稻麦联合收获开沟埋草多功能一体机行走及脱粒性能改进

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摘　要：针对自行研制的稻麦联合收获开沟埋草多功能一体机所存在的行走直线稳定性差、脱粒质量较低等问题。依据横向轴流式滚筒对稻麦秸秆的传递导流作用和二次复脱的原则，通过增设辅助滚筒的方法既改变出草口位置使开沟总成中移，消除整机偏向力，又延长秸秆在滚筒内的作用时间，提高谷物脱粒质量。其中：辅助滚筒总长为855 mm，导流角为18°，滚筒直径为452 mm，转速为1 350 r/min。性能测试表明：改进后自研一体机在0.27、0.58、0.85 m/s 3种不同工况下行走偏移度分别降低了 93.9%、94.4%、93.3%，行走直线稳定性显著提高；小麦和水稻总损失率分别降低 20.9%和11.8%，含杂率分别降低45.7%和21.4%。尽管水稻破碎率增加了7.4%，但脱粒的综合质量有较大提高。该研究增进了多功能一体机的适用性，为稻麦秸秆机械化集沟还田提供了参考。

关键词：农业机械；优化；设计；秸秆还田；行走偏移度；脱粒质量；轴流滚筒

doi：10.11975/j.issn.1002-6819.2015.18.005

中图分类号：S225.3　　**文献标志码：**A　　**文章编号：**1002-6819(2015)-18-0026-08

方志超，陈玉仑，丁为民，刘玉涛，秦　宽，李　雪. 稻麦联合收获开沟埋草多功能一体机行走及脱粒性能改进[J]. 农业工程学报，2015，31(18)：26－33.　doi：10.11975/j.issn.1002-6819.2015.18.005　http://www.tcsae.org

Fang Zhichao, Chen Yulun, Ding Weimin, Liu Yutao, Qin Kuan, Li Xue. Improvement of walking stability and threshing performance for harvest ditch and stalk-disposing machine[J]. Transactions of the Chinese Society of Agricultural Engineering (Transactions of the CSAE), 2015, 31(18): 26－33. (in Chinese with English abstract)　doi：10.11975/j.issn.1002-6819.2015.18.005　http://www.tcsae.org

0　引　言

秸秆还田是保护性耕作的重要内容[1-3]，能有效避免资源浪费和环境污染[4]。在国外很多国家采用秸秆直接还田来培肥地力等。德国波恩大学研究表明，每公顷施入6.5 t秸秆并补充氮肥，土壤有机质含量从1.02%增加到1.48%；英国洛桑试验站长期定位试验（Broadbald 试验和Hoosfield试验）表明秸秆持续还田的土壤有机质含量能明显提高[5]。在日本已立法来保证秸秆直接还田的有效执行[6]；而在国内，秸秆还田也得到相当重视，各地区都在探索适合本地区的秸秆还田模式与技术。针对年均秸秆产生量高达（16.5±0.75）t/hm^2[7]的江苏等稻麦两熟地区，常规还田难以做到全量直接还田，且存在碳汇作用受限等问题[8]。而秸秆集中入沟还田这一技术体系，它能将当季秸秆全部集于沟内且不影响作物的种植和生长[9]，对高产下的秸秆全量还田适应性极好，且能固碳减排[10-12]，所产生的秸秆留茬在还田中具有较高水分利用效率和经济效益[13]，且留茬高度的合理控制既能保证秸秆的有效利用[14]，对秸秆还田具有积极意义[15]。

针对秸秆集沟还田这一模式[16]，本课题组研制出稻麦联合收获开沟埋草多功能一体机[17-20]（简称自研一体机），该机一次下地完成收获、开沟、集草入沟等作业，实现了秸秆机械化入沟还田，且开沟深度由秸秆留茬高度所决定，当留茬越高时对应排出的秸秆量就越少，开沟深度也相应越浅，二者负相关。然而实践表明，自研一体机因收割机出草口右置使开沟总成也相应右偏，造成整机单侧承重，作业时直线行走性较差；同时，该机还存在脱粒不净，损失率和含杂率较高等问题。对此，本文在自研一体机的基础上增设一个辅助滚筒[21-22]，利用滚筒对稻麦秸秆的传递导流特性和二次复脱原理，解决直线行走稳定性差和收获质量低下的问题，并进行田间试验。以期为秸秆机械化集沟还田的推广提供参考。

1　原自研一体机分析

原自研一体机的主体部分-收割机（4LL-1.8型全喂入联合收割机）相关参数如表1所示，结构如图1所示。

表1　原自研一体机相关参数

Table 1　Initial parameters of multifunctional machine

原自研一体机参数 Initial parameters of multifunctional machine	数值 Numerical value	原自研一体机参数 Initial parameters of multifunctional machine	数值 Numerical value
外形尺寸 Boundary dimension/mm^3	5 300×2 250×2 350	主滚筒后部可用高度 Available height of main threshing cylinder /mm	539
整机质量 Total weight/kg	2 195	主滚筒后部可用宽度 Available width of main threshing cylinder/mm	527
配套动力 Matched power/kW	63	出草口与履带中心面间距 Distance from grass outlet to central surface of track /mm	573
主滚筒转速 Main threshing cylinder speed/(r·min^{-1})	1 180		

收稿日期：2015-07-22　　修订日期：2015-08-19

基金项目：国家科技支撑计划项目资助（2013BAD08B04）

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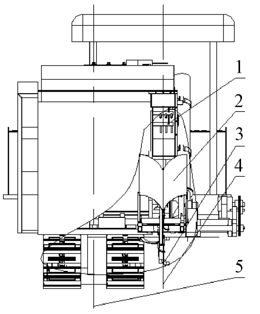


1.出草口　2.碎土板　3.开沟总成　4.出草口中心面　5.履带中心面
1.Grass outlet　2.Breaking plate　3.Ditcher assembly　4.Center surface of grass outlet　5.Central surface of track

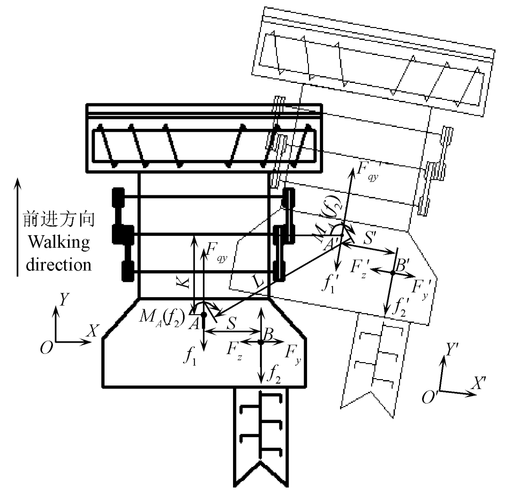
图 1　原自研一体机主视方向开沟总成位置示意图

Fig.1　Relative position diagram of original multifunctional machine ditch assembly in front view

图 2 为原自研一体机受力分析，由图 2 可知，当整机作业时，受到发动机的驱动力 F_{qy} 和摩擦力 f_1 的共同作用，开沟过程中刀片受到沟两端的反推力 F_z、F_y 和开沟阻力 f_2（刀盘反转，水平分力与机器前进方向相反），f_2 对收割机重心 A 产生一力矩 $M_A(f_2)$，当机器在前进方向位移为 K 时，在扭矩 $M_A(f_2)$的作用下，重心从 A 移至 A'，实际位移 $L>K$，机器发生侧向偏移，运动轨迹为曲线。

$$M_A(f_2) = f_2 \cdot S \tag{1}$$

式中：$M_A(f_2)$为 f_2 对整机重心 A 的扭矩，N·m；f_2 为开沟阻力，N；S 为开沟阻力到重心的垂直距离，m。



注：X,Y 分别为初始位置横、纵坐标；A,E 分别为收割机与开沟器初始位置的重心；F_{qy}，f_1 分别为收割机初始状态所受到的驱动力和地面摩擦力，N；$M_A(f_2)$为收割机初始位置重心所受力矩，N·m；F_z，F_y 和 f_2 分别为初始位置开沟器所受到沟面两侧向推力及开沟阻力，N；S 为初始位置开沟阻力到收割机重心 A 的距离，m；K,L 分别为整机理论位移和实际位移，m；$X', Y', A', E', F_{qy}', f_1', M'A(f_2), F_z', F_y', f_2', S'$分别为整机在末位置时的对应参数。
Note: X and Y are original coordinate axis; A and E are original gravity centers of harvester and ditching device, respectively; F_{qy} and f_1 are driving force and force of ground friction of harvester original state, respectively, N; $M_A(f_2)$ is moment of harvester original gravity center position, N·m; F_z, F_y, f_2 are thrust of ditch sides and ditching resistance, respectively, N; S is distance between f_2 and A, m; K and L are theoretical displacement of machine and practical displacement of machine, respectively, m; $X', Y', A', E', F_{qy}', f_1', M_A'(f_2), F_z', F_y', f_2', S'$ are corresponding parameters of whole machine at ultimate position, respectively.

图 2　原自研一体机俯视方向受力分析图

Fig.2　Top view force analysis diagram of original multifunctional machine

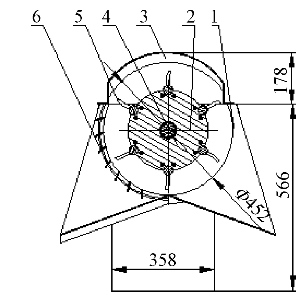
2　改进方案及部件设计

2.1　改进措施

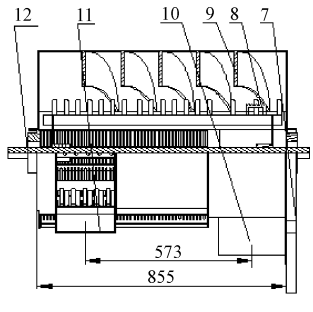
收割机的固有结构决定了开沟总成的位置，导致履带两侧受力不均产生差速现象而影响直线行走稳定性；单滚筒的联合收割机通常是通过调整滚筒转速或凹板间隙来提高脱净率，但这种调整方式容易造成成熟度高、饱满易脱的籽粒在反复打击下损伤[23-25]，造成损失率过大，杂质过多等问题。综合上述问题，采用横向轴流式辅助滚筒与主滚筒平行放置，构成双滚筒结构。辅助滚筒安装于主滚筒后侧，谷物清选风机上部，进草口与主滚筒出草口相连，排草位置与履带阻力中心面对齐，开沟总成相应中移至履带中心面。

2.2　总体设计

根据主滚筒后部可用空间的结构和相关尺寸对辅助滚筒相关参数进行设计[26]；以阻力中心面的位置确定双滚筒出草口中心距为 573 mm，辅助滚筒总长 L=855 mm，以主滚筒后部空间设计辅助滚筒直径 Φ=452 mm。各部分名称及主要尺寸如图 3 所示。



a. 辅助滚筒左视图
a. Left view diagram of deputy threshing cylinder



b. 辅助滚筒主视图
b. Front view diagram of deputy threshing cylinder

1.机架　2.滚筒　3.顶盖　4.辐板　5.脱粒钉齿　6.筛板　7.侧挡板　8.碎草刀　9.导草板　10.进草口　11.出草口　12.传动轴
1.Frame　2.Threshing cylinder　3.Tectum　4.Wheel disk　5.Threshing spike-tooth　6.Sieve plate　7.Side board　8.Cutter for breaking grass　9.Plate for guiding grass　10.Grass entrance　11.Grass outlet　12.Drive shaft

图 3　辅助滚筒结构示意图

Fig.3　Structure diagram of deputy threshing cylinder

双滚筒通过 B 型皮带相连进行动力传输。辅助滚筒排草能力不应小于主滚筒，以此为依据设计转速 ω=1 350 r/min。双滚筒装配关系如图 4 所示。

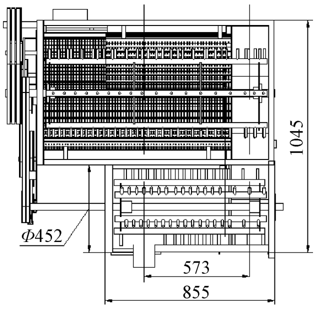


图 4　双滚筒装配关系示意图

Fig.4　Diagram of double threshing cylinder assembly relations

2.3　关键部件-导草装置

导草装置是实现秸秆导流和传递的关键，导草板的导流倾角控制着秸秆、谷粒等在脱粒室内的运动方向和速度[27]。导流角 θ 与秸秆轴向运动的线速度 v_l 的关系为：

$$v_l = v_r \cdot \tan\theta = \frac{v_{cir}}{v} \cdot 2\pi(R + \frac{h}{2}) \cdot n_d \cdot \tan\theta \qquad (2)$$

式中：v_l 为秸秆轴向运动的线速度，m/s；v_r 为秸秆径向运动的线速度，m/s；v_{cir} 为秸秆圆周运动的合速度，m/s；v 为辅助滚筒自身的线速度，m/s；R 为辅助滚筒的半径，mm；n_d 为辅助滚筒的转速，r/min；h 为辅助滚筒脱粒元件的高度，mm；θ 为导向板导流角，(°)。

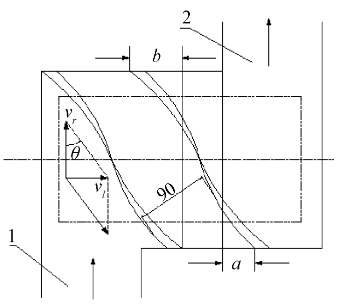
为保证秸秆在滚筒内不发生堵塞，单位时间内排出作物的体积不应小于喂入体积，则：

$$q \leqslant v_l \cdot s_d \cdot \rho = \frac{v_{cir}}{v} \cdot 2\pi(R + \frac{h}{2}) \cdot n_d \cdot \tan\theta \cdot s_d \cdot \rho \qquad (3)$$

$$\Rightarrow \theta \geqslant \arctan\left(\frac{q \cdot v}{2\pi v_{cir} \cdot (R + \frac{h}{2}) \cdot n_d \cdot s_d \cdot \rho}\right) \qquad (4)$$

式中：s_d 为茎秆压缩面积，s_d=163 564.0 mm^2；q 为茎秆喂入量，q=5.1 kg/s；ρ 为茎秆在辅助滚筒内的压缩密度，ρ=76.5 kg/m^3。

计算取整得导流角 θ=18°。一般导向板高约 20～30 mm，与钉齿齿顶间隙约 10 mm，导流角可适当调大以适应湿脱需要。设计时，在喂入口处有一块导向板横跨整个喂入口，如图 5 所示，且有一定重叠量（b≈50 mm），以免喂入口处返草。末端导向板应伸到出草口宽 1/3 处（a=100～150 mm），以保证顺利排草[28]。



1.进草口 2.出草口　1. Grass entrance　2. Grass outlet

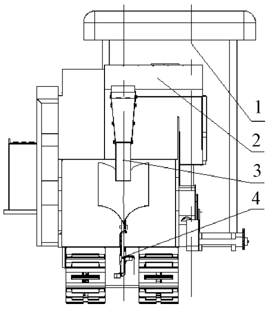
注：a 为末尾导向板伸到出草口的宽度，m；b 为喂入口处导向板的重叠量，m；θ 为导向板导流角，(°)；v_l 为秸秆轴向运动的线速度，m·s^{-1}；v_r 为秸秆径向运动的线速度，m·s^{-1}。

Note: a is distance from guide plate end to grass outlet, m; b is overlap of guide plate, m; θ is diversion angle, (°); v_l is straw axial linear velocity of movement, m·s^{-1}; v_r is straw radial motion of linear velocity, m·s^{-1}.

图 5　导草板排列及夹角示意图

Fig.5　Arrangement and angles of grass guide plate

改进后出草口中心面与履带中心面重合，开沟装置中移。自研一体机主视图如图 6 所示。



1.原出草口中心面　2.辅助副滚筒　3.现出草口中心面　4.开沟装置
1.Primary central surface of grass outlet　2.Deputy threshing cylinder　3.Improved central surface of grass outlet　4.Ditcher assembly

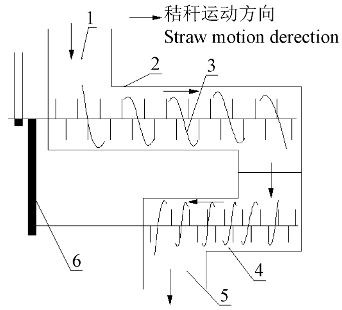
图 6　改后整机主视方向开沟总成位置示意图

Fig.6　Relative position diagram of improved multifunctional machine ditch assembly in front view

3　改后整机分析

3.1　秸秆运动轨迹分析

作业过程中，秸秆经输送装置进入主滚筒，在各部件的共同作用下呈间歇性螺旋运动；加辅助滚筒后，秸秆从主滚筒出草口由过渡连接处在惯性作用和辅助滚筒的抓取下进入辅助滚筒内并反向运动，在导向板的法向推送力和钉齿的共同作用下最终被送至辅助滚筒出草口排出，如图 7 所示，出草位置由辅助滚筒长度决定。



1.进草口　2.主滚筒　3.秸秆运动轨迹　4.副滚筒　5.出草口　6.传动皮带
1.Grass entrance　2.Main threshing cylinder　3.Straw trajectory　4.Deputy threshing cylinder　5.Grass outlet　6.Drive belts

图 7　加装辅助滚筒后秸秆运动轨迹图

Fig.7　Different straw trajectory charts compared machine with deputy threshing cylinder and machine without deputy threshing cylinder

3.2　重心测试

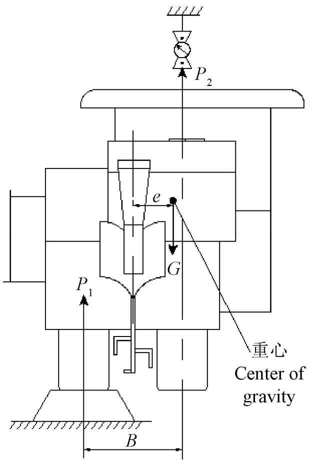
为考察改进后多自研一体机的稳定性，应对整机的重心位置进行测定[29]，如图 8 所示。

整机在水平位置时，测定右侧履带支撑力 P_1，左侧履带 P_2 的支承点可用测力计测定吊挂点的拉力，重心离履带中心面的距离 e 根据式（5）计算：

$$e = \frac{P_2 - 0.5G}{G} \cdot B \qquad (5)$$

式中：e 为中心面至重心的距离，m；G 为整机所受重力，G=21 511 N；P_2 为左侧吊挂力，P_2=10 858 N；B 为两侧履带间距，B=1.2 m。

通过计算得 e=0.0057 m，整机横向上重心与阻力中心面可认为重合，两侧履带受力较均，开沟总成对整机横向稳定性影响很小。

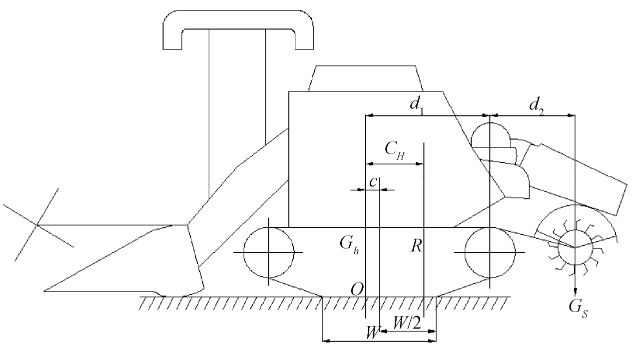


注：P_1 为右侧履带支撑力，N；P_2 为左侧吊挂力，N；G 为整机所受重力，N；e 为中心面至重心的距离，m；B 为两侧履带间距，m。
Note: P_1 is on supporting force from right side track, N; P_2 is on left hanging force, N; G is gravity of machine, N; e is distance between gravity center and surface center, m; B is interval of tracks, m.

图 8　改后自研一体机主视方向重心位置示意图
Fig.8　Gravity center of improved multifunctional machine in front view

3.3　纵向稳定性分析计算

为保证稻麦联合收获开沟埋草多功能一体机的正常作业，需对机组纵向稳定性进行校核[29]，如图 9 所示。



注：G_h 为收割机质量，kg；G_S 为开沟总成质量，kg；R 为地面支撑力，N；W 为履带与地面接触长度，m；d_1, d_2 分别指收割机和开沟总成的重心坐标，m；c 指履带支承面中心到收割机重心的距离，m；C_H 指一体机压力中心相对于收割机重心的纵向位移，m；O 为中心面接触点。
Note: G_h is harvesters weight, kg; G_S is ditching assembly weight, kg; R is ground support force, N; W is track length of contact with the ground, m; d_1 and d_2 respectively, harvester and furrowing assembly's center of gravity coordinate, m; c is track bearing surface from center to center of gravity harvester, m; C_H is multifunctional machine center of pressure relative to gravity center of harvester longitudinal displacement, m; O is center surface contact points.

图 9　改后自研一体机左视方向纵向稳定性测试示意图
Fig.9　Diagram of longitudinal stability test of improved multifunctional machine in left view

压力中心的纵向位移系数 m 的计算由式（6）确定：

$$m=\frac{C_H-c}{W} \tag{6}$$

式中：m 为压力中心的纵向位移系数；C_H 指一体机压力中心相对于收割机重心的纵向位移，m；c 指履带支承面中心到收割机重心的距离，c=0.92 m；W 指履带与地面接触长度，W=2.1 m。

分析上式可知，当 m=1/6 时，履带支承面上的压力分布图由原来的梯形转变为三角形。若由开沟总成质量决定的 C_H 的值进一步增大，支承面前端便不再与地面接触。为保证履带与地面间良好接触附着，纵向位移系数 m 理论上不应大于 1/6。压力中心位移 C_H 的计算根据 O 点的力矩平衡方程式：

$$C_H=\frac{G_h(d_1+d_2)}{R}=\frac{G_h(d_1+d_2)}{G_s+G_h}=\frac{\delta}{1+\delta}(d_1+d_2) \tag{7}$$

式中：G_h 为收割机质量，G_h=2 006 kg；G_S 为开沟总成质量，G_S=189 kg；δ 为开沟总成与收割机的质量比；d_1 和 d_2 指收割机和开沟总成的重心坐标，d_1=0.9 m，d_2=0.4 m。

将已知条件代入式（6）、式（7）得 C_H=1.19 m，m=0.128。对于履带收割机压力中心的纵向位移系数的最大允许值为[m]=0.2[28]，该机组 m≤[m]，则整机的纵向稳定性符合要求。

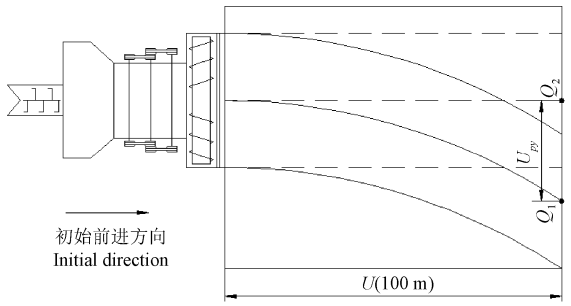
4　试验与结果分析

4.1　试验条件

2013 年 6 月 15 号和 2013 年 11 月 6 号在江苏省盐城市黄海农场二分场进行田间试验，试验田面积为 33.3×160 m^2。土质为黏土，0～10，>10～20 cm 深的土壤坚实度分别为 168.4、316.5 N。种植方式为稻麦轮作，免耕直播，一年两熟。试验作物：小麦（淮麦-22，小麦平均茎秆长度 75.2 cm，籽粒千粒质量 34.84 g，草谷比为 1.2，籽粒含水率 15.95%，茎秆含水率 27.44%）；水稻（盐粳-311，自然高度为 1.2 m，单产 7 500 kg/hm^2，籽粒含水率为 20.8%，茎秆含水率为 67.2%）。

4.2　试验方法

参考液压挖掘机行走直线性试验方法和 GB/T 5982-2005《脱粒机试验方法》，对改进后的稻麦联合收获开沟埋草多功能一体机行走直线性和收获脱粒质量的相关指标进行测试，图 10 为行走直线稳定性测试。



注：U 为试验长度，m；U_{py} 为偏移距离，m；Q_1、Q_2 分别为实际停止点和理论停止点。
Note: U is length of test, m; U_{py} is offset distance, m; Q_1 and Q_2 are actually stop point and theory stop point.

图 10　机器直线行走稳定性测试俯视图
Fig.10　Straight walking stability test of multifunctional machine in top view

取与田埂方向平行的 100 m 作为单次试验长度，过

终点并垂直于理论前进方向的直线为机器停止线，标定机器的试验理论停止点 Q_2。机器在不调整操纵手柄的情况下通过试验区，记录实际停止点 Q_1 与理论点 Q_2 的距离作为机器偏移量，同一油门下低速档（0.27 m/s）、中速挡（0.58 m/s）、高速档（0.85 m/s）各重复 3 次取均值。

在收获质量的测试试验中，待机器空载运转正常后，均匀喂入试验物料，待脱粒室内的物料流稳定后开始取样，取样时间不应小于 20 s。相关计算公式如下：

$$Z_{py} = \frac{U_{py}}{U} \times 100\% \tag{8}$$

式中：Z_{py} 为机器偏移度，%；U 为试验长度，m；U_{py} 为偏移距离，m。

$$Z_Z = \frac{W_{xz}}{W_x} \times 100\% \tag{9}$$

式中：Z_Z 为含杂率，%；W_{xz} 为样品杂质质量，g；W_x 为样品总质量，g。

$$Z_p = \frac{W_p}{W_x} \times 100\% \tag{10}$$

式中：Z_p 为破碎率，%；W_p 为样品破碎籽粒质量，g。

$$\begin{aligned} S_z &= S_w + S_q + S_j + S_f \\ &= \left(\frac{W_w}{W_{wz}} + \frac{W_q}{W_{qz}} + \frac{W_j}{W_{jz}} + \frac{W_f}{W_{fz}}\right) \times 100\% \end{aligned} \tag{11}$$

式中：S_z 为总损失率，%；S_w 为样品未脱净损失率，%；S_q 为样品清选损失率，%；S_j 为样品夹带损失率，%；S_f 为样品飞溅损失率，%；W_w 为样品未脱净损失质量，g；W_q 为样品清选损失质量，g；W_j 为样品夹带损失质量，g；W_f 为样品飞溅损失质量，g；W_{wz}、W_{qz}、W_{jz}、W_{fz} 分别为对应样品的总质量，g。

4.3　试验结果与分析

表 2 为机具改进前后主要性能参数对比，加装辅助滚筒使开沟总成同步中移后，由于辅助滚筒转速高于主滚筒，且半径和长度较主滚筒小，因此单位时间内对物料的排出能力要强于主滚筒。试验表明：正常作业时双滚筒内排草连贯、良好，不会出现拥塞及滞留秸秆的现象。3 种不同工况下（低速 0.27 m/s、中速 0.58 m/s、高速 0.85 m/s）偏移度降幅分别为 93.9%、94.4%、93.3%，偏移度显著下降。通过多重比较可知改进前机器速度与偏移度有显著差异（低速与中速间 P=0.034、低速和高速间 P<0.01、中速与高速间 P=0.01），速度越高偏移度相对越低，这是由于偏移量与时间呈正相关所致。而改进后细微的偏移度与速度无明显关系（P>0.05）。这是由于整机偏向力消除后履带受力均匀，轻微偏移量主要是由于田间不平等误差造成；对稻麦的脱粒性能进行比较分析可知，小麦和水稻秸秆的总损失率（未脱净率、夹带损失率、清选损失率和飞溅损失率）分别减少了 20.9%和 11.8%，含杂率分别降低了 45.7%和 21.4%，且二者含杂率在改进前后下降均达到显著差异（改进前 P=0.016，改进后 P=0.036）。尽管水稻的破碎率增加了 7.4%，但自研一体机的综合收割质量有所提高。

表 2　机具改进前后主要性能参数对比

Table 2　Performance parameters contrast between original machine and improved machine

性能对比 Performance comparison	不同工况 Different working conditions	机器偏移度 Machine offset degree/m	小麦 Wheat 总损失率 Total loss rate/%	小麦 Wheat 含杂率 Dirt rate/%	水稻 Rice 总损失率 Total loss rate/%	水稻 Rice 含杂率 Dirt rate/%	水稻 Rice 破碎率 Damage rate/%
改进前 Original	低速 Low speed (0.27 m·s^{-1})	21.4±0.60a	4.3±0.36a	3.5±0.30a	5.1±0.36a	4.2±0.21a	2.7±0.21a
	中速 Medium speed (0.58 m·s^{-1})	19.6±1.10b					
	高速 High speed (0.85 m·s^{-1})	16.3±0.25c					
改进后 Improved	低速 Low speed (0.27 m·s^{-1})	1.3±0.10d	3.4±0.15a	1.9±0.26b	4.5±0.17a	3.3±0.20b	2.9±0.25a
	中速 Medium speed (0.58 m·s^{-1})	1.1±0.20d					
	高速 High speed (0.85 m·s^{-1})	1.1±0.06d					
变化率/% Change rate	低速 Low speed (0.27 m·s^{-1})	93.9	20.9	45.7	11.8	21.4	7.4
	中速 Medium speed (0.58 m·s^{-1})	94.4					
	高速 High speed (0.85 m·s^{-1})	93.3					

注：根据 Duncan 多重比较，同一列中不同处理后的不同字母表示差异显著（P<0.05）。

Note: According to multiple comparison, different dispose with different letters in same column represent significant difference (P<0.05).

5　结　论

1）辅助滚筒对秸秆的传递导流作用能有效改变出草口位置而解决自研一体机直线行走性差的问题。在 0.27、0.58、0.85 m/s 3 种工况下降幅分别达到 93.9%、94.4%、93.3%，与改进前差异显著（P<0.01），偏移度大大下降，直线行走稳定性好，且不受速度影响；同时辅助滚筒的二次复脱功能使小麦含杂率和损失率分别降低了 45.7%和 20.9%，水稻降低了 21.4%和 11.8%，破碎率略有增加，整机的综合收获质量得到了一定的提高。

2）辅助滚筒的应用较好的解决了上述存在的 2 个问题，既提高了稻麦联合收获开沟埋草多功能一体机的适用性，也为稻麦轮作下机械化秸秆集沟还田这一方式提供了参考。

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Improvement of walking stability and threshing performance for harvest ditch and stalk-disposing machine

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Abstract: In rice-wheat double-cropping area, the top soil is compactible and the soil disturbance frequency is relatively lower, which eventually results in soil hardening due to the implementation of lacking tillage or no tillage. In order to increase soil disturbance frequency and improve soil environment based on conservation tillage, and solve excessive straw yield in full amount of straw returning under high-yield production, a new straw returning model named straw-returning-to-ditch was proposed. According to operation characteristics of this model, a multifunction machine combining harvester and ditch opener was designed. This machine could complete harvesting, ditching and stalk-disposing at one time, and realize multiple assignments with high efficiency. The ditching device of the multifunctional machine was biased in line with the unilateral grass outlet, which should however be aligned with the grass outlet to ensure the rice straw to be collected into the ditch reliably. So, after long-term practice, the assignment of the ditching device led to unilateral bearing, poor straight walking performance and running deviation for this machine, which was difficult to manipulate. Meanwhile, the machine still had some threshing problems, for example, under threshing and high loss rate in the process of harvesting. Therefore, based on the principle of transfer guide function and secondary threshing of the lateral cylinder, an assisted threshing cylinder that was located at the rear of the main threshing cylinder was equipped, which constituted a double threshing cylinder structure. And the grass inlet was connected with the main threshing cylinder. The double threshing cylinder structure could effectively solve the problem of the machine unilateral bearing and low harvest quality through changing the position of grass outlet, making the ditching device located in the middle surface of the track to eliminate deflecting force, and increasing the threshing time to improve harvesting quality. Combined with the characteristics and the available space in the rear of the main threshing cylinder, the overall length and diameter of the assisted threshing cylinder were 855 and 452 mm respectively, the diversion angle of cylinder end cover plate was 18°, and the angular speed was 1 350 r/min. The modified grass outlet and crawler surface were kept on the same vertical plane, and the distance between their centers was 573 mm. The performance test showed that the walking position deflections of modified machine under 3 different working speeds (low speed 0.38 m/s, medium speed 0.56 m/s and high speed 0.74 m/s) were reduced by 93.9%, 94.4% and 93.3%, respectively. Furthermore, the straight walking stability and control performance were significantly improved. In the harvest process, the total loss rates of wheat and rice were decreased by 20.9% and 11.8%, respectively, while the impurity rates were decreased by 45.7% and 21.4% respectively. The comprehensive performance of rice threshing was enhanced although the broken rate was increased by 7.4%; and meanwhile the broken rate could be acceptable for the use of harvesting machine. The result indicated that the double threshing cylinder structure was feasible for solving the problems of the machine. This research has improved the applicability of the multifunctional machine and provided the reference for the mechanical returning of the collected straw to the ditch.

Key words: agricultural machinery; optimization; design; stalk returning field; walking migration rate; threshing quality; axial flow cylinder