

考虑啮合冲击的电动汽车二级斜齿轮传动动力学特性分析

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摘要:【目的】探究啮合冲击对电动汽车二级斜齿轮传动系统动力学特性的影响规律, 为该类传动系统的优化设计提供参考。【方法】首先, 推导斜齿轮副啮入冲击力计算式, 为动力学激励精准量化提供依据; 其次, 基于集中质量法建立16自由度动力学模型, 构建弯曲-扭转-轴向耦合的动力学微分方程组, 表征系统多维度耦合振动特性; 然后, 采用龙格-库塔法求解方程组, 获取不同转速工况下的系统动力学响应; 最后, 分析各级齿轮副振动加速度与动态啮合力变化规律, 明确啮合冲击的影响机制。【结果】结果表明, 随输入转速从3 000 r/min升至12 000 r/min, 输入端、输出端齿轮副啮入冲击加速度峰值分别从686.21、223.35 m/s²升至2 291.61、1 131.77 m/s²; 转速升高显著放大了啮入冲击对系统的影响, 加剧了系统的不稳定性, 且输入端齿轮副的影响大于输出端; 相同转速下, 输入端对输出端啮入冲击的影响更强, 二者互扰频率与自身啮合频率一致。

关键词: 二级斜齿轮传动; 啮合冲击; 动力学特性; 转速; 加速度

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0 引言

斜齿轮副传动具有承载能力强、啮合平稳、传动效率高等优点, 被广泛应用于航空、船舶、汽车以及工程机械等各大领域。其动力学性能直接决定传动系统的运行可靠性与服役寿命, 一直是相关领域学者的研究重点。不同运行工况下, 多级斜齿轮传动系统中各级齿轮副的啮入冲击力通过轴系耦合作用, 会对系统动力学特性产生显著影响^[1-2]。因此, 明确啮入冲击对传动系统动力学特性的影响机制具有重要工程价值。

啮合冲击是齿轮系统的重要激励之一, 对动态齿轮冲击振动具有显著影响^[3-4]。MU等^[5]提出一种基于啮合冲击模型的振动控制优化设计方法, 该方法可有效减小啮合冲击, 提高弧齿锥齿轮的动态啮合性能。IMIN等^[6]建立了光滑粒子流体动力学离散方程组, 对齿轮的啮合冲击过程进行了模拟, 发现应力集中主要发生在节圆位置。DI等^[7]提出一种能够较

为准确模拟齿轮啮合冲击行为的光滑粒子流体动力学-有限元耦合方法。QI等^[8]提出一种考虑啮合误差、时变啮合刚度和啮合冲击的双输入双路径分流转矩传动系统动力学模型, 精确计算了各齿轮副的时变啮合刚度和啮合冲击。TEODORESCU等^[9]以微电子机械系统中的微齿轮副为对象, 建立了齿轮啮合冲击动力学的分级数学模型, 针对不同冲击速度开展了仿真分析。LIN等^[10]提出一种齿轮传动在任意位置处的啮合生成方法, 开发了轮齿啮合分析的自动建模程序, 为分析齿侧间隙对啮合冲击特性的影响提供了新思路。HE等^[11]提出一种可准确确定斜齿轮啮合冲击的方法, 并采用其自主开发的有限元程序验证了该啮合冲击分析模型的准确性。王峰等^[12]研究了多载荷状态下某船舶人字齿轮左端啮合副周向的振动特性, 发现啮合冲击激励对外载荷的敏感性高于啮合刚度激励。

综上所述, 现有研究主要围绕单对齿轮副的啮

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合冲击模型与冲击特性优化展开,缺乏在相关工况下对二级斜齿轮传动系统啮合冲击的系统分析。为此,以某电动汽车二级斜齿轮传动系统为研究对象,考虑啮合冲击力,采用集中质量法建立二级斜齿轮传动系统多自由度模型和动力学微分方程组,并采用龙格-库塔法进行求解,系统分析啮合冲击对各级斜齿轮副动力学特性的影响。

1 啮合冲击力的计算

齿轮副啮合冲击分为啮入冲击与啮出冲击,主要由啮合过程中接触面微小间隙、齿面不规则性或齿轮加工误差等因素引发的瞬时力或转矩突变所产生^[13]。啮合冲击力的产生具有不可避免性,主要来源包括受载变形与加工误差、啮合状态变化以及边界冲击三个方面^[14]。轮齿误差和弹性变形是影响啮合冲击点的两个主要因素,而啮合冲击点位置的确定可依据啮合原理与几何关系进行推导。由于在实际啮合过程中以啮入冲击为主^[15],所以研究只考虑啮入冲击的影响。斜齿轮传动啮合冲击计算原理如图1所示。

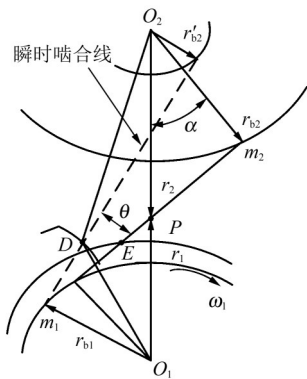


图1 斜齿轮啮入冲击示意图

Fig. 1 Schematic diagram of the meshing-in impact of helical gears

啮合冲击力计算式^[16]为

$$J_n = \frac{\pi \rho b}{2} (r_{bn}^4 - r_{bn}^4), n = 1, 2 \quad (1)$$

$$M_n = \frac{J_n}{r_{bn}^2}, n = 1, 2 \quad (2)$$

$$E_k = \frac{1}{2} \cdot \frac{J_1 J_2}{[J_1 (r'_{b2})^2 + J_2 r_{b1}^2]} v_s^2 \quad (3)$$

$$\theta = \arccos \frac{r'_{b2}}{r_{o2D}} - \angle PO_2 D - \alpha \quad (4)$$

$$F_{max} = v_s \sqrt{\frac{J_1 J_2}{[J_1 (r'_{b2})^2 + J_2 r_{b1}^2] (q_s + \cos^2 \theta \cdot q_q)}} \quad (5)$$

式中, J_1 、 J_2 分别为啮合齿轮主、从动轮的瞬时转动惯量; ρ 为齿轮材料密度; b 为齿宽; r_{b1} 、 r_{b2} 分别为主、从动轮基圆半径; r_{b1} 、 r_{b2} 分别为主、从动轮轮

毂内孔半径; M_1 、 M_2 分别为啮合齿轮主、从动轮瞬时啮合线上的诱导质量; E_k 为斜齿轮副任意节点 k 处的啮入冲击动能; r'_{b2} 为从动轮线外啮合瞬时基圆半径; v_s 为线外啮入冲击点的速度; α 为标准压力角; θ 为从动轮外啮入冲击点 D 处的法向载荷投影角; q_s 为啮合轮齿在线外初始啮合点 D 处的单齿对柔度; q_q 为啮入时刻除当前冲击齿对外,其他齿对的综合柔度; F_{max} 为最大啮合冲击力。

式(5)中, q_s 反映了单个齿对在载荷作用下的变形能力; q_q 反映了齿轮副的整体变形特性。这两个参数可通过轮齿综合啮合刚度乘以各自载荷分配系数获得。 D 、 E 两点之间的任意位置处的啮合冲击力皆可使用式(5)计算获得。

2 二级斜齿轮动力学建模与微分方程组

2.1 二级斜齿轮传动动力学模型

图2为某电动汽车二级斜齿轮传动系统简图。图3所示为采用集中质量法建立的其动力学模型。将系统中齿轮视为刚体圆盘,齿轮的轮齿及支撑等效为仅具有阻尼和刚度的阻尼器与弹簧;斜齿轮副的啮合力始终作用于啮合平面内;齿轮支撑等效作用于齿轮中心的等效节点上。在分析中不考虑齿轮的扭摆振动,引入坐标系 $O-xyz$, x 轴沿齿轮轴线方向; y 轴沿齿厚方向; z 轴沿齿高方向。 K_{xi} 、 K_{yi} 、 K_{zi} 分别为斜齿轮 $i(i=1, 2, 3, 4)$ 的 x 、 y 、 z 方向支撑处的等效支撑刚度; C_{xi} 、 C_{yi} 、 C_{zi} 分别为斜齿轮 $i(i=1, 2, 3, 4)$ 的 x 、 y 、 z 方向支撑处的等效支撑阻尼; K_{12} 、 K_{34} 分别为斜齿轮副12、34的时变啮合刚度; C_{12} 、 C_{34} 分别为斜齿轮副12、34的啮合阻尼; K_{23} 与 C_{23} 分别为中间弹性轴23的扭转刚度与扭转阻尼。以上参数依据文献[17]进行求解。 M_i^{eq} 与 I_i 分别为齿轮 $i(i=1, 2, 3, 4)$ 的等效集中质量与转动惯量; T_1 、 T_4 分别为系统输入、输出转矩; $\varphi_i(i=1, 2, 3, 4)$ 分别为构件 i 绕各自中心轴的扭转自由度。

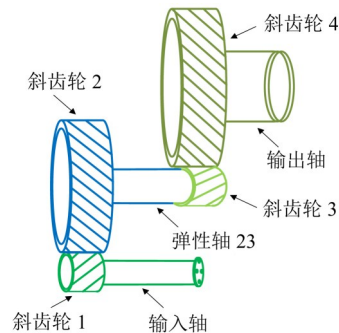


图2 二级斜齿轮传动系统结构简图

Fig. 2 Structural schematic diagram of the two-stage helical gear transmission system

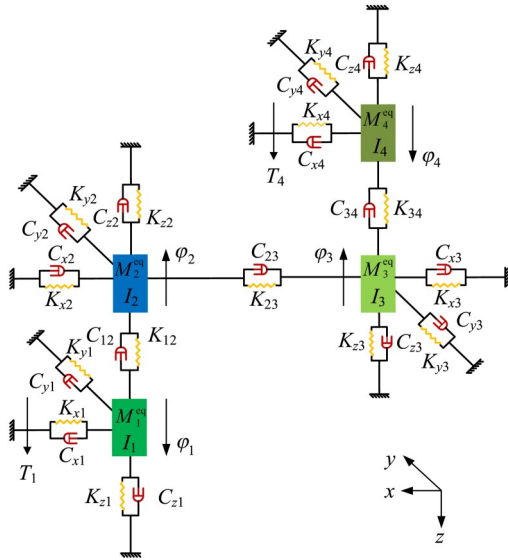


图3 二级斜齿轮传动系统集中质量动力学模型

Fig. 3 Lumped-mass dynamic model of the two-stage helical gear transmission system

2.2 齿轮副动态啮合力

图3所示的二级斜齿轮传动系统动力学模型是包含12个平移自由度和4个转动自由度的振动系统，其广义位移向量 U 可表示为

$$U = [x_1, y_1, z_1, \varphi_1, x_2, y_2, z_2, \varphi_2, x_3, y_3, z_3, \varphi_3, x_4, y_4, z_4, \varphi_4] \quad (6)$$

式中， x_i 、 y_i 、 z_i 分别为斜齿轮 i ($i=1, 2, 3, 4$) 在 x 、 y 、 z 方向的平移自由度。

设 Δx_i 、 Δy_i 、 Δz_i 分别表示两对斜齿轮副中第 i ($i=1, 2, 3, 4$) 个斜齿轮在 x 、 y 、 z 方向上的安装误差；斜齿轮副12、34因振动、误差产生的沿啮合线方向上的相对位移分别用 δ_{12} 、 δ_{34} 表示，其计算式为

$$\begin{cases} \delta_{12} = (\varphi_1 r_{b1} - \varphi_2 r_{b2}) \cos \beta_{12} - (y_1 - y_2 + \Delta y_1 - \Delta y_2) \sin \zeta_{12} \cos \beta_{12} - (z_1 - z_2 + \Delta z_1 - \Delta z_2) \cos \zeta_{12} \cdot \cos \beta_{12} - (x_1 - x_2 + \Delta x_1 - \Delta x_2) \sin \beta_{12} \\ \delta_{34} = (\varphi_3 r_{b3} - \varphi_4 r_{b4}) \cos \beta_{34} + (y_3 - y_4 + \Delta y_3 - \Delta y_4) \sin \zeta_{34} \cos \beta_{34} - (z_3 - z_4 + \Delta z_3 - \Delta z_4) \cos \zeta_{34} \cdot \cos \beta_{34} + (x_3 - x_4 + \Delta x_3 - \Delta x_4) \sin \beta_{34} \end{cases} \quad (7)$$

式中， ζ_{12} 、 ζ_{34} 分别为斜齿轮副12、34的啮合线与 z 轴正向夹角； β_{12} 、 β_{34} 分别为斜齿轮副12、34的螺旋角。

系统中两对斜齿轮副的弹性啮合力 P_{12} 、 P_{34} 和啮

$$\begin{cases} a_{12} = [(\ddot{\varphi}_1 r_{b1} - \ddot{\varphi}_2 r_{b2}) - (\ddot{y}_1 - \ddot{y}_2) \sin \zeta_{12} - (\ddot{z}_1 - \ddot{z}_2) \cos \zeta_{12}] \cos \beta_{12} - (\ddot{x}_1 - \ddot{x}_2) \sin \beta_{12} \\ a_{34} = [(\ddot{\varphi}_3 r_{b3} - \ddot{\varphi}_4 r_{b4}) - (\ddot{y}_3 - \ddot{y}_4) \sin \zeta_{34} - (\ddot{z}_3 - \ddot{z}_4) \cos \zeta_{34}] \cos \beta_{34} + (\ddot{x}_3 - \ddot{x}_4) \sin \beta_{34} \end{cases} \quad (12)$$

3 算例与分析

以某电动汽车二级斜齿轮传动系统为研究对象，系统输入转矩为100 N·m，基本参数如表1所示。

合阻尼力 D_{12} 、 D_{34} 可分别表示为

$$\begin{cases} P_{12} = K_{12} \delta_{12} \\ P_{34} = K_{34} \delta_{34} \end{cases} \quad (8)$$

$$\begin{cases} D_{12} = C_{12} \dot{\delta}_{12} \\ D_{34} = C_{34} \dot{\delta}_{34} \end{cases} \quad (9)$$

式中， $\dot{\delta}_{12}$ 、 $\dot{\delta}_{34}$ 分别为斜齿轮副12、斜齿轮副34沿啮合线方向的相对速度，可由式(7)对时间求导得到。

斜齿轮副12、斜齿轮副34的动态啮合力 F_{12} 、 F_{34} 的计算式分别为

$$\begin{cases} F_{12} = P_{12} + D_{12} \\ F_{34} = P_{34} + D_{34} \end{cases} \quad (10)$$

2.3 动力学微分方程组的建立与求解

建立二级斜齿轮传动系统弯曲-扭转-轴向耦合的动力学运动微分方程组，可表示为

$$\begin{cases} M_1^{\text{eq}} \ddot{x}_1 + C_{x1} \dot{x}_1 + K_{x1} x_1 = F_{12} \sin \beta_{12} \\ M_1^{\text{eq}} \ddot{y}_1 + C_{y1} \dot{y}_1 + K_{y1} y_1 = F_{12} \cos \beta_{12} \sin \zeta_{12} \\ M_1^{\text{eq}} \ddot{z}_1 + C_{z1} \dot{z}_1 + K_{z1} z_1 = F_{12} \cos \beta_{12} \cos \zeta_{12} \\ I_1 \ddot{\varphi}_1 = T_1 - F_{12} \cos \beta_{12} r_{b1} - F_{s12} r_{b1} \\ M_2^{\text{eq}} \ddot{x}_2 + C_{x2} \dot{x}_2 + K_{x2} x_2 = -F_{12} \sin \beta_{12} \\ M_2^{\text{eq}} \ddot{y}_2 + C_{y2} \dot{y}_2 + K_{y2} y_2 = -F_{12} \cos \beta_{12} \sin \zeta_{12} \\ M_2^{\text{eq}} \ddot{z}_2 + C_{z2} \dot{z}_2 + K_{z2} z_2 = -F_{12} \cos \beta_{12} \cos \zeta_{12} \\ I_2 \ddot{\varphi}_2 + C_{23} (\dot{\varphi}_2 - \dot{\varphi}_3) + K_{23} (\varphi_2 - \varphi_3) = F_{12} \cos \beta_{12} r_{b2} + F_{s12} r_{b2} \\ M_3^{\text{eq}} \ddot{x}_3 + C_{x3} \dot{x}_3 + K_{x3} x_3 = -F_{34} \sin \beta_{34} \\ M_3^{\text{eq}} \ddot{y}_3 + C_{y3} \dot{y}_3 + K_{y3} y_3 = -F_{34} \cos \beta_{34} \sin \zeta_{34} \\ M_3^{\text{eq}} \ddot{z}_3 + C_{z3} \dot{z}_3 + K_{z3} z_3 = F_{34} \cos \beta_{34} \cos \zeta_{34} \\ I_3 \ddot{\varphi}_3 + C_{23} (\dot{\varphi}_3 - \dot{\varphi}_2) + K_{23} (\varphi_3 - \varphi_2) = -F_{34} \cos \beta_{34} r_{b3} + F_{s34} r_{b3} \\ M_4^{\text{eq}} \ddot{x}_4 + C_{x4} \dot{x}_4 + K_{x4} x_4 = F_{34} \sin \beta_{34} \\ M_4^{\text{eq}} \ddot{y}_4 + C_{y4} \dot{y}_4 + K_{y4} y_4 = F_{34} \cos \beta_{34} \sin \zeta_{34} \\ M_4^{\text{eq}} \ddot{z}_4 + C_{z4} \dot{z}_4 + K_{z4} z_4 = -F_{34} \cos \beta_{34} \cos \zeta_{34} \\ I_4 \ddot{\varphi}_4 = F_{34} \cos \beta_{34} r_{b4} + F_{s34} r_{b4} - T_4 \end{cases} \quad (11)$$

式中， F_{s12} 、 F_{s34} 分别为齿轮副12、34的啮合力。

为消除刚体位移对求解结果的影响，避免方程组求解过程中因物理量数量级差异过大导致的求解失真，对微分方程组进行无量纲化处理^[18]。采用龙格-库塔法^[19]求解无量纲化后的微分方程组，获得各级齿轮副沿啮合线方向的相对振动加速度，计算式为

结合上文分析与表1的数据，代入式(11)，计算不同转速下各级齿轮副啮合力，结果如表2所示。

为明确啮入冲击激励下系统的时域动力学特性，选取输入转速3 000、6 000、12 000 r/min工况下，

求解得到啮入冲击激励下齿轮副12与齿轮副34的啮合线方向加速度时程曲线,分别如图4~图9所示。

表1 斜齿轮传动系统参数

Tab.1 Parameters of the helical gear transmission system

参数	一级		二级	
	1	2	3	4
齿轮	1	2	3	4
旋向	左	右	左	右
模数 m/mm	4	4	4	4
齿数	23	80	22	87
压力角 $\alpha/(\circ)$	20	20	20	20
螺旋角 $\beta/(\circ)$	20	20	20	20
齿顶高 h_a/mm	4	4	4	4
齿根高 h_f/mm	5	5	5	5
齿宽 b/mm	33	31.5	40	38
径向支撑刚度 $K_r, K_r/(\text{N/m})$	2.0850×10^9	1.8269×10^9	1.8569×10^9	2.3698×10^9
轴向支撑刚度 $K_x/(\text{N/m})$	2.2159×10^9	1.9949×10^9	1.9949×10^9	1.4097×10^9

表2 不同转速下各级齿轮副啮合力对比

Tab.2 Comparison of meshing forces of gear pairs under different rotational speeds

输入转速/(r/min)	齿轮副12啮合力 F_{s12}/N	齿轮副34啮合力 F_{s34}/N
3 000	1 245.94	89.21
6 000	2 419.89	178.41
12 000	4 983.77	357.21

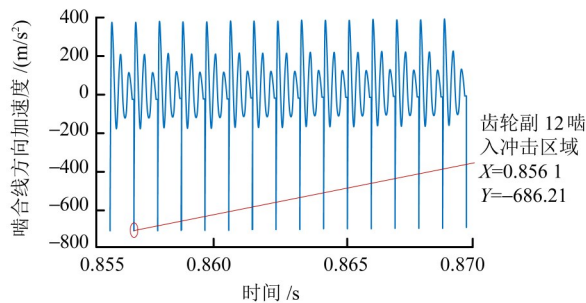


图4 输入转速3 000 r/min下齿轮副12啮合线方向加速度曲线
Fig.4 Acceleration curve in the meshing line direction of the gear pair 12 at an input speed of the 3 000 r/min

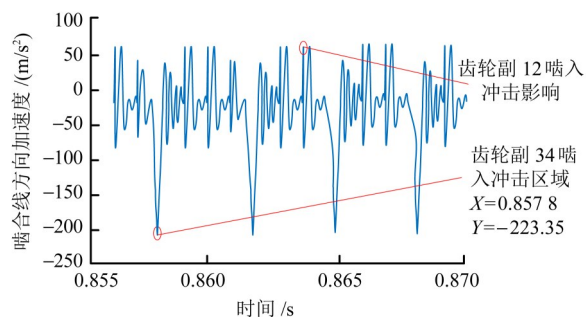


图5 输入转速3 000 r/min下齿轮副34啮合线方向加速度曲线
Fig.5 Acceleration curve in the meshing line direction of the gear pair 34 at an input speed of the 3 000 r/min

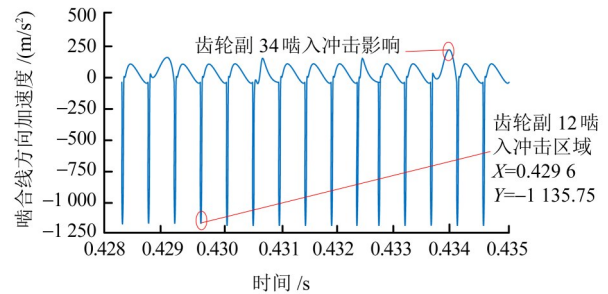


图6 输入转速6 000 r/min下齿轮副12啮合线方向加速度曲线
Fig.6 Acceleration curve in the meshing line direction of the gear pair 12 at an input speed of the 6 000 r/min

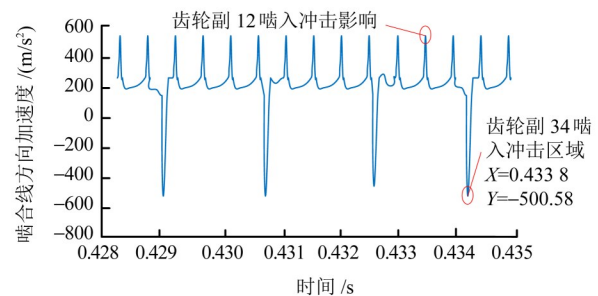


图7 输入转速6 000 r/min下齿轮副34啮合线方向加速度曲线
Fig.7 Acceleration curve in the meshing line direction of the gear pair 34 at an input speed of the 6 000 r/min

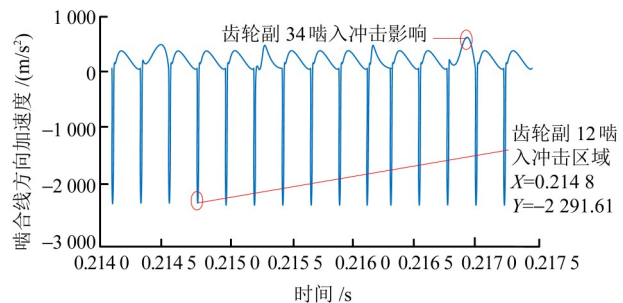


图8 输入转速12 000 r/min下齿轮副12啮合线方向加速度曲线
Fig.8 Acceleration curve in the meshing line direction of the gear pair 12 at an input speed of the 12 000 r/min

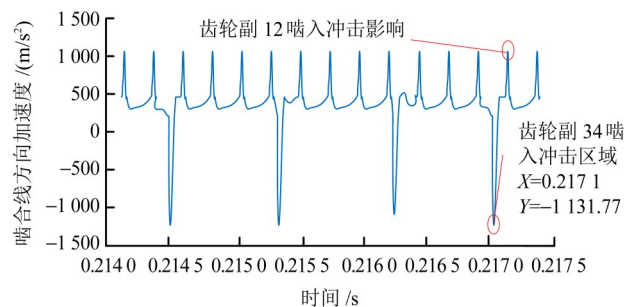


图9 输入转速12 000 r/min下齿轮副34啮合线方向加速度曲线
Fig.9 Acceleration curve in the meshing line direction of the gear pair 34 at an input speed of the 12 000 r/min

由图4、图5可知,当输入转速为3 000 r/min时,齿轮副34的啮入冲击对齿轮副12啮合线方向加速度的影响可忽略;齿轮副12的转速相对于齿轮副34较高,其啮入冲击对齿轮副34啮合线方向加速度的影响较为明显。

由图4、图6、图8可知,输入转速从3 000 r/min升至12 000 r/min时,齿轮副12啮入冲击区域的加速度峰值分别为686.21、1 135.75、2 291.61 m/s^2 。由图5、图7、图9可知,相同转速区间内,齿轮副34啮入冲击区域的加速度峰值分别为223.35、500.58、1 131.77 m/s^2 。转速升高加剧了齿轮间的相对运动,使各级齿轮副啮入冲击区域的加速度峰值显著增大,系统运行不稳定性加剧;相同转速(12 000 r/min)下,齿轮副12的加速度峰值为2 291.61 m/s^2 ,始终高于齿轮副34的加速度峰值1 131.77 m/s^2 ,说明输入端齿轮副的啮入冲击对系统振动的影响大于输出端齿轮副。

此外,对比相同转速下各级齿轮副的啮合线方向加速度曲线(图4与图5、图6与图7、图8与图9)可知,齿轮副12啮入冲击引发的加速度波动幅值显著大于齿轮副34,说明输出端齿轮副受到的啮入冲击耦合影响强于输入端齿轮副。其核心原因在于齿轮副12与输入轴直接相连,相同输入转速下,齿轮副12的啮合转速高于齿轮副34,啮入冲击力更大;同时,各级齿轮副相互影响的频率与自身啮合频率完全一致。

4 结论

考虑啮入冲击力,根据集中质量法建立了电动汽车二级斜齿轮传动系统的16自由度动力学模型,并建立了其动力学微分方程组,系统分析不同输入转速下各级齿轮副的动力学响应规律,得出如下主要结论:

1) 输入转速升高显著放大啮入冲击对传动系统的影响,各级齿轮副啮入冲击区域的加速度峰值随转速升高呈线性增长趋势,系统运行不稳定性加剧;相同转速变化区间内,输入端齿轮副啮入冲击对系统振动的影响程度大于输出端齿轮副。

2) 相同转速工况下,输入端齿轮副对输出端齿轮副啮入冲击的耦合影响,显著强于输出端齿轮副对输入端的反向影响;两级齿轮副间存在相互耦合的啮入冲击效应,且二者相互影响的频率与自身啮合频率完全一致。

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Dynamic characterization of two-stage helical gear transmission for electric vehicles considering mesh impact

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Abstract: [Objective] The influence of meshing impact on the dynamic characteristics of two-stage helical gear transmission systems in electric vehicles was investigated to provide references for the design of such transmissions. [Methods] A two-stage helical gear transmission system of an electric vehicle was taken as the research object. A calculation formula for meshing impact force was derived; a 16-degree-of-freedom dynamic model was established based on the lumped mass method, and a set of dynamic differential equations considering bending-torsional-axial coupling was constructed. The Runge-Kutta method was used for solution, and the dynamic characteristics of each gear pair under different rotational speeds were analyzed. [Results] The results show that when the input speed increases from 3 000 r/min to 12 000 r/min, the peak meshing-in impact acceleration of the input-end and output-end gear pairs rises from 686.21 m/s² and 223.35 m/s² to 2 291.61 m/s² and 1 131.77 m/s², respectively. The increase of rotational speed significantly amplifies the influence of meshing-in impact on the system and aggravates the system instability, and the influence of the input-end gear pair is greater than that of the output-end one. At the same rotational speed, the input-end gear pair has a stronger influence on the meshing-in impact of the output-end gear pair, and the mutual interference frequency between the two is consistent with their own meshing frequencies.

Key words: Two-stage helical gear transmission; Mesh impact; Dynamic characterization; Rotational speed; Acceleration

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Test analysis of anti-friction/wear of fullerene C60 lubricating oil to planetary gear shaft

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Abstract: [Objective] Effective lubrication is one of the most efficient methods for reducing friction and minimizing wear on metal component surfaces in mechanical motion pairs. To address the rapid friction and wear failure of the planetary gear shaft in differential assemblies, the anti-wear and friction-reduction effects of fullerene C60 nano-carbon sphere lubricant on planetary gear shafts were investigated. [Methods] Firstly, friction and wear tests were conducted under the action of fullerene C60 lubricant to analyze the influence of C60 nano-carbon sphere concentration on the friction coefficient and wear scar surface morphology. Secondly, a differential assembly fatigue life test bench was utilized to perform fatigue wear tests on the differential planetary gear shaft, observing the friction-reducing effects of C60 nano-carbon spheres on the gear shaft. [Results] The analysis results demonstrate that the average friction coefficient under the action of fullerene C60 nano-carbon sphere lubricant is as low as 0.080 1, representing a 71.1% reduction compared to the base lubricant. Furthermore, the wear amounts on both ends of the planetary gear shaft pins decrease by approximately 60%.

Key words: Fullerene C60; Friction and wear; Differential assembly; Lubricating oil additive

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