位轮对左、右车轮, *i*=1~6。从图2中可以看出,随着运行里程地增加,车轮径向偏差幅值逐渐增大,运行至 17×10^4 km时,最大值达到了0.867 mm,同时也反映了车轮磨损发展较快。目前,我国针对重载机车尚未形成统一的径向偏差限值标准,参考德国铁路规定的车轮径向偏差幅值不应超过0.6 mm的标准^[10],显然现场测得的最大值已超过这一标准。运营里程达到17万km时,全车12个车轮中有6个超过0.6 mm,运营里程同样也超过了HXD1机车的镟轮周期^[1]。从量值演变看,机车运营里程从 4.3×10^4 km增至 17×10^4 km时,各车轮的径向偏差幅值均逐渐增大,只是增幅不同,5L车轮对应的径向偏差幅值增幅最小,为0.1 mm,6R车轮对应值增幅最大,达到了0.695 mm。

此外,测试结果也进一步反映了机车的实际运行状态,如二、五位轮对的服役情况要优于其他轮对,这是由于机车均要往返运行,通过曲线线路时前、后转向架发挥导向作用的主要是机车的一和三、四和六位轮对,轮轨作用剧烈,而二、五位轮对均属转向架中间轮对,运行状态较好。综合全部车轮, 4.3×10^4 、 10×10^4 和

图1 (网络版彩色)现场测试场景
Figure 1 (Color online) Field test scene

图2 (网络版彩色)不同服役里程的车轮径向偏差幅值对比
Figure 2 (Color online) Comparison of wheel radial deviation amplitudes for different service distances

17×10^4 km对应的车轮径向偏差幅值平均值为0.18, 0.28和0.49 mm。

1.2 车轮不圆顺几何特征分析

由于6R车轮处径向跳动最为明显,为此着重对该车轮的不圆顺数据进一步分析,以获取车轮径向几何特征及其变化过程。图3比较了 4.3×10^4 、 10×10^4 和 17×10^4 km服役里程对应的车轮径向偏差幅值和1~20阶谐波阶次分布特征。通过傅里叶变换分析得到图3(b),图中显示,随着里程数的增加,1~3, 6~8阶次谐波增幅最为明显,其中,1阶谐波变化尤为显著,2阶谐波次之,6, 7, 8阶谐波基本相当。例如,运行里程由 4.3×10^4 km增至 17×10^4 km,不圆顺的主要谐波成分变为1, 2, 3, 7阶,而1阶谐波增幅达到了0.2 mm,表明车轮圆周向长波缺陷的发展速率较快。

图3 (网络版彩色)极坐标图(a)和谐波阶次分布(b)
Figure 3 (Color online) Polar diagram (a) and harmonic order distributions (b)

值得注意的是, 机车运营里程超过 10×10^4 km后, 开始出现径向偏差幅值超过0.6 mm的车轮, 建议在此时进行车轮不圆检修. 此外, 所测重载机车车轮不圆主要成分大多出现在1~10阶, 主导谐波按出现频率递减排序为1, 2, 3, 5, 7阶, 对应的主波波长集中在556~3890 mm范围内, 当机车以60~80 km/h速度运行时, 可激发频率范围4~40 Hz的垂向振动.

2 考虑电机驱动的机车-轨道耦合动力学模型

为研究车轮不圆顺对重载机车车轮轨动态相互作用的影响, 在机车-轨道垂向统一模型^[11]的基础上, 进一

步考虑系统纵向运动, 建立了机车-轨道纵垂耦合动力学模型, 如图4所示, 图中相关变量含义可参见文献[12]. 模型中, 车体、构架和轮对均考虑了沉浮、纵向和点头运动, 由于电机采用抱轴式悬挂, 因此牵引电机简化为仅绕轮轴具有点头运动的刚体, 其另一端通过电机吊杆吊挂于构架上. 整车共计33个自由度. 轨道系统中只考虑钢轨、轨枕和道床的垂向振动.

车体、构架及轨道结构的振动方程可根据文献[11,12]得到, 而牵引电机由于采用抱轴悬挂, 电机与轮对间存在较复杂的耦合振动^[13,14]. 鉴于此, 本文仅给出轮对、电机的耦合振动方程及其解耦过程. 式(1)~(4)分别为轮对的纵向、垂向、点头振动方程和牵引电机

图4 机车-轨道纵垂耦合动力学模型
Figure 4 Vehicle-track longitudinal and vertical coupled dynamic model

的点头运动方程:

$$M_w \ddot{X}_{wi} - M_m h_{mw} \ddot{\beta}_{mi} = F_{cx} + F_{xf} + F_{mx}, \quad (1)$$

$$(M_w + M_m) \ddot{Z}_{wi} - (-1)^n M_m l_1 \ddot{\beta}_{mi} = -P_i + F_{mzi} + F_{zi} + (M_w + M_m)g, \quad (2)$$

$$I_{wy} \ddot{\beta}_{wi} = r_i F_{cx} - M_q, \quad (3)$$

$$(I_{my} + M_m l_1^2) \ddot{\beta}_{mi} - (-1)^n M_m l_1 \ddot{Z}_{wi} - M_m h_{mw} \ddot{X}_{wi} = (-1)^{n-1} F_{mzi} (l_1 + l_2) - F_{mx} h_{dw} + M_q, \quad (4)$$

式中 $i=1\sim6$, 为一到六位轮对; $n=1, 2$, 分别为第一台和第二台转向架; M_w, M_m 分别为轮对质量和电机质量, I_{wy} 为轮对绕 y 轴的转动惯量; I_{my} 为电机绕 y 轴的转动惯量; $\ddot{X}_{wi}, \ddot{Z}_{wi}, \ddot{\beta}_{wi}$ 分别是轮对纵向加速度、垂向加速度和点头加速度; $\ddot{\beta}_{mi}$ 是电机点头加速度; F_{cx} 为纵向蠕滑力; P_i 是轮轨垂向力; F_{zi} 是一系悬挂垂向力; F_{mx}, F_{mz} 分别是电机悬挂在 x, z 方向的分量; M_q 为驱动力矩; r_i 为车轮滚动圆半径; l_1 为电机质心距轮对质心的纵向距离, l_2 为电机质心距吊杆的纵向距离, h_{dw}, h_{mw} 分别为轮对质心距构架处电机吊挂点和电机质心的垂向距离, 由于牵引电机采用抱轴式, 导致了上述轮对方程组的质量阵为非对角阵, 下面将对(1), (2), (4)进行解耦, 转化成如下标准形式, 以便于模型的数值求解:

$$\text{令 } T_{1i} = F_{cx} + F_{xf} + F_{mx}, \quad (5)$$

$$T_{2i} = -P_i + F_{mzi} + F_{zi} + (M_w + M_m)g, \quad (6)$$

$$T_{3i} = (-1)^{n-1} F_{mzi} (l_1 + l_2) + (-1)^n F_{mx} h_{dw} + M_q. \quad (7)$$

转化成标准形式:

$$A \ddot{X}_{wi} = (M_w M_m l_1^2 + M_m I_{ny}) T_{1i} + (-1)^n M_m^2 h_{mw} l_1 T_{2i} + (M_w M_m + M_m^2) h_{mw} T_{3i}, \quad (8)$$

$$A \ddot{\beta}_{wi} = (M_w M_m + M_m^2) h_{mw} T_{1i} + (-1)^n M_w M_m l_1 T_{2i} + (M_w M_m + M_m^2) T_{3i}, \quad (9)$$

$$A \ddot{Z}_{wi} = (-1)^n M_m^2 h_{mw} l_1 T_{1i} + (M_w M_m l_1^2 + M_m I_{ny} - M_m^2 h_{mw}) T_{2i} + (-1)^n M_w M_m l_1 T_{3i}, \quad (10)$$

$$\text{其中, } A = (M_w M_m + M_m^2) I_{ny} + M_w^2 M_m l_1^2 - (M_m^2 M_w + M_m^3) h_{mw}^2. \quad (11)$$

式(8)~(10)即为归一化的振动方程, 具有对角质量阵. 本文模型中, 轮轨接触法向力计算基于经典的Hertz非线性接触理论, 选择名义滚动圆处的车轮踏面几何参数作为接触斑计算的输入参数. 采用Kalker简化理论求解轮轨纵向蠕滑力. 为研究单节机车牵引状态下的运行行为, 需在车钩部位施加一个与机车牵引力大小相等、方向相反的作用力, 使得机车时刻在纵向处于受力平衡状态, 同时又能反映出驱动力矩对机车运行状态及轮轨接触状态的影响.

3 车轮不圆顺对重载机车轮轨相互作用的影响

基于上述动力学模型, 本节以径向偏差幅值最大的6R车轮不圆顺作为轮轨界面的输入激励, 研究牵引工况下其对机车轮轨作用力及接触应力的影响. 其中, 机车运行速度60 km/h, 对应的牵引力为290 kN, 该牵引力可进一步转化为方程(7)中牵引电机的驱动力矩 M_q .

图5给出了6R车轮运行 17×10^4 km时不圆顺激励下的轮轨纵向力、法向力、切向应力和法向应力. 可以看出, 牵引工况下, 不圆顺激发起轮轨作用力的较大波动, 纵向蠕滑力、轮轨法向力波动范围分别在20~70和47.5~194 kN, 由此可得到轮重减载率已达到0.61, 根据

图5 (网络版彩色)轮轨纵向蠕滑力和轮轨法向力(a)与轮轨切向应力和法向应力(b)

Figure 5 (Color online) (a) Wheel-rail longitudinal creep force and normal force; (b) wheel-rail tangential and normal stress

图6 (网络版彩色)轮轨垂向力响应(a)和轮轨垂向力频谱(b)

Figure 6 (Color online) Response of wheel-rail vertical force (a) and spectrum of wheel-rail vertical force (b)

GB5599-85^[15],超过了0.6的允许限度.对于轮轨接触应力,切向应力最大为800 MPa,而法向应力最大值则接近2100 MPa.

为研究车轮服役过程中不圆顺演化对轮轨动力作用的影响,以 $4.3 \times 10^4 \sim 17 \times 10^4$ km对应的6R车轮不圆顺作为输入激励,比较了不同激励下的轮轨垂向力动态响应,相应的时程曲线和频谱特性如图6所示.图6(a)表明,随着机车运营里程的增加,轮轨垂向力急剧增大, 4.3×10^4 , 10×10^4 , 17×10^4 km车轮不圆顺激发的轮轨垂向力最大值依次为134.3, 149.1, 194 kN,但轮轨力并未出现明显的高频冲击现象,主要以谐波组成为主.此外,轮轨垂向力随运营里程并非线性增大,越到后期变化越剧烈.例如,运营里程 $4.3 \times 10^4 \sim 10 \times 10^4$ km,里程增幅 5.7×10^4 km,对应的轮轨力增量15 kN,而运营里程在 $10 \times 10^4 \sim 17 \times 10^4$ km,里程虽只增加 7×10^4 km,轮轨力则增大44 kN之多.轮轨力的频谱特性对比显示,轮轨垂向力的主频出现在30 Hz,对应车轮不圆顺7阶谐波,且存在10,

72 Hz两个边频,分别对应3, 16阶车轮不圆顺谐波.

可见,车轮不圆顺演化过程中各阶谐波均不同程度地增大,但各阶谐波对轮轨动力作用的影响并不相同,对于六轴重载机车,7阶谐波的影响尤为显著,应在日常检修时予以重视.

4 结论

基于现场测试的不同运营里程下的重载六轴机车车轮不圆顺数据,得到了车轮不圆顺演变规律,三轴转向架端部轮对不圆顺发展速率比中间轮对更快,运行里程由 4.3×10^4 km增至 17×10^4 km时,车轮径向偏差幅值最大增幅约0.7 mm,1~3, 6~8阶谐波构成了不圆顺的主要成分,且随里程变化其增幅也最为明显.车轮不圆顺极易引起较大的轮轨垂向冲击振动和轮重减载现象,尤以7阶谐波影响最为显著,不利于机车牵引性能的稳定发挥.同时综合其他类型重载机车镟修周期,运营里程超过 10×10^4 km时,建议进行车轮不圆检修.

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Summary for “重载机车车轮不圆演变规律及动态影响”

Wheel polygon evolution and dynamic influence on heavy haul locomotive

Pengfei Liu^{1*}, Tianlong Wang¹, Kaikai Lü², Yunqiang Cao¹, Kailong Zhang¹ & Jingjing Song¹

¹ State Key Laboratory of Mechanical Behavior and System Safety of Traffic Engineering Structures, Shijiazhuang Tiedao University, Shijiazhuang 050043, China;

² State Key Laboratory of Traction Power, Southwest Jiaotong University, Chengdu 610031, China

* Corresponding author, E-mail: lpfswjtu@163.com

As the main traction equipment of heavy-haul trains in China, the high-power electric locomotive usually runs in a more complicated service environment with respect to other vehicles. The steering force, large traction force and electric braking force may act on the wheel-rail contact interface simultaneously, which has frequently led to some serious problems, such as wheel polygon and tread spalling, during the long-term service of locomotive. Thus, it is urgently needed to carry out the research on the wheel polygon status and its dynamic effect on the wheel-rail coupling vibration. The findings will have great significance for improving the locomotive dynamic performance and proposing the reasonable wheel maintenance procedures.

Taking the heavy-haul locomotive with 25 t axle load as the research object, the field tracking test of the wheel polygon was conducted. The test data indicates that the wheel polygon of the end wheelsets in three-axle bogie has a more rapid formation rate relative to the middle wheelset. When the locomotive running distance reaches 170000 km, the wheel radial deviation of the sixth wheelset is close to 0.9 mm which exceeds the normal limit value of 0.6 mm. The 1st, 2nd, 3rd, 6th, 7th and 8th order harmonics are the dominant components of the wheel polygon, which can excite the 4–40 Hz vertical vibration for the locomotive running at the speed range of 60–80 km/h.

To evaluate the influence of wheel polygon on the locomotive dynamic performance, a locomotive-track longitudinal and vertical coupling dynamic model was established. For the axle-hung traction motor, it is simplified as a rigid body which only has a pitching degree of freedom relative the wheel axle, and the other end is suspended on the bogie frame. The coupling vibration equations of wheelset and traction motor are derived in detail. In order to take the traction effect into account, a longitudinal running resistance is applied to the coupler, which is equal to the traction force at any time. So, the locomotive can be kept in a longitudinal equilibrium state without acceleration or deceleration, and the influences of motor driving torque on the wheel-rail dynamic interaction can also be reflected fully. There are total of 33 degrees of freedom in the locomotive dynamic submodel. For the ballasted track system, the rail is regarded as an elastic beam discretely supported at rail-sleeper junctions by three layers of springs and dampers. The vertical vibrations of rail, sleeper and ballast are considered.

By inputting the wheel polygon test data of into the wheel-rail contact interface of the proposed dynamic model, the wheel-rail shock vibration, normal contact stress and tangential stress are analyzed in this paper. The calculated results indicate that, the longer running distance causes the more severe wheel polygon which finally induces the larger wheel-rail vertical force and higher contact stress. For example, if the running distance increases from 43000 km to 100000 and 170000 km, the corresponding wheel-rail vertical forces can reach 134.3, 149.1 and 194 kN in sequence. Viewed from the spectrum analysis, the 7th-order harmonic wave has the most significant effect. Combining with the numerical results and the wheel repair cycle of other heavy-haul locomotives, it is recommended that the wheel polygonal maintenance should be carried out once the locomotive running distance exceeds 100000 km.

heavy haul locomotive, wheel polygon, coupling dynamics, wheel-rail contact

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