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基于拉格朗日法的悬臂输送槽阻尼摆模型构建与验证

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摘要：为准确分析联合收割机输送槽在作业时产生的不平衡摆动现象，该研究建立了基于液压缸支撑的联合收割机悬臂输送槽阻尼摆模型。首先对悬臂输送槽系统进行受力分析，并基于拉格朗日法建立其动力学微分方程。为进一步确定动力学微分方程中的参数，利用有限元法和模态分析法对液压缸刚度和阻尼进行参数标定；并进行田间试验获取输送槽所受外界激励。利用二次积分与快速傅里叶变换（fast fourier transform，FFT）将田间试验采集到的加速度信号处理得到悬臂输送槽振幅及振动的主频成分，为动力学微分方程的验证提供依据。最终求解悬臂输送槽角速度和摆角的表达式，将理论得到的初始状态下悬臂输送槽摆幅与田间试验实际摆幅进行对比。结果表明：稳定状态下，理论求解得到的位于同一水平位置两测点对应的摆幅分别约为0.0979、0.0957mm，与试验实际摆幅误差约为1.11%和4.30%，验证了该联合收割机悬臂输送槽阻尼摆模型的准确性。研究结果可为长江中下游地区水稻联合收割机悬臂输送槽结构优化提供理论依据。

关键词：模型；仿真；阻尼摆；拉格朗日方程；参数标定

0 引 言

水稻联合收割机的工作环境较为恶劣多变[1-3]，整机作业时产生的剧烈振动会严重影响驾驶舒适性[4-7]。其中，联合收割机悬臂输送槽在多种激励作用下产生的不平衡摆动会大大降低机械的使用寿命和收获效率[8]。因此，建立联合收割机悬臂输送槽阻尼摆模型准确描述输送槽在作业时的不平衡摆动现象对于联合收割机输送槽的结构优化具有重要意义[9-12]。

对于悬臂输送槽阻尼摆结构，不少学者对其进行了相关研究。首先是收割机部件所受激励影响方面，丁翰韬等通过开展田间变喂入量水稻收获振动试验，得知受喂入量对脱粒滚筒振动影响最为明显[13]；王瑞等通过搭建载荷测试系统和产量检测系统构建动力学模型以获得极限工况下联合收割机不同部件的载荷[14]。部分学者针对不同摆系统的动力学特性展开了研究，HAN等在粘性阻尼和谐波激励足够小的扰动下，从理论上和数值上研究了具有无理和分数非线性恢复力的经典摆系统[15]；Rubio R M等研究了摆在恒定高斯曲率表面上的外力自由运动的动力学问题，得到了常曲率面上摆的无外力运动的解析方法[16]；程杰等借助MATLAB仿真技术,实现了同时考虑阻尼和受迫的复杂情况下单摆非线性运动仿真模拟[17]。也有部分学者聚焦于摆系统动力学模型建立方面，金诚谦等采用离散元法（Discrete Element Method，DEM）结合机械系统耗散函数和拉格朗日公式对振动输送机建立数学模型[18]；林洪文等建立了一种扭摆振动非线性数学模型，减小传统扭摆模型在大尺寸板状物体转动惯量的测量误差[19]。

相关研究主要集中在对联合收割机其他部件的模态测试、振动采集、载荷测试等工作，能够解决单个部件的结构优化和振动问题[20-23]。但对于考虑喂入量影响，建立水稻联合收割机有阻尼悬臂输送槽系统动力学模型的相关研究还未报道。鉴于此，本文基于拉格朗日法和振动信号，通过开展半割幅和满割幅两种工况下联合收割机田间振动试验，对输送槽系统激振特性展开分析，建立联合收割机悬臂输送槽阻尼摆模型，以期为水稻联合收割机结构优化提供参考。

1 悬臂输送槽阻尼摆模型

1.1悬臂输送槽系统受力分析

如图1所示，悬臂输送槽系统结构复杂且受力形式多变，收割机工作过程中拨禾轮的转动、割刀的剪切运动、割台搅龙的输送运动等都会对输送槽系统运动状态产生影响。

a 悬臂输送槽系统结构图

a Structural diagram of cantilever conveying trough system

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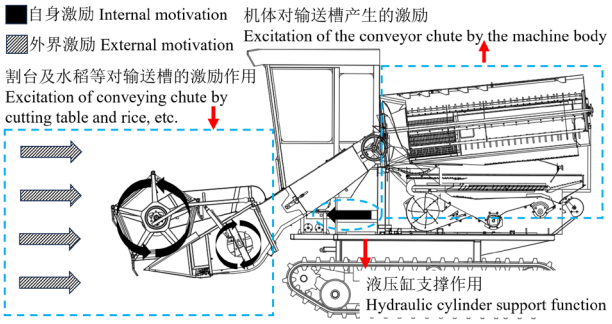


b 悬臂输送槽系统所受激励

b Excitations for cantilever conveyor trough system

图1 悬臂输送槽系统及所受激励

Fig.1 Cantilever conveyor trough system and excitations

通过测量获取该型号收割机液压缸各构件及行程参数值如表1所示：

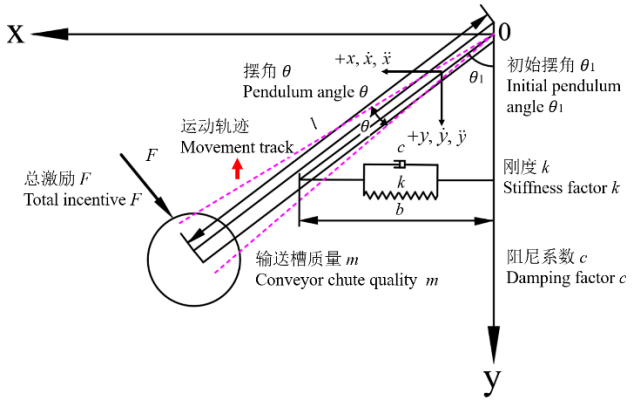
表1 液压缸各构件参数

Tab.1 Parameters of hydraulic cylinder and stroke components

mm

参数Parameters	数值Value
活塞杆长度Piston rod length	450
活塞杆半径Piston rod radius	20
活塞长度Piston length	50
内腔长度Inner chamber length	450
内腔半径Inner cavity radius	25
缸筒长度Cylinder length	700
缸筒半径Cylinder radius	32.5
工作行程Work itinerary	300
有杆腔长度Length of rod cavity	450
无杆腔长度Length of rodless cavity	150

联合收割机悬臂输送槽是一种复杂的受迫摆结构，对于其力学方程参数的求解可以先简化其模型，再根据受迫摆模型求解方式获取[24-26]。依据联合收割机工作机理与结构受力特性，对系统结构进行简化处理，简化后的有阻尼输送槽模型如图2所示。此外，复杂的激励源并不容易分析，同时还会降低所建立的动力学微分方程的准确性与可靠性。所以本研究将上述所有激励考虑为悬臂输送槽系统的总激励。



注：θ 表示悬臂输送槽摆角。l 表示悬臂输送槽长度，mm。b 表示旋转轴与液压缸连接点的距离，mm。

Note：θ denotes the swing angle of the cantilever conveyor trough. l denotes the length of the cantilever conveyor trough, mm. and b denotes the distance between the rotary axis and the connection point of the hydraulic cylinder, mm.

图2 有阻尼悬臂输送槽简化模型

Fig.2 Simplified model of damped cantilever conveyor trough

悬臂输送槽所受外力主要分为两部分：一部分为水稻等对输送槽的作用力，用混合激励 F 表示并作用在输送槽与割台连接处。另一部分则是液压缸元件在工作过程中所受到的阻尼力 F'，可表示为

$$\begin{cases} F = ma \\ F' = cb\dot{\theta}\left[\cos\left(\theta_1+\theta\right)\right]^2 \end{cases} \tag{1}$$

式中 a 表示输送槽与割台连接处的加速度，m/s²。$\dot{\theta}$ 表示悬臂输送槽广义角速度，rad/s。

1. 2悬臂输送槽阻尼摆模型

对于已知的n自由度系统，其动力学微分方程可通过拉格朗日方程求解得到[27-29]。如式（2）所示：

$$\frac{\mathrm{d}}{\mathrm{d}t}\left(\frac{\partial T}{\partial \dot{q}_j}\right)-\frac{\partial T}{\partial q_j}+\frac{\partial V}{\partial q_j}=Q_j^{(n)} \tag{2}$$

上式中，j 为系统的自由度，$j=1,2,3,\cdots,n$。T 为系统的动能，J。V 为系统的势能，J。q_j 为广义坐标。$\dot{q}_j=\partial q_j/\partial t$ 为广义速度，m/s²。$Q_j^{(n)}$ 为相对于广义坐标 q_j 的非保守广义力，N。

如图2所示，选择广义坐标来描述悬臂输送槽在平面摆动的角位移，其水平与竖直方向上的位移分量（x、y）与速度分量（$\dot{x}$、$\dot{y}$）为

$$\begin{cases} x=\dfrac{1}{2}l\sin\left(\theta+\theta_1\right)-\dfrac{l}{2}\sin\theta_1 \\ y=\dfrac{1}{2}l\cos\theta_1-\dfrac{1}{2}l\cos\left(\theta+\theta_1\right) \end{cases} \tag{3}$$

$$\begin{cases} \dot{x}=\dfrac{l}{2}\dot{\theta}\cos\left(\theta+\theta_1\right) \\ \dot{y}=\dfrac{l}{2}\dot{\theta}\sin\left(\theta+\theta_1\right) \end{cases} \tag{4}$$

悬臂输送槽的动能 T 为

$$T=\frac{1}{2}m\left(\dot{x}^2+\dot{y}^2\right)+\frac{1}{2}J\dot{\theta}^2 \tag{5}$$

式中 J 表示输送槽相对旋转轴的转动惯量且 $J=ml^2/3$，kg·m²。

联立式（4）~（5）可得：

$$T=\frac{7}{24}ml^2\dot{\theta}^2 \tag{6}$$

悬臂输送槽系统的重力势能V_1和弹性势能V_2为：

$$\begin{cases} V_1=\dfrac{1}{2}mgl\left[\cos\left(\theta_1+\theta\right)-\cos\theta_1\right] \\ V_2=\dfrac{1}{2}kx_y^2 \end{cases} \tag{7}$$

式中x_y为输送槽系统液压缸水平位移，mm。

则悬臂输送槽系统势能V为

$$\begin{aligned} V=&kb^2\left[\sin\left(\theta_1+\theta\right)-\sin\theta_1\right]^2/2 \\ &+mgl\left[\cos\left(\theta_1+\theta\right)-\cos\theta_1\right]/2 \end{aligned} \tag{8}$$

根据拉格朗日方程法的计算原理，需计算动能与势能分别对广义角位移与广义角速度的偏微分及微分表达式。

其中，系统动能T对广义角速度$\dot{\theta}$的偏微分计算结果为：

$$\frac{\partial T}{\partial\dot{\theta}}=\frac{7}{12}ml^2\ddot{\theta} \tag{9}$$

系统势能V对广义角位移θ的偏微分计算结果为：

$$\begin{aligned} \frac{\partial V}{\partial\theta}=&kb^2\sin\left(\theta_1+\theta\right)\cdot\cos\left(\theta_1+\theta\right) \\ &-kb^2\cos\left(\theta_1+\theta\right)\cdot\sin\theta_1 \\ &+\frac{1}{2}mgl\sin\left(\theta_1+\theta\right) \end{aligned} \tag{10}$$

将式（9）~（10）代入式（2）即可得到基于拉格朗日方程法推导的在θ坐标下悬臂输送槽动力学微分方程：

$$\begin{aligned} &\frac{7}{12}ml^2\ddot{\theta}+kb^2\sin\left(\theta_1+\theta\right)\cdot\cos\left(\theta_1+\theta\right) \\ &-kb^2\cos\left(\theta_1+\theta\right)\cdot\sin\theta_1+\frac{1}{2}mgl\sin\left(\theta_1+\theta\right) \\ &=Fl-cb^2\dot{\theta}\left[\cos\left(\theta_1+\theta\right)\right]^2 \end{aligned} \tag{11}$$

由于悬臂输送槽在水平与竖直自由度上平衡是以力的形式呈现，因此在x坐标下的结果如下：

$$\begin{aligned} &m\ddot{x}+\frac{1}{2}[ml\cos(\theta_0+\theta)]\ddot{\theta} \\ &+\frac{1}{2}[ml\sin(\theta_0+\theta)]\dot{\theta}^2 \\ &=F\cos(\theta_0+\theta)-c\dot{x} \end{aligned} \tag{12}$$

同理，在y坐标下的结果如下：

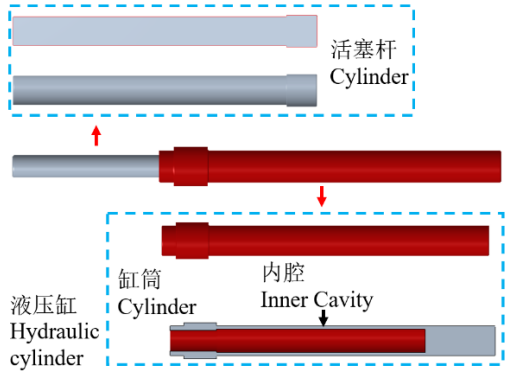
$$\begin{aligned} &m\ddot{y}+\frac{1}{2}[ml\sin(\theta_0+\theta)]\ddot{\theta} \\ &-\frac{1}{2}[ml\cos(\theta_0+\theta)]\dot{\theta}^2 \\ &=F\sin(\theta_0+\theta) \end{aligned} \tag{13}$$

式(11)~(13)即为三自由度下悬臂输送槽阻尼摆模型，在上述系统动力学微分方程中，参数k、c和F需要确定。

2悬臂输送槽动力学微分方程参数

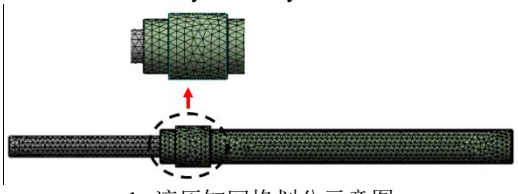
2.1液压缸刚度k

利用液压缸参数在Creo中绘制液压缸模型（活塞杆与缸筒）并导入到ANSYS Workbench中。对液压缸模型施加材料属性，缸筒与活塞杆的材料为45号钢，材料参数如下：密度7890 kg/m^3，弹性模量209 GPa，泊松比0.269。液压油为不可压缩性液体材料，其材料参数为：密度870 kg/m^3，弹性模量1.6 GPa，泊松比0.49。



a. 液压缸模型图

a. Hydraulic cylinder model



b. 液压缸网格划分示意图

b. Hydraulic cylinder mesh diagram

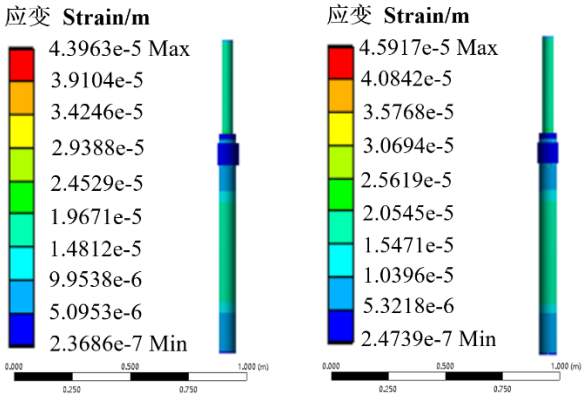
图3 液压缸模型及网格划分示意图

Fig.3 Hydraulic cylinder model and mesh diagram

对液压缸模型施加作用在活塞杆上表面的载荷，通过查阅该款联合收割机的规格参数，可知

该型号联合收割机割台质量约为330 kg，输送槽质量约为120 kg，所以静载状态下液压缸受力最小约为4500 N。当联合收割机处于工作状态时，水稻茎秆的质量等会增大液压缸的受载，但影响较小。故本研究设定工作状态下液压缸受力约为4700 N。

完成上述步骤后，分析液压缸在静载状态和工作状态下的应变情况，应变分布云图如下：



a.静载状态　　b. 工作状态

a. Static load state　　b. Working state

图4 液压缸应变分布云图

Fig.4 Strain distribution of hydraulic cylinder

由图4可知，液压缸静载状态和工作状态的最大应变分别为4.3963×10^{-5} m和4.8848×10^{-5} m。因此可求解两种状态下液压缸的最小刚度分别为1.02358802×10^{8}与1.02358336×10^{8} N/m，故液压缸刚度为1.0236×10^{8} N/m。

2.2液压缸阻尼参数c

2.2.1 液压缸阻尼

求解阻尼参数可以通过对数衰减法来实现，同时引入无量纲参数阻尼比ζ[30]：

$$\zeta=\frac{c/\left(2m_1\right)}{\sqrt{k/m_1}}=\frac{c}{2\sqrt{m_1k}}=\frac{c}{2m_1\omega_n}=\frac{c}{c_c} \tag{14}$$

式中m_1表示液压缸质量($m_1=17.09\text{kg}$)。ω_n表示系统的固有频率，Hz。c_c表示系统临界阻尼系数($c_c=2m_1\omega_n$)。

有阻尼系统的自由振动振幅按指数规律衰减，设经过一个自然周期T_d，同方向相邻两个振幅分别为A_i与A_{i+1}，表达式为

$$\begin{cases} A_i=A\mathrm{e}^{-\zeta\omega_n t_i}\sin\left(\omega_d t_i+\varphi\right) \\ A_{i+1}=A\mathrm{e}^{-\zeta\omega_n\left(t_i+T_d\right)}\sin\left[\omega_d\left(t_i+T_d\right)+\varphi\right] \end{cases} \tag{15}$$

式中ω表示无阻尼频率。ω_d为$0\le\xi<1$时的有阻尼频率且$\omega_d=\omega\sqrt{1-\xi^2}$。

则利用相邻两振幅之比取自然对数可以获得振幅对数衰减率δ为

$$\delta=\ln\eta=\zeta\omega_nT_d=\frac{2\pi\zeta}{\sqrt{1-\zeta^2}} \tag{16}$$

式(16)表明，振幅对数衰减率仅与阻尼比ζ有关。当阻尼比很小时，对数衰减率δ为

$$\delta=2\pi\zeta \tag{17}$$

联立式（14）~（17）可得液压缸阻尼参数表达式为

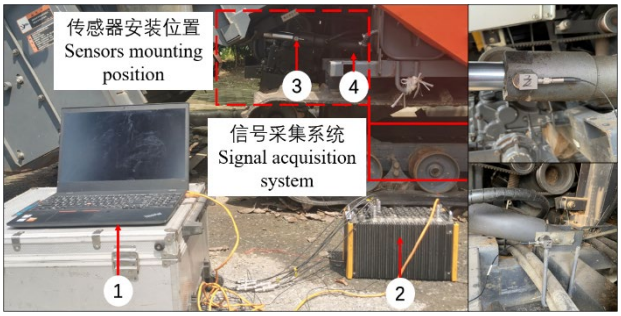
$$c=2\zeta m_1\omega_n \tag{18}$$

利用ANSYS Workbench获取液压缸前6阶约束模态频率分别为60.604、60.672、291.92、292.01、702.77和702.83 Hz，液压缸的主振型对应的固有频率ω_n为60.604 Hz。

2.2.2液压缸振动采集试验

本文拟通过采集液压缸振动信号以获取阻尼比ζ。试验机器为某水稻联合收割机，将机器启动保证各部件都处在运动状态下，发动机转速稳定在额定转速。信号采集系统包括DH5902信号采集分析仪、1A312E三向加速度传感器和电脑。

在液压缸缸筒前端与末端分别设置测点，传感器的x轴方向对应液压缸伸缩方向，采样频率设为1 kHz，加速度传感器安装位置如图4所示。



1.电脑 2.DH5902信号采集分析仪 3.加速度传感器1 4.加速度传感器2

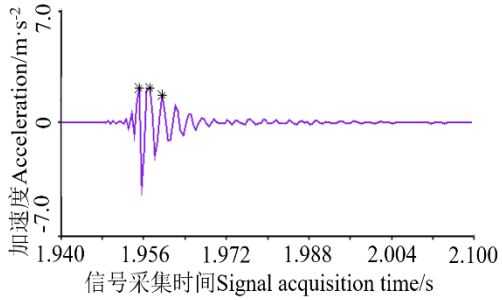
1. Laptop 2. DH5902 signal acquisition analyser 3. Measurement point 1 4. Measurement point 2

图5 液压缸振动采集现场

Fig.5 Hydraulic cylinder vibration acquisition site

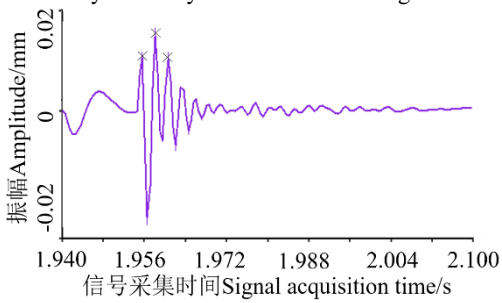
2.2.3液压缸振动信号处理

本次试验采集10 s加速度信号，并选取其中较为稳定时间区段（1.94~2.1 s）进行分析，如图6所示:



a.液压缸加速度信号

a. Hydraulic cylinder acceleration signal



b.液压缸振幅

b. Hydraulic cylinder amplitude

图6 液压缸振动信号处理

Fig.6 Vibration signal processing for hydraulic cylinder

本次采集的加速度响应信号整体成衰减的趋势，符合液压缸的实际工作情况。液压缸加速度响应的前三阶峰值点分别为2.286、2.319和1.740 m/s^2，振幅响应的前三阶峰值点分别为0.002、0.001和0.001 mm。因此计算振幅对数衰减率时应选取1、3号相邻振幅来进行计算。

联立式(14)~(17)计算可得：

$$\delta = \ln 2，\ c = 228.58\text{N}\cdot\text{s/m}$$

2.3外界激励

2.3.1田间试验

试验场地位于中国江苏省镇江市丹徒区辛丰镇的某处稻田。为了考虑喂入量对割台与输送槽连接点加速度的影响，将联合收割机设置为半割幅和满割幅两种状态下匀速行进，发动机转速稳定在额定转速。每次试验长度为35 m，同时为考虑联合收割机在试验过程中以稳定喂入量工作，在工作区前预留5 m作物。利用采集仪记录联合收割机在试验区域25 s内的振动信号。

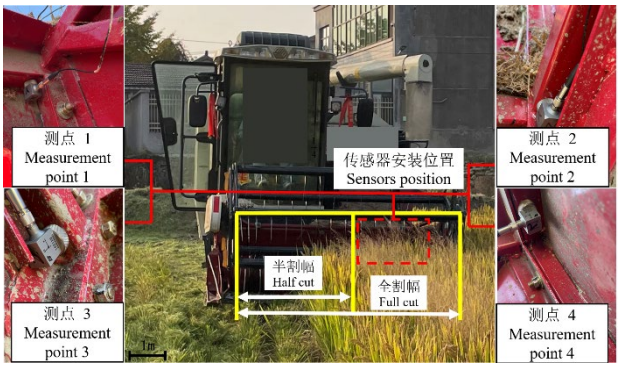


图7 悬臂输送槽振动采集现场

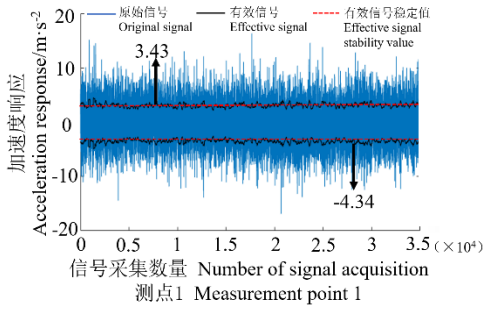
Fig.7 Cantilever conveyor trough vibration acquisition site

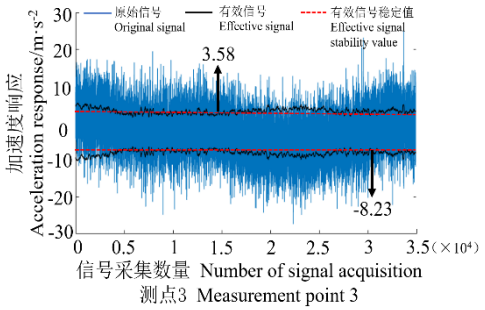
在联合收割机割台与输送槽连接处的4个对角设置对称测点，加速度传感器放置位置如图7所示。每个传感器测量*XYZ*三轴的加速度，共12个通道，测量联合收获机割台与输送槽连接处在田间作业时不同喂入量情况下振动加速度变化情况。

2.3.2田间试验信号处理

当联合收割机在田间匀速作业时，传感器开始采集振动信号。为了能够更好地反映悬臂输送槽受力情况，选取相对稳定的信号区段（15~50 s）。联合收割机半割幅和满割幅两种工况下采集到的加速度响应信号如图8所示。

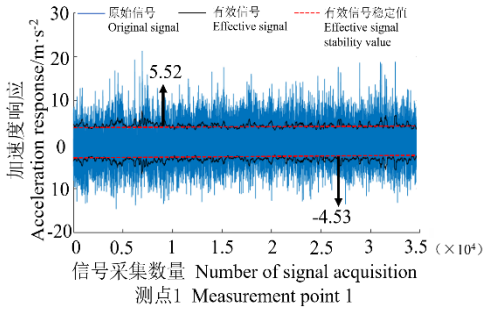
从图中可以发现：位于连接处下方的传感器加速度响应有效信号稳定值范围普遍大于连接处上方的传感器加速度响应有效信号稳定值范围。



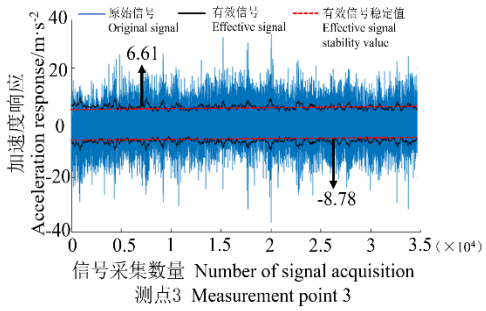


a. 半割幅工况下加速度响应

a. Acceleration response at half cut



测点1 Measurement point 1



测点3 Measurement point 3

b. 满割幅工况下加速度响应

b. Acceleration response at full cut

图8 满割幅和半割幅工况下田间试验振动信号

Fig.8 Field test vibration signals at full and half cut conditions

滤除采集到的突变加速度信号，并通过数据拟合获取该处的激励力范围。联合收割机不同割幅下加速度信号稳定值如表2所示。

表2 不同割幅下各测点有效信号稳定值

Tab.2 Acceleration response of each measurement point at different cutting amplitudes m·s^{-2}

测点 Measurement point	半割幅 Half-cut		满割幅 Full-cut	
	峰值 Peak	谷值 Valley	峰值 Peak	谷值 Valley
1	3.43	-4.34	5.52	-4.53
2	3.19	-4.71	5.59	-4.07
3	3.58	-8.23	6.61	-8.78
4	6.56	-6.71	9.12	-4.3

通过各测点有效信号稳定值的初步分析可以发现：随着喂入量的增加，加速度响应的峰谷值稳定值基本随之增加，故联合收割机在满割幅工况下外界对输送槽的激励力最大。

2.3.3外界激励*F*求解

输送槽所受外界激励*F*可近似为正弦函数，其幅值可以通过加速度信号的峰值稳定值与谷值稳定值确定；周期可以通过相对稳定区段任取两个时间区段的加速度信号峰值确定。通过对田间试验中采集到的信号分析可得：在24~26 s内相邻峰值之间的时间间隔约为0.121 s，而38~40s 内相邻峰值之间的时间间隔约为0.19 s，因此可得输送槽所受外界激励周期约为0.12 s。

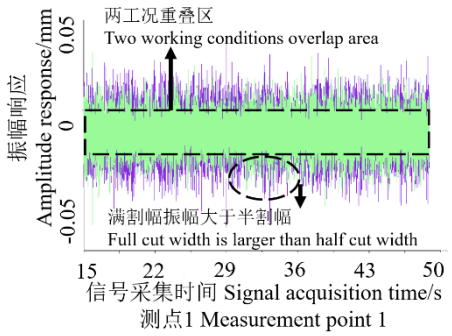
将输送槽质量、输送槽加速度周期和幅值(此处取满割幅工况下测点4信号峰值)代入式(1)可得，外界对联合收割机输送槽激励力的表达式为：

$$F = 1094.4\sin\frac{50}{3}\pi t \tag{19}$$

至此，经过上述分析与试验，悬臂输送槽阻尼摆模型中参数 k、c 和 F 已求解完毕。

2.4输送槽系统激振特性分析

为进一步分析悬臂输送槽系统的振动特性，对田间试验中半割幅和满割幅工况下输送槽与割台连接处4个测点加速度信号（15~50 s）的振幅响应及对应的频率成分进行分析，测点1和测点3的振幅响应及对应的频率如图9和图10所示：



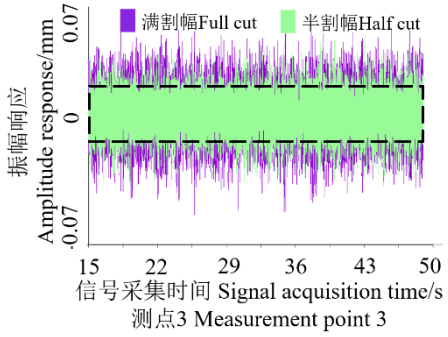


图9 不同工况下输送槽振幅响应

Fig.9 Amplitude response of conveyor chute under different operating conditions

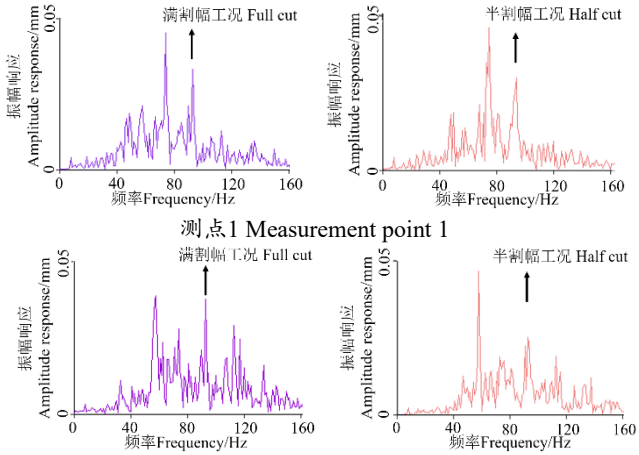


图10 不同工况下输送槽频谱图

Fig.10 Spectrum of conveyor trough under different working conditions

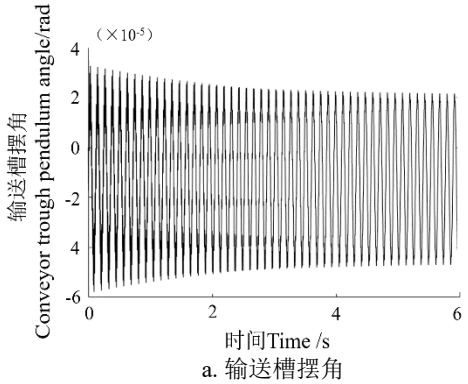
由图9可知：满割幅工况下的幅值略高于半割幅工况下的幅值。

由图10可知：1）悬臂输送槽连接处的振动频率主要集中在低频部分（50~100 Hz）且与作业工况及测点位置无关。2）输送槽振幅与割幅大小有关，而偏移量与传感器位置有关。且负方向位移偏差会随着传感器位置的下移而变大。

3 悬臂输送槽阻尼摆模型验证

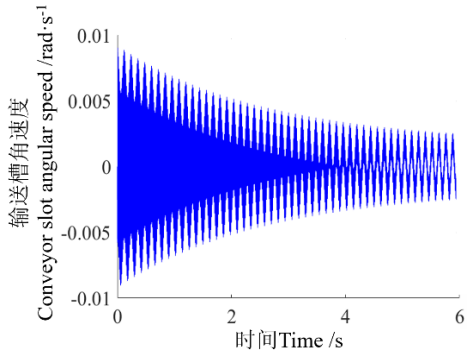
为验证前文悬臂输送槽阻尼摆模型的准确性，将联合收割机悬臂输送槽参数赋值给函数中相应的变量以分析输送槽在作业过程中外界激励对输送槽摆动的影响。田间试验中联合收割机悬臂输送槽系统初始参数如下：初始摆角 θ_1 为46.40 °，旋转轴与液压缸连接点的距离 b 为1 m，悬臂输送槽长度 l 为1.504 m。

利用MATLAB中ODE45函数处理悬臂输送槽阻尼摆模型，求解并绘制输送槽摆角随时间变化函数 $\theta(t)$ 与输送槽角速度随时间变化函数 $\dot{\theta}(t)$，如图10所示：



a. 输送槽摆角

a. Conveyor chute pendulum angle



b. 输送槽角速度

b. Conveyor chute angular velocity

图11 输送槽摆角与角速度

Fig.11 Conveyor chute angular velocity and pendulum angle change curves of measurement point 4

由图11可知：输送槽摆角和角速度变化量在前3秒减小，之后趋于稳定；输送槽摆角变化量的最大值约为9.05×10^{-5} rad，稳定状态下摆角变化量约为6.75×10^{-5} rad。

通过悬臂输送槽阻尼摆模型求解得到初始条件下输送槽4号测点摆幅为0.1310 mm，稳定状态下输送槽测点4摆幅为0.0979 mm。故在稳定状态下，理论求解的摆幅结果与试验获取的摆幅（0.099 mm）极为接近，误差约为1.11%。

为了减小模型验证的偶然性与误差，将满割幅工况下位于4号传感器同一水平的3号传感器数据代入微分方程中，结果表明：3号测点的输送槽角速度变化量和摆角变化量在前3秒内减小，之后趋于稳定。稳定状态下，理论推导的输送槽摆幅与试验获取的摆幅（0.1 mm）同样接近，误差约为4.30%。

4. 结论

本文基于拉格朗日法建立联合收割机悬臂输送槽阻尼摆模型，同时通过有限元法及田间试验等求解方程中未知参数，兼顾了输送槽受喂入量影响情况，并针对输送槽系统的激振特性进行分析，最终对满割幅工况下振幅响应较大的两个测点进行数据分析，得出以下结论：

1）随着喂入量的增加，割台与输送槽连接处加速度响应的峰谷值稳定值基本随之增加，故联合收割机在满割幅工况下外界对输送槽的激励力最大。

2）模型求解的联合收割机角速度和摆角变化量在前3秒内逐渐减小，之后趋于稳定，这主要是因为悬臂输送槽在初始时存在不稳定现象，与联合收割机的工作特性相符。

3）理论求解得到的悬臂输送槽下方测点摆幅在稳定条件下与试验结果较为接近，误差分别约

为1.11%与4.30%，验证了本研究建立的悬臂输送槽阻尼摆模型准确可行。

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Construction and validation of damped pendulum model for cantilever conveyor trough based on Lagrangian method

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Abstract: During the operation of rice combine harvester, the cantilever conveying trough will produce unbalanced swing, which leads to the problems of high failure rate and low harvesting efficiency of the whole machine. The purpose of this paper is to establish a damped pendulum model of the cantilever conveyor trough of the combine harvester based on the support of the hydraulic cylinder in order to accurately analyse the unbalanced pendulum phenomenon generated during operation. In this paper, a certain model of rice combine harvester is taken as the research object, and its cantilever conveying trough system is analysed by force and its dynamic model is established based on Lagrange equation. In order to determine the values of hydraulic cylinder stiffness k, damping c and total excitation F in the damped pendulum model of the cantilever conveyor trough, firstly, a static analysis of the hydraulic cylinder was carried out by using the modal analysis software Workbench, and the value of hydraulic cylinder stiffness k was solved to be 1.0236×10^8 N/m. Then, the modal analysis of the hydraulic cylinder was carried out, and it was learnt that its main vibration mode corresponded to the intrinsic frequency of 60.604 Hz, and the vibration acquisition

test was carried out for the hydraulic cylinder, and measurement points were set up at the front end and the rear end of the hydraulic cylinder. The hydraulic cylinder amplitude is obtained by integrating the acceleration signal, and the logarithmic attenuation rate of the amplitude is ln2. The intrinsic frequency and logarithmic attenuation rate of the amplitude are substituted into the damping formula to find out the damping c of the hydraulic cylinder, which is 228.58 N • s/m. Finally, half-width and full-width field harvesting tests were carried out to obtain the external excitation of the conveyor trough *F*. Four measurement points were set up at the connection between the cutting platform and the conveyor trough, and the acceleration signals collected during the field tests were processed using the quadratic integration method and the Fast Fourier Transform (FFT) to obtain the amplitude of the cantilevered conveyor trough and the main frequency components of vibration, so that the excitation characteristics were obtained and the basis for the verification of the subsequent dynamic differential equations. The excitation characteristics of the cantilever conveyor trough were obtained and provided the basis for the verification of the subsequent differential equations of the dynamics. After solving the parameters in the damped pendulum model of the cantilever conveyor trough, it is necessary to verify its accuracy. The ODE45 function in MATLAB is used to solve the equations of angular velocity and pendulum angle of the damped pendulum model of the cantilever conveyor trough, and the amplitudes of the pendulums of the measurement points 3 and 4 in the initial and steady state are solved, and compared with the amplitudes of the actual pendulums of the field test, which show that the theoretical solution of the amplitudes of the pendulums of measurement points 3 and 4 under the steady condition is very good. The results show that the theoretically solved pendulum amplitudes of measurement points 3 and 4 are very close to the test results under stable conditions, with errors of about 1.11% and 4.30%, respectively, which verifies the accuracy and feasibility of the damped pendulum model of the cantilever conveyor trough established in this study.
Keywords: model; simulation; pendulum; lagrangian equation; parameter calibration

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