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拖拉机驾驶室悬架油气弹簧设计与试验*

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摘要: 基于 CF700 型拖拉机驾驶室悬架参数要求和流体力学理论建立了一种油气弹簧的弹性力及阻尼力模型。计算了油气弹簧关键参数,设计了阻尼可调的油气弹簧。试验研究了激励、节流阀开度及单向阀开度对油气弹簧输出力的影响。试验结果表明,所研制的油气弹簧有较大的阻尼力调节范围,节流阀开度同时对油气弹簧压缩与复原行程输出力有影响,单向阀开度只对压缩行程输出力有影响,节流阀开度对输出力影响较单向阀明显。试验验证了设计思路和方法的有效性,为拖拉机驾驶室油气悬架减振性能的研究奠定了基础。

关键词: 拖拉机　驾驶室悬架　油气弹簧　设计　试验

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引言

油气弹簧具有非线性变刚度、结构紧凑、悬架锁止、水平调节及阻尼可调等优点,已广泛用于非公路车辆悬架[1-5]。研究表明,油气悬架可有效提高拖拉机通过性与行驶速度,改善拖拉机的乘坐舒适性及操纵稳定性。国外对油气弹簧及油气悬架进行了大量研究[6-13]。国内也对油气悬架进行了大量理论及试验研究,但基本上集中在车身悬架系统,有关驾驶室油气悬架的研究却鲜有报道[14-15]。本文设计一种节流阀及单向阀开度可调的拖拉机驾驶室用油气弹簧,研究节流阀及单向阀开度对输出力的影响。

1　油气弹簧刚度和阻尼

以 CF700 型拖拉机为研究对象,驾驶室悬架为全浮式,其前、后悬架的左右支撑点均采用油气弹簧作为弹性——阻尼元件,其 4 自由度平面振动模型如图 1 所示。建立系统的振动微分方程如下

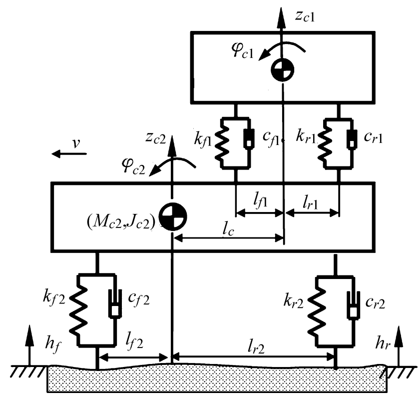

图 1　带全浮式驾驶室的拖拉机振动模型

Fig. 1　Vibration model of full-floating cab tractor

$$
\begin{aligned}
M_{c1}\ddot{z}_{c1} = {} & c_{f1}[-\dot{z}_{c1} + l_{f1}\dot{\varphi}_{c1} + \dot{z}_{c2} + (l_c - l_{f1})\dot{\varphi}_{c2}] + \\
& c_{r1}[-\dot{z}_{c1} - l_{r1}\dot{\varphi}_{c1} + \dot{z}_{c2} + (l_c + l_{r1})\dot{\varphi}_{c2}] + \\
& k_{f1}[-z_{c1} + l_{f1}\varphi_{c1} + z_{c2} + (l_c - l_{f1})\varphi_{c2}] + \\
& k_{r1}[-z_{c1} - l_{r1}\varphi_{c1} + z_{c2} + (l_c + l_{r1})\varphi_{c2}]
\end{aligned} \tag{1}
$$

$$
\begin{aligned}
J_{c1}\ddot{\varphi}_{c1} = {} & c_{f1}[\dot{z}_{c1} - l_{f1}\dot{\varphi}_{c1} - \dot{z}_{c2} - (l_c - l_{f1})\dot{\varphi}_{c2}]l_{f1} - \\
& c_{r1}[\dot{z}_{c1} + l_{r1}\dot{\varphi}_{c1} - \dot{z}_{c2} - (l_c + l_{r1})\dot{\varphi}_{c2}]l_{r1} + \\
& k_{f1}[z_{c1} - l_{f1}\varphi_{c1} - z_{c2} - (l_c - l_{f1})\varphi_{c2}]l_{f1} - \\
& k_{r1}[z_{c1} + l_{r1}\varphi_{c1} - z_{c2} - (l_c + l_{r1})\varphi_{c2}]l_{r1}
\end{aligned} \tag{2}
$$

$$
\begin{aligned}
M_{c2}\ddot{z}_{c2} = {} & c_{f1}[\dot{z}_{c1} - \dot{z}_{c2} - l_{f1}\dot{\varphi}_{c1} - (l_c - l_{f1})\dot{\varphi}_{c2}] + \\
& c_{r1}[\dot{z}_{c1} - \dot{z}_{c2} + l_{r1}\dot{\varphi}_{c1} - (l_c + l_{r1})\dot{\varphi}_{c2}] + \\
& k_{f1}[z_{c1} - z_{c2} - l_{f1}\varphi_{c1} - (l_c - l_{f1})\varphi_{c2}] + \\
& k_{r1}[z_{c1} - z_{c2} + l_{r1}\varphi_{c1} - (l_c + l_{r1})\varphi_{c2}] + \\
& c_{f2}(-\dot{z}_{c2} + l_{f2}\dot{\varphi}_{c2} + \dot{h}_f) + c_{r2}(-\dot{z}_{c2} - l_{r2}\dot{\varphi}_{c2} + \dot{h}_r) + \\
& k_{f2}(-z_{c2} + l_{f2}\varphi_{c2} + h_f) + k_{r2}(-z_{c2} - l_{r2}\varphi_{c2} + h_r)
\end{aligned} \tag{3}
$$

$$
\begin{aligned}
J_{c2}\ddot{\varphi}_{c2} = {} & c_{f1}[\dot{z}_{c1} - \dot{z}_{c2} - l_{f1}\dot{\varphi}_{c1} - (l_c - l_{f1})\dot{\varphi}_{c2}](l_c - \\
& l_{f1}) + c_{r1}[\dot{z}_{c1} - \dot{z}_{c2} + l_{r1}\dot{\varphi}_{c1} - (l_c + l_{r1})\dot{\varphi}_{c2}](l_c + l_{r1}) + \\
& k_{f1}[z_{c1} - z_{c2} - l_{f1}\varphi_{c1} - (l_c - l_{f1})\varphi_{c2}](l_c - l_{f1}) + \\
& k_{r1}[z_{c1} - z_{c2} + l_{r1}\varphi_{c1} - (l_c + l_{r1})\varphi_{c2}](l_c + l_{r1}) - \\
& c_{f2}(-\dot{z}_{c2} + l_{f2}\dot{\varphi}_{c2} + \dot{h}_f)l_{f2} + c_{r2}(-\dot{z}_{c2} - l_{r2}\dot{\varphi}_{c2} + \dot{h}_r)l_{r2} - \\
& k_{f2}(-z_{c2} + l_{f2}\varphi_{c2} + h_f)l_{f2} + k_{r2}(-z_{c2} - l_{r2}\varphi_{c2} + h_r)l_{r2}
\end{aligned} \tag{4}
$$

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式中　k_{f2}、k_{r2}——前、后轮刚度，N/m

c_{f2}、c_{r2}——前、后轮阻尼系数，N·s/m

M_{c2}——机身质量，kg

J_{c2}——机身绕质心转动惯量，kg·m²

k_{f1}、k_{r1}——驾驶室前、后悬架的刚度，N/m

c_{f1}、c_{r1}——驾驶室前、后悬架阻尼系数，N·s/m

M_{c1}——驾驶室质量，kg

J_{c1}——驾驶室绕质心转动惯量，kg·m²

l_{f1}、l_{r1}——驾驶室质心到前、后悬架的水平距离，m

l_{f2}、l_{r2}——机身质心到前、后轮轴的水平距离，m

l_c——驾驶室质心到机身质心的水平距离，m

h_f、h_r——前、后轮路面激励，m

在固有频率为 0.8 ~ 2.5 Hz、阻尼比为 0.15 ~ 0.35 及悬架动行程为 ±50 mm 条件下，利用微分方程对振动模型进行驾驶室悬架参数匹配研究。得到驾驶室前、后悬架刚度为 20、20 kN/m，阻尼系数为 1 233、1 726 N·s/m。前、后支点单个油气弹簧刚度为 10、10 kN/m，阻尼系数为 617、863 N·s/m。

2　油气弹簧设计与计算

2.1　油气弹簧结构设计

根据驾驶室悬架参数优化设计结果，设计了驾驶室悬架的油气弹簧，其结构如图 2 所示[16]，该油气弹簧主要由液压油缸、节流阀、单向阀组及隔膜式蓄能器组成。其特点是节流阀和单向阀组的开度可调，以适应拖拉机行驶工况的变化。

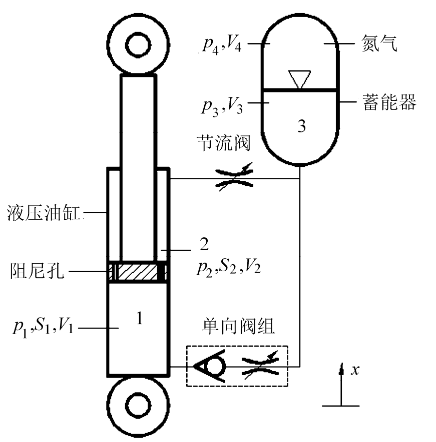

图 2　阻尼可调油气弹簧原理图

Fig. 2　Principle model of damping adjustable hydro-pneumatic spring

2.2　弹性力模型

当活塞杆相对缸体的位移为 x 时，忽略液压油的可压缩性，此时无杆腔 1 容积变化为 $\Delta V_1 = S_1 x$，有杆腔 2 容积变化为 $\Delta V_2 = S_2 x$，3 腔和氮气气室的容积变化均为 $\Delta V = (S_1 - S_2)x$，4 个腔压力从 p_0 变化为 p。缓慢移动时工作介质温度变化忽略不计，按理想气体状态方程 $pV^r = p_0 V_0^r$，则油气弹簧内部压力及静态弹性力可表示为

$$p = \frac{p_0 V_0^r}{V^r} = \frac{pV_0^r}{[V_0 - (S_1 - S_2)x]^r} \tag{5}$$

$$F = p(S_1 - S_2) = \frac{p_0 V_0^r (S_1 - S_2)}{[V_0 - (S_1 - S_2)x]^r} \tag{6}$$

式中　p——压缩后系统压力

p_0——蓄能器初始充气压力

V_0——蓄能器初始充气体积

r——气体多变指数，静态时取 $r = 1$

2.3　阻尼力模型

在油气弹簧压缩或复原时，油缸与活塞杆间的粘滑摩擦力 $F(\dot{x})$ 数学模型为[17]

$$F(\dot{x}) = \mathrm{sign}(\dot{x})[F_c + (F_s - F_c)e^{-(\dot{x}/\dot{x}_s)^2} + F_v \dot{x}] \tag{7}$$

式中　F_c——库仑摩擦力，F_c 为 100 N

F_s——静摩擦力，F_s 为 220 N

$\dot{x}$——油缸与活塞杆相对速度

$\dot{x}_s$——经验参数，$\dot{x}_s$ 为 0.003 m/s

F_v——粘性摩擦系数，F_v 为 2 N·s/m

根据细长孔理论，节流阀两端压力与流量间关系

$$Q_j = \frac{\pi d_j^4}{128\mu l}\Delta p_{23} \tag{8}$$

式中　Q_j——流经节流阀的液压油流量，m³/s

μ——液压油动力粘度，Pa·s

d_j——节流阀孔口通流直径，m

l——孔口通流长度，m

Δp_{23}——蓄能器与有杆腔液体的压力差，Pa

根据厚壁孔理论，单向阀两端压力与流量间关系为

$$Q_d = C_Q S_d \sqrt{2\frac{\Delta p_{31}}{\rho}} \tag{9}$$

式中　Q_d——流经单向阀的液压油流量，m³/s

C_Q——孔口流量系数，C_Q 为 0.82

S_d——单向阀通流面积，m²

Δp_{31}——油气悬架无杆腔与蓄能器液体的压力差，Pa

ρ——油液密度，kg/m³

根据厚壁孔理论，阻尼孔两端压力与流量间关系为

$$Q_z = C_Q S_z \sqrt{2\frac{\Delta p_{21}}{\rho}} \tag{10}$$

式中 Q_z——流经阻尼孔的液压油流量，m^3/s

S_z——阻尼孔通流面积，m^2

Δp_{21}——油气悬架无杆腔与有杆腔间的液体压力差，Pa

由蓄能器内油液变化量与油气悬架缸筒内油液变化量相等可得式(8)～(10)中各流量间的关系为

$$\begin{cases} S_2\dot{x} = \mathrm{sign}(\dot{x})nQ_z + Q_j \\ (S_1 - S_2)\dot{x} = \left[\dfrac{1}{2} + \dfrac{1}{2}\mathrm{sign}(\dot{x})\right]Q_d - Q_j \end{cases} \tag{11}$$

式中 n——阻尼孔个数

由式(8)～(11)得

$$\begin{cases} S_2\dot{x} = \mathrm{sign}(\dot{x})nC_QS_z\sqrt{\dfrac{2}{\rho}(p_1 - p_2)\mathrm{sign}(\dot{x})} + \\ \quad \dfrac{\pi d_j^4}{128\mu l}(p_3 - p_2) \\ (S_1 - S_2)\dot{x} = C_QS_d\left[\dfrac{1}{2} + \dfrac{1}{2}\mathrm{sign}(\dot{x})\right]\cdot \\ \quad \sqrt{\dfrac{2}{\rho}(p_1 - p_3)\mathrm{sign}(\dot{x})} - \dfrac{\pi d_j^4}{128\mu l}(p_3 - p_2) \end{cases} \tag{12}$$

式中 p_3 由蓄能器气体状态方程得到。

由油气弹簧输出力[18]

$$F = p_1S_1 - p_2S_2 \tag{13}$$

得到不考虑摩擦力时的阻尼力

$$F_c' = S_2(p_1 - p_2) \tag{14}$$

考虑摩擦力后阻尼力

$$F_c = S_2(p_1 - p_2) + F(\dot{x}) \tag{15}$$

2.4 设计计算

将油缸内径设计为 32 mm，活塞杆直径设计为 18 mm[10]，油气弹簧行程设计为 100 mm。根据油气弹簧静平衡条件

$$p_0(S_1 - S_2) = Mg \tag{16}$$

得到蓄能器初始充气压力。

由式(6)求导，得到油气弹簧刚度为

$$K(x) = F' = \frac{rp_0V_0^r(S_1 - S_2)^2}{[V_0 - (S_1 - S_2)x]^{r+1}} \tag{17}$$

将已知参数代入式(17)计算得到油气弹簧蓄能器初始充气体积。

根据式(12)与(15)建立油气弹簧阻尼孔、单向阀及节流孔的 Matlab/Simulink 计算模型，如图 3 所示。图中左侧为正弦激励，子系统分别为方程组(12)及式(7)的模型，右侧为输出量，分别为速度与阻尼力关系及位移与阻尼力关系。根据已知参数及油气弹簧阻尼系数可计算得到单个阻尼孔通径及阻尼孔个数、单向阀组通流直径、节流阀通流直径。

通过上述计算确定了油气弹簧的主要参数，如

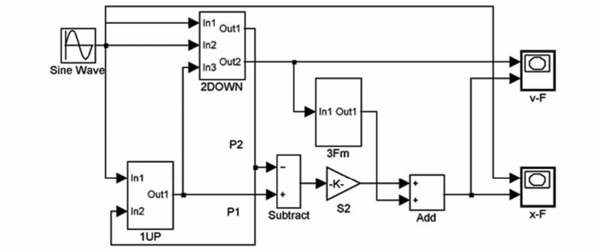

图 3 阻尼力计算模型

Fig. 3 Model of damping force calculating

表 1 所示。其中节流阀和单向阀组通流直径 0～6 mm 可调，从而实现油气弹簧阻尼调节的功能。

表 1 油气弹簧主要参数

Tab. 1 Hydro-pneumatic spring key parameters

参数	数值
液压油缸内径/mm	32
活塞杆直径/mm	18
油气弹簧行程/mm	100
阻尼孔直径/mm	2
阻尼孔数量	5
节流阀通流直径/mm	0～6
单向阀通流直径/mm	0～6
蓄能器初始充气体积/cm^3	43
蓄能器初始充气压力/MPa	4.8

3 油气弹簧性能试验

将油气弹簧、力传感器及位移传感器安装在试验台上，如图 4 所示。试验台激励幅值与频率可调，由 LMS 系统采集记录力与位移信号。

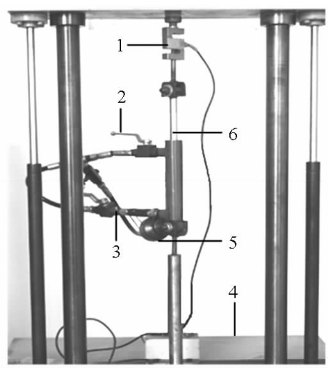

图 4 油气弹簧性能测试

Fig. 4 Hydro-pneumatic spring test

1. 力传感器 2. 节流阀 3. 单向阀组 4. 激振台 5. 蓄能器 6. 液压缸

3.1 试验设计

分别对油气弹簧进行静态和动态试验研究，具体方案见表 2。

(1) 进行静态试验以验证油气弹簧刚度特性，试验台移动 10 mm 记录一次数据。

(2) 在不同激励幅值和频率下进行动态试

验[19]。设置激励振幅 10 mm 及 30 mm，分别调整激励频率、节流阀及单向阀开度，采集力与位移数据。节流阀和单向阀开度 0°时通流直径为 0，90°时通流直径最大。

表 2　试验设计

Tab. 2　Experiment design

幅值/mm	频率/Hz	节流阀开度/(°)	单向阀开度/(°)
50		90	90
30	1,2,3,4	35	30,35,40
		30,40	35
10	1,2,3,4	35	30,35,40
		30,40	35

3.2　试验结果及分析

油气弹簧静态试验结果如图 5 所示。激励幅值为 30 mm、频率为 2 Hz 与激励幅值为 10 mm、频率为 4 Hz 时，不同节流阀和单向阀开度下，油气弹簧输出力随位移变化结果如图 6 ~ 7 所示。

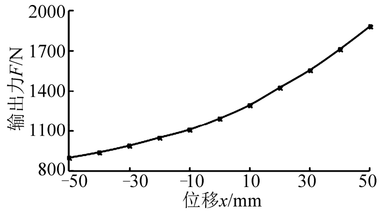

图 5　油气弹簧静态试验

Fig. 5　Hydro-pneumatic spring static test

油气弹簧静态力与位移关系如图 5 所示。由图可知，弹性力随位移变化呈非线性特性，表现出油气弹簧刚度随压缩量增大而增大。根据试验结果拟合出力与位移函数关系式，计算油气弹簧平衡位置静刚度 9.24 kN/m。由于动刚度略大于静刚度，设计符合要求。

力与位移关系随节流阀开度变化规律如图 6 所示。由图可知，输出力由弹性力和阻尼力两部分组成，激励位移较大时，弹性力变化较明显。复原行程输出力随节流孔开度增大而增大，压缩行程输出力随节流孔开度增大而减小。但节流孔开度对复原行程影响明显大于压缩行程，原因是复原行程油液只能从节流孔通过，而压缩行程油液可同时从节流孔和单向阀通过。

力与位移关系随单向阀开度变化规律如图 7 所示。由图可知，复原行程输出力几乎不随单向阀开度变化而变化，原因是只有在压缩行程时油液才从单向阀中通过，单向阀开度对复原行程无影响。压缩行程输出力随单向阀开度增大而减小。

从上述试验结果可以看出，所设计的油气弹簧具有很好的非线性刚度及较大范围的输出力调节能

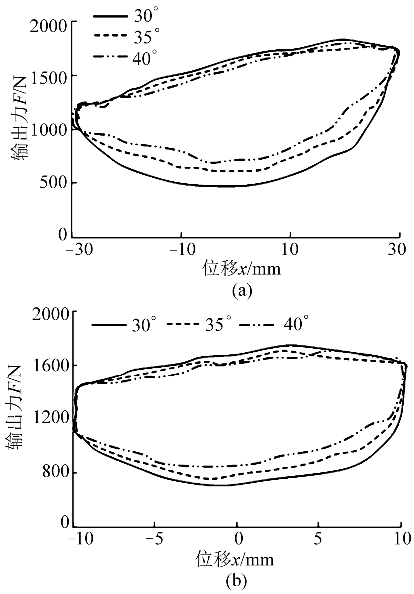

图 6　输出力随节流阀开度变化规律

Fig. 6　Influence of throttle valve on output force

(a) $A = 30$ mm, $f = 2$ Hz　(b) $A = 10$ mm, $f = 4$ Hz

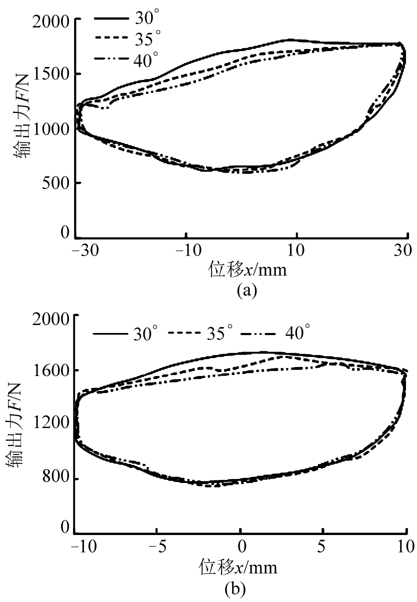

图 7　输出力随单向阀开度变化规律

Fig. 7　Influence of check valve on output force

(a) $A = 30$ mm, $f = 2$ Hz　(b) $A = 10$ mm, $f = 4$ Hz

力。通过调节节流孔及单向阀组的开度，能较好地满足拖拉机在不同路面及不同作业方式下行驶的减振要求。

4　结论

（1）以 CF700 型拖拉机为研究对象，建立了其振动模型，进行了驾驶室悬架参数匹配研究，并在此基础上研制了阻尼可调驾驶室悬架油气弹簧。

（2）对自行研制的阻尼可调油气弹簧进行了性能测试，研究了不同激励、节流阀开度及单向阀开度对输出力影响。结果表明，复原行程随节流阀开度

增大而增大,压缩行程输出力随节流阀开度增大而减小;压缩行程输出力随单向阀开度增大而减小,复原行程无影响。

(3)试验验证了油气弹簧设计的合理性,可为阻尼可调半主动驾驶室油气悬架研究提供依据。

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size, kinetic energy applied per unit drop volume and distances along the sprinkler, specific power and distances along the sprinkler were analyzed. Results indicated that there were proportional relation between the kinetic energy and drop size, and the maximum value of drop kinetic energy from test location increased with the distances along the sprinkler increased. At the same location from the sprinkler, the maximum and average values of drop kinetic energy decreased with the increase of working pressure. The kinetic energy per unit volume increased with the distances along the sprinkler increased, and the change between kinetic energy per unit volume and distances along the sprinkler met exponential function. At the same location from the sprinkler, the kinetic energy per unit volume decreased with the increase of working pressure. At the test location between 0 ~ 6 m along the sprinklers, the specific power values were less than 0.02 W/m^2 and the differences were small when operating pressure values were set at 100, 150, 200 kPa. At the test location from 6 m to the end of the sprinklers, the maximum values of the specific power were 0.117 2, 0.082 7 and 0.052 2 W/m^2, and the value of the specific power decreased with the increase of working pressure.

Key words: Sprinkler　Drop size　Droplet kinetic energy　Specific power　Operating pressure

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Design and Experiment of Hydro-pneumatic Spring of Tractor Cab Suspension

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Abstract: Based on the requirement of CF700 tractor cab suspension parameters, the elastic force and damping force model of hydro-pneumatic spring was established using fluid mechanics theory. The key parameters of hydro-pneumatic spring were calculated, and the damping adjustable hydro-pneumatic spring was designed. The effect of the excitation, throttle valve size and check valve size on the output force were studied by experiments. Test result is shown that the damping force can be in a large range of adjustment. The throttle valve size had influence on output force of both compression and rebound stroke, while the check valve size just had influence on the output force of compression stroke, and the throttle valve size had more obvious influence on the output force. The effectiveness of design ideas and methods were verified, and the research base of tractor cab hydro-pneumatic suspension was provided.

Key words: Tractor　Cab suspension　Hydro-pneumatic spring　Design　Experiment