Szent István University

THE OSCILLATION FEATURES OF TRACTORS WITH MECHANICAL FRONT WHEEL DRIVE, CHANGING OF TRACTION PARAMETERS

Thesis of the doctoral (PhD) dissertation

Zoltán Kovács

Gödöllő

2012

The doctor's school

designation: **Mechanical Engineering Ph.D School**

discipline: **Agricultural Technical Science**

Head of school: **Dr. István Farkas**
professor, DSc
Szent István University,
Faculty of Mechanical Engineering
Gödöllő

Supervisor: **Dr. Lajos Laib**
Professor, CSc
Szent István University,
Faculty of Mechanical Engineering
Institute of Process Engineering
Gödöllő

Co-supervisor: Dr. Kiss Péter PhD
university associate professor
Szent István University,
Faculty of Mechanical Engineering
Institute of Process Engineering

……………………………
Approved by head of school

………………………………
Approved by supervisors

CONTENTS

1. THE PREMISES OF THE WORK, GOALS

In the recent years the dominant agricultural tractor factories have done several technical changes on their work machines, the solution of the front wheel suspension falls into this category as well. According to the manufacturers by using the suspended front axle not only the comfort of the driving improves but as the front wheels stay on the ground continiously the tracking force of the power machine during field work increases, the slip reduces and the on roads the stability of the tractor also increases.

The rate of the universal power machines with mechanical front wheel drive has been continously increasing since the 1980s among the tractors used. Often these tractors have suspended front axle. Based on this the theme might be justified, since the knowing of the dynamical effects of the tractor is getting more important as the engine performance and the moving speed is increasing. The presently used high performance universal power machines with mechanical front wheel drive should perform great traction force under extreme conditions so they put a great demand on the tires.

The usage of the suspended front axle instead of rigid axle alters the shocking phenomena of the tractor. The earlier researches and examinations mainly focused on the improvement of the comfort level, the shocking acceleration on the cab and on the driver. Only little attention was paid to study and clarify the traction and energetic parameters.

Another problem of the tractors with mechanical front wheel drive is the power hop phenomena. This can be observed usually in case of great drawbar force and at given speed and worsens the traction features of the tractor and increases the dynamical usage of the some parts. The publications and other sources regarding this phenomena do not deal with in detail with the analyses and the study of it.

Based on this, I chose the study and examination of the traction features and shocking features of tractors with suspended front axle with mechanical front wheel drive. I defined the aims of the research as it follows:

1) Studying the main traction parameters (drawbar force, speed, traction performance, slip) of the tractors with mechanical front wheel drive and the connection between the slip and the drawbar force in case of active and inactive front axle suspension. To establish and compile an unique measuring system taking into consideration the dynamical effects.

2) Statistical analyses (deviation analyses, defining the dynamical factor range, etc.) of the main traction parameters and upright shocking. Studying the "energy absorving" effect of damping and shocking during the bouncing of the tractor.

3) Studying and analysing the power hop phenomena. Defining the main parameters of the effect of upright shock acceleration on the drawbar force and the traction performance in case of active and inactive suspended front wheel axle.

2. MATERIAL AND METHOD

I chose the methods of the experiments and examinations in accordance with the aims set. It means that in order to reach an aim I made individual experiment or by using an experiment I collected data for different aims.

In order to realise the research aims I did field tests three times at two locations. The base of the field tests was always traction experiment.

2.1. The main technical parametres of the tractor examined

The experiments were done by a JOHN DEERE 6920S type tractor. The tractor is front wheel steering with mechanical front wheel drive. The main technical parametres of the power machine can be found in Table 1.

Table 1.: Main technical parameters of the JOHN DEERE 6920S type power machine

Parameters	**Unit**	**Dimensions**
Lenght	mm	5815
Distance between axes	mm	2650
Hight of dragbar	mm	850
Performance	[kW/min^{-1}]	110/2100
Suspension mode	-	Front: rigid, suspended bridge Rear: rigid, unsuspended bridge
Tires	-	Front: TAURUS 14.9 R28 Rear: TAURUS 520/70 R38

Due to the aims set I needed a tractor that has front axle suspension but it can be turned off so the rigid suspension can also be examined. The tractors examined was equipped with a suspended front axle. The essence of the suspension system is that the rigid front axle is joined to the frame with two hydraulic work cylinders. A simplified line diagram can be seen on Figure 1.

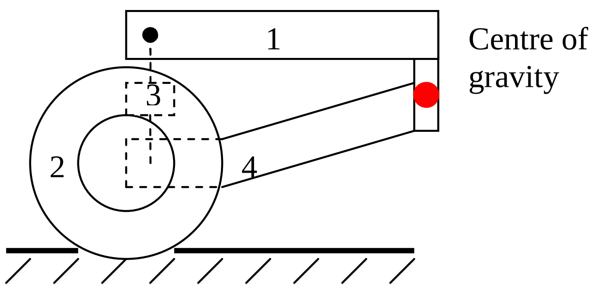

Figure 1. Scheme of suspended front axle
1 – frame; 2 – front wheel; 3 – hydraulic workcylinder; 4 – block of driving

2.2. The retarding tractor and the retarding cart

During the traction experiements at the field tests a retarding tractor and a retarding cart were used. The retarding tractor was a New Holland TM165 type power machine. The retarded and the retarding tractors were joined by a 20 m long cable rope and a dynamometer was built in the tractor examined.

This retarding vehicle was prepared from a MAZ 537 rocket and tank mover vehicle. The tractor examined and the retarding vehicle were joined by a 5 m long coupling rod and a dynamometer was built in the tractor examined.

2.3. Measuring equipment used for the experiments

During the measurements I used the properties of the Hungarian Institute of Agricultural Engineering. During the filed tests measurements for the given measuring tasks I used the following measuring tools:

Defining the soil compression: done by a EIJKELKAMP 06.15.01 type penetrometer.

Measuring the speed: the measurement of the speed was done by the own calibrated radar speed counter of the tractor.

Measuring the rotational number of the wheels: for counting the rotational number of the rear wheel hub a HEIDENHAIN ROD 430 type transmitter was used which has a 500 imp/rotation impulse number. The transmitter was driven by an adhesion disc that was connected to directly to the hub. The measurement of the rotation number of the front wheel hub was done with a similar transmitter and method.

Measuring the traction force: a force measuring transmitter, type HBM U2B, this was built in the coupling rod.

Measuring the upright shock acceleration: by a HBM B12/200 type acceleration sensor. The sensors were built in the right and left side of the front axle and above the axle on the frame.

The construction of the data collecting system: a SPIDER Mobil measuring and data collecting system with 16 channels received the data from the different transmitters.

2.4. The method of the investigations

2.4.1. Field traction tests

Duing the filed experiments the brutto weight of the tractor was set to 7.860 kg with auxiliary weights. The applied statical axle loads were the followings:

1) 33,08 % of the brutto weight (2.600 kg) on the front axle, 66,92 % (5.260 kg) on the rear axle;
2) 40,71 % of the brutto weight (3.200 kg) on the front, 59,29 % (4.660 kg) on the rear axle;
3) 48,35 % of the brutto weight (3.800) on the front, 51,65 % (4.060 kg) on the rear axle.

During the experiments I changed the gears as well (B1=5,5 km/h, B3=7,9 km/h, C2=10,5 km/h), and the position of the system ensuring the suspension of the front axle (turned on – active, turned off – inactive). The pressure of the tyres was set in accordance with the values given by the manufacturer. (Table 2.).

Table 2.: Loadability of the tyre TAURUS 14,9R28

Pressure [bar]	0,6	0,8	1,0	1,2	1,4	1,6
Loadability [kg]	1015	1200	1365	1520	1665	1800
Self-shock num. [s^{-1}]	8,68*	8,94	9,18*	9,47	9,63*	9,80
Shock time [s]	0,724*	0,702	0,684*	0,663	0,652*	0,641
Frequency [Hz]	1,38*	1,42	1,46*	1,51	1,53*	1,56

* - the data were defined by interpolarisation

2.4.2. Analyses the power hop phenomena

The experiment was done with stable brutto weight (7860 kg) with two different statical axle loads (the rate of the front axle load was 45,8 % and 38,7 %) in three different gears (A1=3,8 km/h, B1=5,5 km/h, B2=6,3 km/h). The traction load was generated by a retarding vehicle. The traction load was increased till the wheel hop came into being or the slip reached the 75-80 %. The sample taking frequency of the measurement was set to 100 Hz.

2.4.3.Defining the suspension parametres

On an experimental road I defined the spring characteristics of the tyre and whole trailing system in case of active and inactive suspension. The process of the measurement is shown on Figure 2.

Based on the Table 1. I set the statical load of the front axle and the appropriate tyre pressure. With the two front wheels I parked on the balances and with a help of a hydraulical lifiting machine I lifted the tractor at first at the front axle (1), and at the frame (2) till the wheels left the balances. After this in steady steps I put down the tractor and recorded the connected moving-weight data pairs.

Moreover, with a dropping experiment I recorded the shocking curves of active and inactive front axle suspension under three different axle loads (3600, 3320 és 3040 kg) and related tyre pressured (1,6; 1,4 és 1,2 bar). Based on the data I defined the logarithmical decrementum (δ), the Lehr dumping-number (D_{cs}), and the damping coefficiency (k).

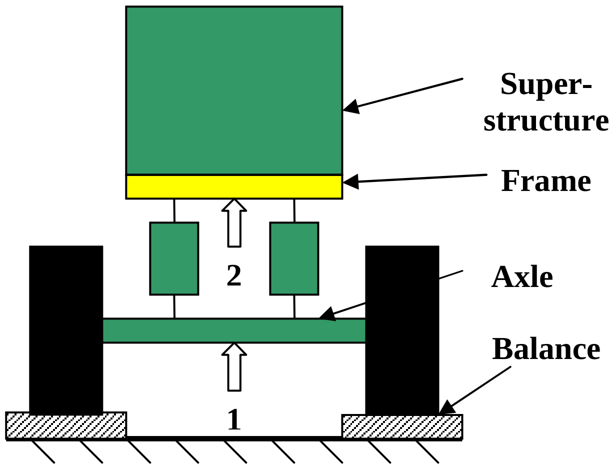

Figure 2. Line drawing of measuring the spring characteristics

3. RESULTS

The most frequent form of operating an agricultural implements is traction. To describe the tractor as a power machine the traction parameters are used. Thes are the followings: traction force, wheel slip, speed, traction performance, adhesive factors. In accordance with the aims I conclude the results of the experiments as it follows.

3.1. Examining the traction parameters of the tractor

3.1.1. Connection between the traction force and the shock acceleration of the frame

During the tractor in operation the load is not stable and constant but changes strongly in time, it is an instacioner load. On Fig. 3 beside 30 kN load the changing of the traction force can be seen in case of active and inactive front axle suspension.

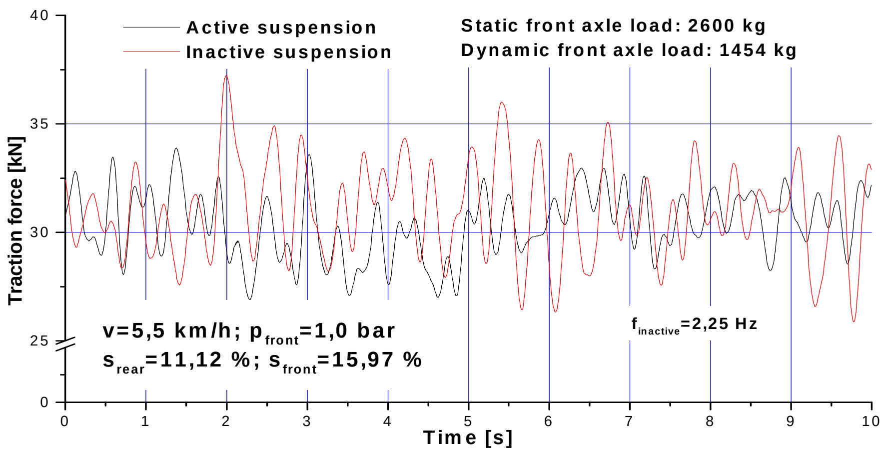

Figure 3. Fluctuation of the traction force beside constant load in case of active and inactive forn axle suspension

If meanwhile we examine the shock acceleration values of the frame even after a visual analyses the difference between active and inactive suspension is obvious (Fig. 4). In order to ensure the more objective and detailed analyses I defined the shock acceleration RMS (Root Mean Square) values of the frame and righ and left front wheel under the traction force examined. The RMS value informs us about the "energy content" of the shock. The higher the value the bigger is the "energy absorbing" ability of the shock. The counted values are summarized in the Table 3.

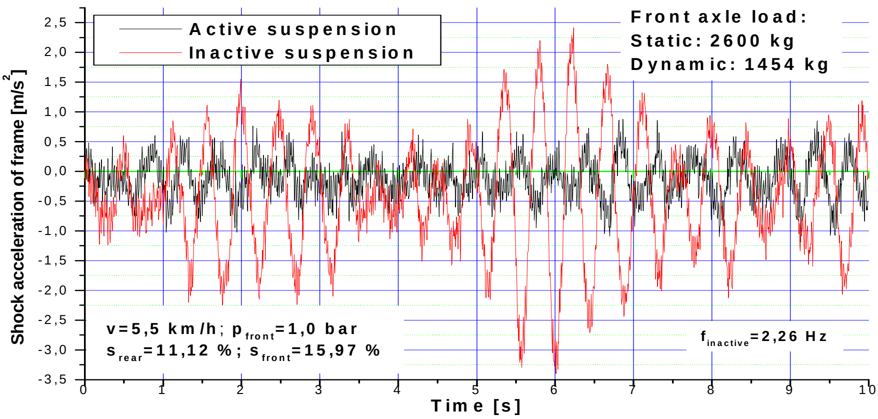

Figure 4. Shock accelerations of the frame under stable load in case of active and inactive front axle suspension

Table 3. The RMS values of the shock accelerations of the frame and the right and left front wheel under different experiment settings ($F_v \approx 30$ kN)

Gear	**SAD* rate [%]**	**RMS [m/s²]**					
		Left front wheel		**Frame**		**Right front wheel**	
		Active suspen.	**Inactive suspen.**	**Active suspen.**	**Inactive suspen.**	**Active suspen.**	**Inactive suspen.**
B1	33/67	0,841	1,298	**0,363**	1,065	0,839	1,354
	40/60	0,922	1,165	**0,566**	0,787	1,152	0,978
	48/52	0,908	1,193	**0,785**	1,070	0,972	1,415
B3	33/67	0,906	1,327	**0,550**	1,042	1,009	1,309
	40/60	1,073	0,788	**0,461**	1,109	1,352	1,089
	48/52	1,139	0,993	**0,667**	0,772	1,280	0,987
C2	33/67	1,263	1,347	**0,675**	0,963	1,076	1,198
	40/60	1,295	1,859	**0,771**	1,154	1,637	1,686
	48/52	1,531	1,767	**0,787**	1,308	1,485	1,662

*SAD= Statical Axle Load

The results obviously support the pre-consumpiton that the active suspension reduces the shock of the frame. This more moderate shock amplitude results that fluctuation of the traction force will have smaller amplitude.

3.1.2. Changes of the connection of the traction force and slip in case of active and inactive suspension

Knowing the connection between the traction force and slip is essential to judge the traction ability of the given tractor. During the traction experiments done under different conditions I collected the data and defined the connection between the traction force and the slip during the whole load cycle under active and inactive suspension. (Figure 5.)

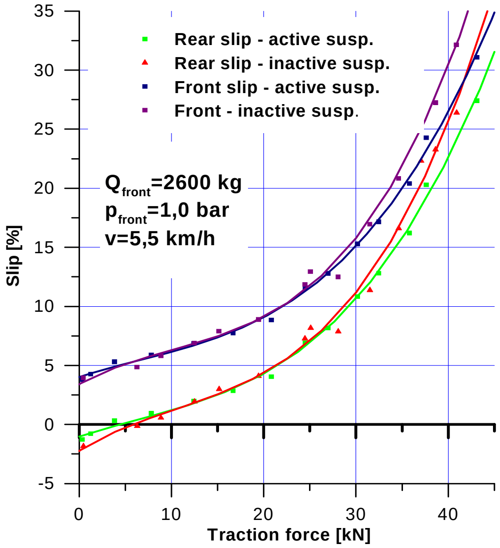

Figure 5. Changes of the wheel slips related to the traction force

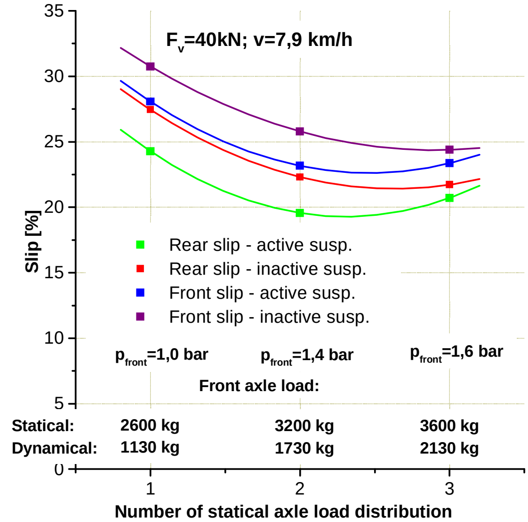

Figure 6. The effect of the changes of the statical axle load on the wheel slips under constant traction force

From the Fig. 5 it can be seen that the connection between the traction force and slip in case of the active and inactive suspension is similar during the whole load cycle. The slip-traction force curves running together to given traction force value. The separation points depends on the traction force, a higher traction force means a bigger pressure in the frontier tyre and the relevant major vertical wheel load. The positive effect of the active front axle suspension can be realized under higher traction force. Generally speaking this means that the use of the active front axle suspension reduces the size of the wheel slip in the upper 50 % of the traction performance ability.

The distribution of the brutto weight of the tractor between the front and rear axles is a very important setting parameter. As an example I refer to the Figure 6 where under constant traction force (F_V=40 kN) the front and rear wheel slips can be seen with active and inactive suspension at the same experimental settings. According to the results the dynamical axle load has an effect on the front and the rear wheel slip. The dynamical axle load is the question of the current traction force. By increasing the statical front axle load (1⇒2) the slip of the front and rear wheels reduces. If we keep on increasing the statical load of the front axle (2⇒3) in case of active suspension we can see that in both wheels a slight slip increase. But in case of inactive suspension a little reduction (in case of the back wheels 0,5 %, and in the front 1,4 %) can be seen.

3.2. Deviation analyses of the traction parameters, defining its dynamical factors

I did the statistical analyses of the important traction parameters, dynamical front axle load and the front upright shock acceleration of the front wheels and the frame. The 200 Hz sample taking frequency offered the chance to examine the deviation and the dynamical factor range of the main traction parameters, such as traction force, speed, traction performance and wheel slips.

3.2.1. Result of the deviation analyses of the traction parameters and the shock acceleration

I defined the deviation of the different parameters by using the following formula:

$$\sigma_{st} = \sqrt{\frac{\sum (x_i - \bar{x})^2}{n-1}}, \qquad (1)$$

where: x_i – the momentary value of the given parameter
$\bar{x}$ - average value of the given parameter;
n – number of data.

The deviation is important to know because in this way we can have a picture if the active front axle suspension has an effect on the changes of the parameter examined. If the dispersion is smaller in case of the active suspension with the same settings than in the case of inactive suspension it means that the tractor can perform smoother traction performance and traction force.

I defined the dispersion of the parameters examined in all the three gears with all the three statical axle load with 20, 30 és 40 kN average traction force. In the Tabel 4. as an example the dispersion values can be seen in B1 gear with average 40 kN traction force.

We can draw the following important conlusions based on the results of the dispersion analyses:

- the dispersion of the traction force values is smaller with active front axle suspension than in inactive suspension with all the three statical axle load.;
- in the dispersion values of the speed there is no significant fluctuation in case of active or inactive suspension;
- the dynamical front axle suspension shows smaller fluctuation in case of active suspension under all settings;
- the dispersion of the shock acceleration values of the frame is smaller in case of active suspension than in case of inactive suspension in all the three statical axle loads;

- if the dispersion values of the shock acceleration of the wheels are treated as base values it can be said that dispersion of the values of the shock acceleration of the frame reduces significantly (50-70 %);
- if we compare the shock acceleration values measured on the wheels with active and inactive suspension the frame's shock acceleration is lower with active suspension. This reduction in the case of smaller traction force (20 kN) is about 50 %, in case of higher traction forces it reduces to 25-30 %.

Table 4: The dispersion values of the traction parameters and the shock acceleration

Speed: v=5,5 km/h; F_v=40 kN s_{active}=26,62 %; $s_{inactive}$=30,14 %						
Traction parameters	**Active susp.**	**Inactive susp.**	**Active susp.**	**Inactive susp.**	**Active susp.**	**Inactive susp.**
	Front axle load [kg] Statical: 2600 Dynamical: 1130		**Front axle load [kg] Statical: 3200 Dynamical: 1730**		**Front axle load [kg] Statical: 3600 Dynamical: 2130**	
Traction force [kN]	2,349	2,396	2,928	3,539	2,632	2,944
Speed [m/s]	0,021	0,018	0,021	0,053	0,015	0,022
Traction performance [kW]	2,782	2,749	3,401	4,259	3,349	3,562
Front wheel slip [%]	1,707	1,451	1,554	3,472	1,319	1,699
Rear wheeé slip [%]	1,461	1,261	1,455	3,454	1,116	1,587
Dynamical front axle load [kg]	78,3	82,7	95,7	116,7	96,3	106,1
Left front wheel shock acceleration [m/s^2]	0,795	0,764	0,740	1,012	0,990	0,956
Frame shock acceleration [m/s^2]	**0,384**	0,498	**0,347**	0,668	**0,421**	0,645
Right front wheel shock acceleration [m/s^2]	0,937	0,751	0,710	0,951	0,767	0,863

3.2.2. Defining the dynamical factor and the CV of the traction parameters.

The explanation of the physical content of the dynamical factor range is the following: the values of the factor range show within what extreme the parameter examined change in comparison with the average value so what is rate of the smallest ($\phi_{din\ min}$) and the highest ($\phi_{din\ max}$) deviation among the average value. Based on this the smaller dynamical factor range means the smaller fluctuation of the values examined. I defined with the following formula the dynamical factor range of the important traction parameters, the dynamical front axle load and the upright shock accelerations of the front wheels and the frame in case of active and inactive suspension:

$$\phi_{din} = \left(\frac{x_{\min}}{\bar{x}} ; \frac{x_{\max}}{\bar{x}} \right), \qquad (2)$$

where: x_{min} and x_{max} – the smallest or highest value of the measured or counted parameter ;
$\bar{x}$ - the average value of the given parameter;

Regading the above mentioned parameters beside the dynamical factor ranges I give the value of the variable coefficient (CV) that was defined by the following formula:

$$\boldsymbol{CV = \frac{\sigma_{st}}{\bar{x}}}. \qquad (3)$$

The CV connects the dispersion and the average values of the values examined. Regarding its physical content it shows the rate of the dispersion of a given parameter compared to the average value.

We can draw the following conlusions based on results of the traction parameters, dynamical factor range of the dynamical front axle load and the CV:

- during operating active front axle suspension the dynamical factor range of the parameters examined under the same settings is usually lower than in inactive suspension. This means that the front axle suspension reduces the dynamics of the paremeters and the tractor will be able to perform smoother performance.;
- with active suspension the CV of the wheel slips in case of the front wheel are between 0,053 – 0,208 (5,3 % - 20,8 %), while in the case of the rear wheels they are between 0,051 – 0,371 (5,1 % - 37,1 %);
- in case of inactive suspension these data were the following: front wheel slip 0,059 – 0,263 (5,9 % - 26,3 %); rear wheel slip 0,060 – 0,495 (6,0 % - 49,5 %);
- to sum up it can be said that in case of all the three static axle load the CV value of the front and rear wheels slips is smaller with active front axle suspension than with inactive suspension.

3.3. The effects of the upright shockings on the traction parameters of the tractor

In accordance with the aims I examined how the upright shockings of the wheel effect the traction force performance. For this we need to be familiar with the springing system of the tractor and the spring characteristics in case of both active and inactive suspension.

3.3.1. Defining the spring characteristics

I defined the front tyre of the tractor and the tyre and the suspension collective, resultant spring characteristics with active and inactive suspension. I did the experiments with three different axle load (3600 kg, 3040 kg and 2400 kg) and related tyre pressures (1,6; 1,2 and 0,8 bar). As an example on Figure 7. a spring characteristics with 3600 kg static front axle suspension can be seen.

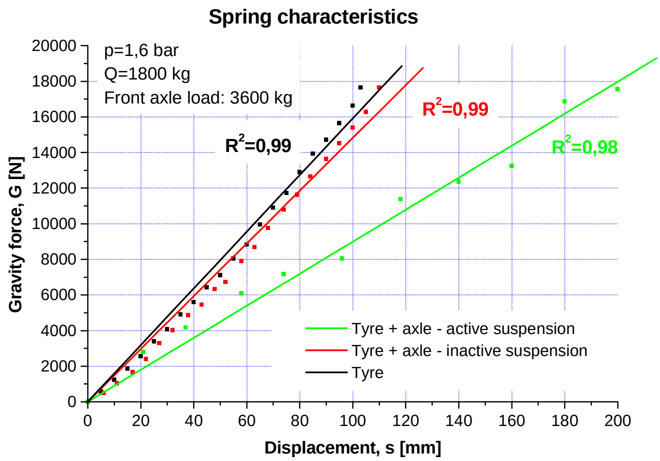

Figure 7. Spring characteristics for 1,6 bar tyre pressure

It can be stated that the spring characteristic of both the tyre and the whole system are basically linear. The counted spring stiffness and spring constant values can be seen in Table 5.

Table 5. Values of spring constans and spring stiffness

Tyre pressure [bar]	Suspension	Spring constant, k_r [N/cm]			Spring stiffness, c_r [mm/kN]		
		tyre	axle	resultant	tyre	axle	resultant
1,6	Inactive	1730	21292	1600	5,78	0,47	6,25
	Active	1730	1876	900	5,78	5,33	11,11
1,2	Inactive	1364	18429	1270	7,331	0,543	7,874
	Active	1364	1529	721	7,331	6,540	13,869
0,8	Inactive	960	17111	909	10,42	0,584	11,00
	Active	960	1403	570	10,42	7,128	17,54

From the data of Table 5 it can be seen that with inactive suspension the spring constant between the axle and the frame is much higher than the tyre's. But with active suspension the spring constant between the axle and frame is about the same as spring constant of the tyre, a bit higher.

3.3.2. Defining the main parameters of the suspension system

By experimentally I recorded the damping curves of the suspension system (Figure 8.). With the help of the recorded damping curves I defined the typical logarithmical decrementum of the given setting and with this several important parameters of the suspension system (e.g. the Lehr damping number, damping coefficient, performance of oscillation, damping loss etc.) can be counted. Further parameters of a given suspension system are the self-shock number (α), the shock time (T) and the frequency (f) (Table 6). The data of the Table 6 gives the dynamic self-shock numbers and the frequencies. If we compare these values with the statical values (Table 2), it can be said that the dynamical values are always higher with every setting.

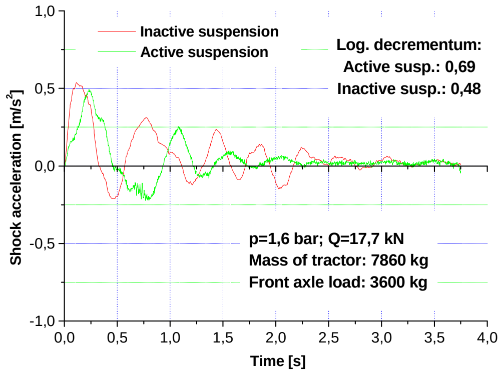

Figure 8.
Damping curve of 1,6 bar front tyre pressure

Table 6. Main counted parameters of the suspension system

Tyre pressure [bar]	**Suspension**	**Self-shock number α [s^{-1}]**	**Shock time T [s]**	**Frequency f [Hz]**	**Damping number D_{cs}**	**Damping coefficient k [Ns/m]**
1,6	Inactive	11,44	0,549	1,82	0,076	3147
	Active	11,17	0,562	1,78	0,109	4418
1,2	Inactive	14,57	0,431	2,32	0,073	3244
	Active	13,51	0,465	2,15	0,112	4643
0,8	Inactive	13,39	0,469	2,13	0,065	2098
	Active	12,56	0,562	2,00	0,115	3116

The Table 7 shows the frequencies of the frame shock acceleration with the three axle load and different traction force performance with inactive suspension. According to the data by increasing the traction force the frequencies of the shock increase as well.

Table 7. Frequencies of the frame shocks

Front axle load [kg]		**2600**			**3200**			**3600**		
Traction force [kN]		**20**	**30**	**40**	**20**	**30**	**40**	**20**	**30**	**40**
Gear	**B1**	2,00	2,25	2,38	2,10	2,25	2,32	2,00	2,18	2,30
	B3	2,00	2,20	2,28	2,00	2,20	2,30	2,00	2,20	2,23
	C2	2,10	2,20	2,25	2,00	2,20	2,25	2,00	2,18	2,20

3.3.3. The transmissibility of suspension system

In case of the inactive suspension the system can be examined as one degree of freedom linear suspension one. The Figure 9 shows the idealized model of the suspension system in case of inactive suspension. For the rate of circle frequency of the excitation ω and the self circle frequency without damping α the following notation can be introduced:

$$\xi = \frac{\omega}{\alpha} = \frac{2\pi f}{\alpha} \qquad (4)$$

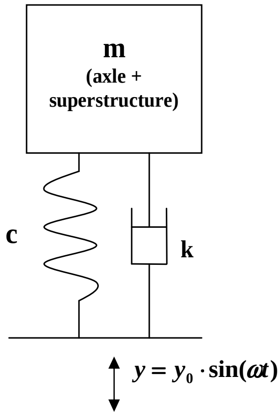

Figure 9. Simplified model of the inactive suspension

The rate of the steady shocking and the amplitude of the excitation is the so-called transmissibility (N). The transmissibility depends on the current excitation frequency, its definition was done by the formula (5):

$$N(\xi) = \frac{K_{áll}}{K_{st}} = \frac{\sqrt{1 + 4D_{cs}{}^{2}\xi^{2}}}{\sqrt{(1-\xi^{2}) + 4D_{cs}{}^{2}\xi^{2}}}, \ (5)$$

where: $K_{áll}$ – shock amplitude during the steady moving of the weight,
K_{st} – statical amplitude (excitationed amplitude),
D_{cs} – Lehr damping number,
ξ - the rate of the excitationel and the self circle frequency.

During the counting I gradually increased the values of the excitationed circle frequency ω between 0-20 rad/s. The result can be seen on Figure 10. The analyses of the curves show that the resonance frequency in case of inactive front axle suspension appears in the range of 1,5 Hz and slightly depends on the upright load of the wheels. The peak of the curves appear where with given setting the self-shock number and the excitationed frequency (ω) are equal. This resonance causes the “power hop” phenomena.

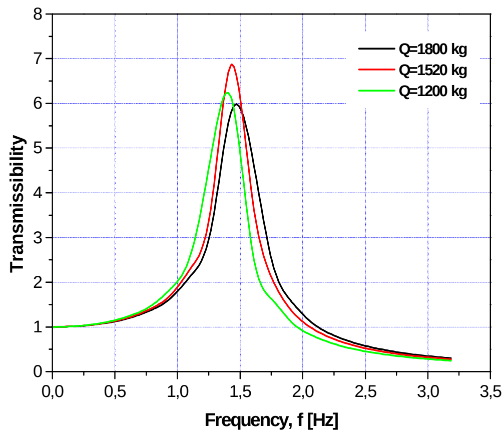

Figure 10. Transmissibility function of the inactive suspension

In case of active suspension there

is a suspending-damping unit between the axle and the frame. During the counting we may model it with a spring and a damper. The simplified model can be seen on Figure 11. The weight of the front axle is stable (m_1=400 kg/wheel), the m_2 can be defined depending on the current axle load. I did the countings in the f=0-10 Hz (ω=0-62,8 rad/s) excited frequency range.

The motion formulas can be written by using the Newton's Second Law as it follows:

$$m_2\ddot{y}_2 + k_2\dot{y}_2 + c_2 y_2 = c_2 y_1 + k_2\dot{y}_1 + f \qquad (6)$$

$$m_1\ddot{y}_1 + k_2\dot{y}_1 + (c_2 + c_1)y_1 = c_2 y_2 + k_2\dot{y}_2 + c_1 y \qquad (7)$$

where: m_2 – the weight of the construction,
m_1 – the weight of the front axle,
k_1 – damping factor of the tyre,
k_2 – damping factor,
c_2 – spring constant,
c_1 – spring constant of the tyre,
y_2 – displacement of the construction,
y_1 – displacement of the axle,
y – displacement of the road exciting,
f – inner exciting forces of the vehicle.

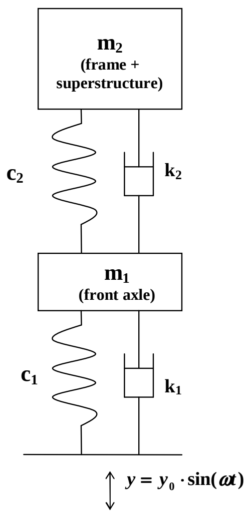

Figure 11. Simplified model of the active front axle suspension

Based on this transmissibility between the road exciting and frame it can be defined by formula (8):

$$\frac{y_2}{y} = \frac{c_1(c_2 + \delta k_2\omega_g)}{(c_2 + c_1 - m_1\omega_g{}^2 + \delta k_2\omega_g)(c_2 - m_2\omega_g{}^2 + \delta k_2\omega_g) - (c_2 + \delta k_2\omega_g)^2}, \qquad (8)$$

where: δ - the logarithmical decrement;
ω_g – excited frequency.

The results of the countings can be seen on Figure 12. The transmissibility functions have two peaks under all the settings. This two-peak transmissibility function is the feature of the two-weighted shock system. The first peak in all the three cases appears at 1,25 Hz-excited frequency.

This is because of the stable m_1 weight. The second peak though appears at a higher – between 4-6 Hz – frequency. This change can be explained by the reduction of the axle load (m_2 variable). The smaller is the value of the m_2 the smaller is the value of the excited frequency where the second peak appear.

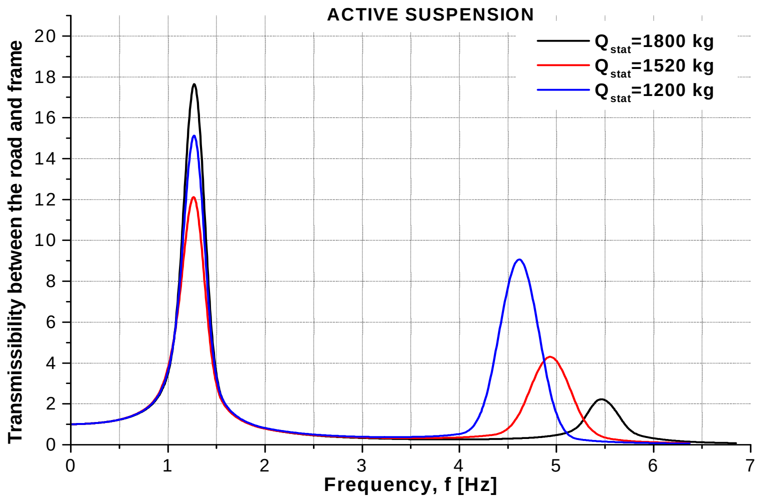

Figure 12. The transmissibility function of the active front axle

3.3.4. Examining the shock acceleration of the frame

I defined the RMS values of the shocks of the frame under all the experiemental settings regarding the whole load cycle. The result can be seen on Figure 13. From the figure it can be seen that in case of active and inactive front axle load the shock acceleration RMS values of the frame changes different in function of traction force.

In case of active suspension the values fluctuate around a stable (≈0,4) value. This means that the damping of the frame virtually remains the same during the whole load cyle. But in the case of inactive suspension the values – as the traction performance increases –

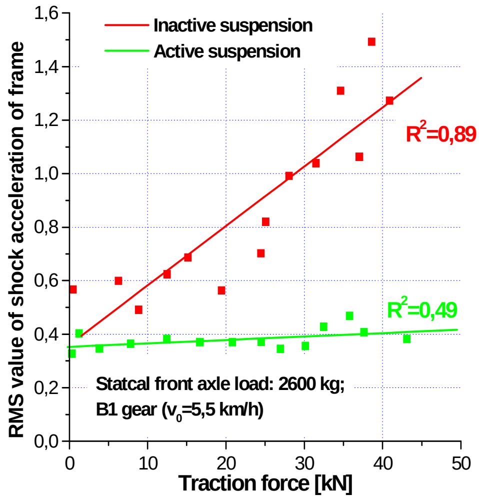

Figure 13. Changing of the shock acceleration RMS values of the frame regarding the traction force performed

increase along a line. These results also justify that without suspension more significant shocks appear on the frame.

3.4. Examining the formation of wheel bouncing

3.4.1. Formation of power hop

The examination of the formation of power hop phenomena of was done by traction examination. I processed the data of the examinations in the following way:

- I "put on" each other the experimental data of the active and inactive suspension with the same settings (Figure 14.);
- based on the same average traction force from the whole data stream I chose four – 20 seconds long each – measurement section for further analyses;
- I defined the values of the average traction force and traction performance of each measurement section (Figure 15.);

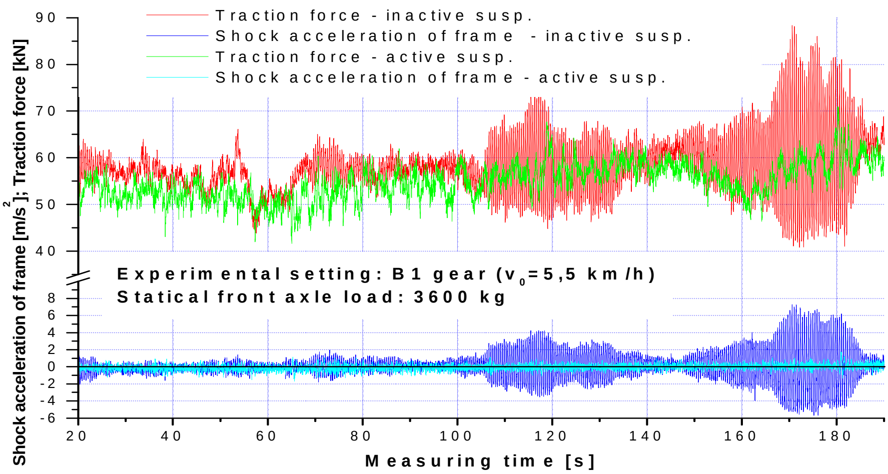

Figure 14. Changing of the traction force and shock acceleration of frame during the measurement period

On the Fig. 14 it can be seen clearly that though the average traction force seems to be the same with active and inactive suspension, the rate of the fluctuation is more significant in case of inactive suspension. If we examine the changes of the shock acceleration of the frame it can be seen that during active suspension same shocks appear in the whole measurement period. But in case of inactive suspension significant shock amplitudes come up under the examined settings between the 100-120 and the 150-170 seconds intervals. In these intervals the front wheels and the tractor is "bouncing" so the phenomena called power hop appears.

On Fig. 15 we can see four – each is a 20 seconds long – measurement section and their average values of traction force and traction performance. The four marked sections are: 1) between 40-60 s; 2) between 70-90 s; 3) between 100-120 s; 4) between 150-170 s. Based on the average traction force performance we can say that there was no signficiant difference between the active and inactive suspension at eaxh measurement section. But this can be misleading since the average values themselves do not show the dynamics of the traction force or the traction performance.

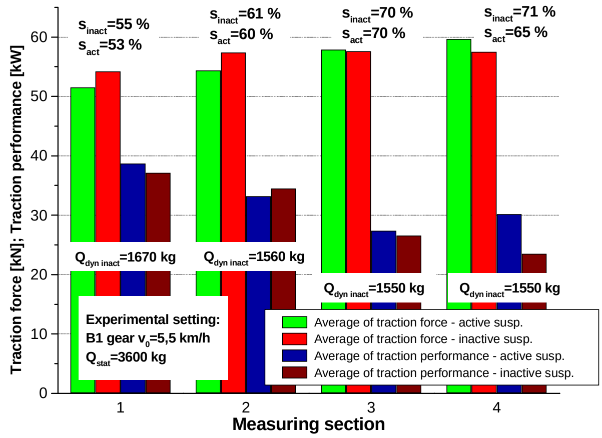

Figure 15. Average values of the traction force and traction performance with active and inactive suspension

The deviation of the values of the shock acceleration of the frame with active suspension is about 38,5 – 50,3 % of the deviation of the values of the left wheel shock acceleration. This rate is between deviation of the values of the shock acceleration of the right wheel and frame is 37,5 – 54,3 % . This means that the suspension reduces to about the half or even lower the deviation of the values of the shock acceleration on the wheels. In case of inactive suspension thos reduction is smaller: in relation with the front left wheel the deviation of the values of the shock acceleration of the frame is 64,1 – 97,6 till in relation with the front right wheel it is 57,2 – 99,3 %.

3.4.2. Defining the main parameters of the shocks paremeters during the bouncing

When the power hop appears a significant fluctuation of the frame and the traction force can be seen. The fluctuation has wave-like characteristics so a periodical damp movement appears. Under the different examined settings I defined the main parameters of the shock accelerations of the bouncings (Table 8).

From the data it can be seen clearly that there is not significant fluctuation in the damping time of the bouncings. Accordingly the frequency of the dampings and its circle frequency stay within a relative narrow range (ν=2,13…2,51 Hz; ω=13,62…15,78 rad/s). More important differences can be seen regarding the shock acceleration and the damping amplitudes.

Table 8. Main counted parameters of the periodical shock accelerations during bouncing

Gear	Statical axle load rate front/rear [%]	Section	Shock time T [s]	Frequency f [Hz]	Circle frequency ω [Hz]	Amplitude of shock acceler. K [m/s^2]	Amplitude of oscillation r [mm]
A3	46/54	3	0,415	2,41	15,13	1,91	8,4
	39/61	4	0,405	2,47	15,51	3,72	15,5
B1	46/54	3	0,400	2,50	15,7	2,34	9,5
		4	0,440	2,23	14,27	3,87	19,0
	39/61	3	0,461	2,13	13,62	7,29	39,3
B2	46/54	3	0,425	2,35	14,78	5,19	23,8
		4	0,443	2,26	14,18	5,21	25,9
	39/61	4	0,399	2,51	15,74	4,18	16,9

We find a drifference between the statical frequencies (Table 2) and the dynamical frequencies defined with dropping examinations (Table 6). The traction force increases the dynamical frequencies (Table 7). This is not only means that there is a weight realignment between the front and rear axle, but beside the road excite there is a force excite as well that cause traction force fluctuation. This force excite makes its effect via the trailer hitch through a "virtual" spring with great spring constant. This case can be seen on the damping system on Figure 16. The results of the countings can be found in Table 9. After analysing the data it can be said that the value of the spring constant is between 3,45-4,09 MN/m during bouncing if the statical load of the front axle is 3600 kg. If we reduce the statical load to 3000 kg the damping frequency of the bouncing appears only at smaller spring constant (2,31-3,58 MN/m).

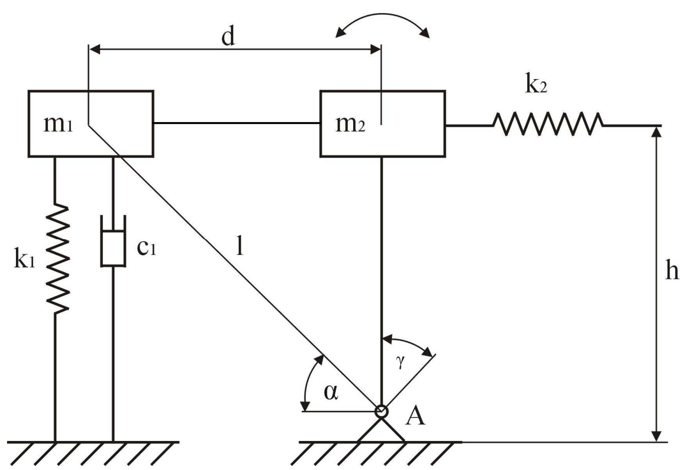

Figure 16. Modeling the excite of road and force
(based on Sitkei)

Table 9. The counted virtual spring constants of the damping system

Gear	Statical axle load rate front/rear [%]	Section	Measured frequency f [Hz]	Measured circle frequency ω [Hz]	Counted virtual spring constant k_2 [N/m]
A3	46/54	3	2,41	15,13	3.681.365
	39/61	4	2,47	15,51	3.580.686
B1	46/54	3	2,50	15,7	4.094.023
		4	2,23	14,27	3.451.367
	39/61	3	2,13	13,62	2.309.394
B2	46/54	3	2,35	14,78	3.964.804
		4	2,26	14,18	3.577.493
	39/61	4	2,51	15,74	3.742.308

If we drawing the amplitude of shock acceleration of the frame in function of amplitude of traction force, we can say that the connection between two amplitudes is linear (Figure 17 and 18). The frequency is constant when tractor is bouncing, so the amplitude of shock acceleration are equal to displacement. Due to it, the straight line gives the virtual spring constant. The results of the countings can be found in Table 10.

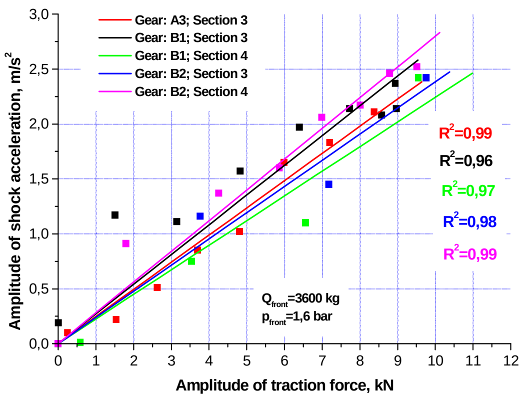

Figure 17. The amplitude of shock acceleration in function of amplitude of traction force (rate of front/rear axle load 46/54 %)

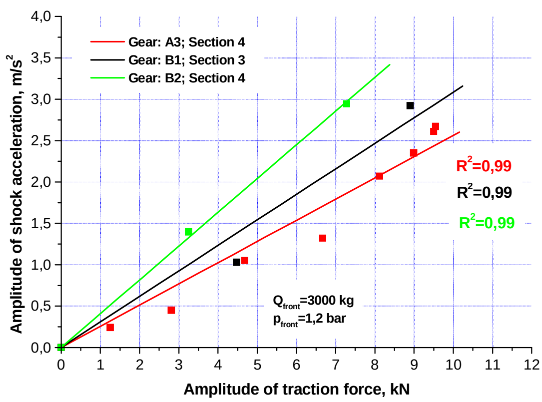

Figure 18. The amplitude of shock acceleration in function of amplitude of traction force (rate of front/rear axle load 39/61 %)

To compare the data of Table 9 and 10 we can say, that the frequencies counted by virtual spring constant from amplitudes of shock acceleration and traction force equal to frequencies measured in case of power hop.

Under plough-land circumstance at the tractor examined power hop appeared only at high traction force (45-60 kN) and wheel slips. Due to the great traction force performance the load of the front axle reduced significantly.

Table 10. The counted frquencies from virtual spring constant

Gear	Rate of statical axle load front/rear [%]	Section	Virtual spring constant k_{virt} [N/m]	Counted circle frequency ω [Hz]	Counted frequency f [Hz]
A3	46/54	3	3.600.000	15,04	2,39
	39/61	4	3.650.000	15,64	2,49
B1	46/54	3	3.830.000	15,36	2,46
		4	3.730.000	15,22	2,42
	39/61	3	2.150.000	13,11	2,09
B2	46/54	3	3.820.000	15,34	2,44
		4	3.580.000	15,01	2,39
	39/61	4	3.560.000	15,50	2,47

The rate of the reduction can even reach the 60 % of the statical value. The excitation of the road and the traction force is adding to each other. It is eventuate resonance, namely power hop. Therefore the amplitude of shock acceleration and traction force will be significant on the front axle and on dragbar separately. Analysis of the data which recorded in case of power hop, we can say, there is strongly linear connection between the amplitudes of shock acceleration and traction force (Figure 17 and 18). This connection we can describe by the next formula:

$$\Delta \boldsymbol{F_v} = \boldsymbol{k_{emp}} \cdot \Delta \boldsymbol{a} \quad]\boldsymbol{kN}], \qquad (9)$$

where: ΔF_v – the rate of fluctuation of traction force [kN];
k_{emp} – emphirical constans, valid in case of bouncing, depending on the type of vehicle, its value is 3;
Δa – shock acceleration measured on the frame [m/s^2].

In the formula (9) the value of "k_{emp}" means that the amplitude of the traction force changings how many times higher than the amplitude of the shock acceleration of the frame.

3.4.3. Defining the "delay time"

If we examine the temporary shock acceleration and traction force related to the measurement time in a common diagram we see that changings of the traction force follow well the changing of the values of the shock acceleration of the frame. At the same time the wave curve describing the changing of the values of the traction force "lags behind" the curve describing the changing of the damoings of the frame. Thus, by “delay time” I mean the time when the extreme values appear than the maximum and minimum of the frame damping. The “delay time” of the dampings of the traction force with different experimental settings can be seen on Table 11.

Table 11. Values of the delay times during bouncing under different experimental settings

Gear	**Rate of statical axle load front/rear** [%]	**Section**	**Delay time T_k** [s]	[^{0}]	[rad]	**Ratio to shock time** [%]
A3	46/54	3	0,032	27,8	0,48	7,72
	39/61	4	0,036	32,0	0,56	8,89
B1	46/54	3	0,025	22,5	0,39	6,25
		4	0,041	33,5	0,58	9,32
	39/61	3	0,039	30,5	0,53	8,46
B2	46/54	3	0,025	21,2	0,37	5,88
		4	0,023	18,7	0,33	5,19
	39/61	4	0,027	24,4	0,42	6,77

Based on the results the conlusion can be drawn that the delay time is much less in comparison with the damping time. The rate of the delay time within the dumping time under the experimental settings was between 5-10 %.

3.5. New scientific results

I conclude the new scientific results of the themes examined in the sequence of notification of the results. The themes can be treated individually but they organically belong to each other.

1. **During the field traction experiments I found that the slip-traction force curves running together to given traction force value. The separation points depends on the traction force, a higher traction force means a bigger pressure in the frontier tyre and the relevant major vertical wheel load. Due to literary data I determined that the optimal slip on investigated loamy soil not bigger than 15 % which connect 30 kN traction force at investigated tractor.**

2. **I found that with active suspension the 40/60 % front/rear axle load is the best setting for the connection between the slip and the traction force. This rate of statical axle load can take advantages at upper 50 % of the traction performance ability, where increasing of traction force can reduce the front axle load. In this traction force interval (30-40 kN) the dynamical**

load of front axle reducing to 64-54 % of statical value, while the dynamical load of rear axle increasing to 25-32 % of statical value.

3. I defined the dispersion values of the more important traction parameters (traction force, traction performance) and the dynamical front axle load. **I found that with a given experiment setting from the point of the examined parameters more moderate dispersion values can be seen in the case of active front axle suspension. The more moderate dispersion values mean more moderate tractor operation.**

I also found that with active front axle suspension the dispersion of the shock acceleration of the frame with active front axle suspension is always smaller than the inactive suspension. This state is true even if the dispersion of the shock acceleration of the front wheels with active front axle suspension is higher than with the inactive one. This means that the active suspension significantly reduces the shocks transferred from the wheels to the frame.

4. I defined dynamical factor range of the important traction-energetic parameters (traction force, speed, traction performance, wheel slips), and the dynamical front axle load with the following formula:

$$\phi_{din} = \left(\frac{x_{\min}}{\overline{x}} ; \frac{x_{\max}}{\overline{x}} \right),$$

where: x_{min} and x_{max} – the minimum or the maximum value of the measured or counted parameter;
$\overline{x}$ - average value of the given parameter;

I found that the values of the dynamical factors of the parameters examined during the plough-land operation dynamically fluctuate between a minimum and a maximum value. I showed that the active front axle suspension reduces the extent of the fluctuation so the value range of the dynamical factor of a given parameter is always smaller with all settings than the inactive one.

5. I defined the spring characteristics of the front axle suspension as a trailing system, its logarithmical decrement and resonance frequencies. **Additional I defined in experimental road the damping factor of tyre and suspension, as well as dynamical own frequencies which belong to nominal load of tyre. I detected, that this dynamical frequency – which is about 2 Hz – bigger than the statical frequency of tyre.**

Additional I detected that resonance frequency in case of inactive suspension is about 1,5 Hz, while in case of active suspension this value reducing to 1,25 Hz.

6. **I found that between the RMS value of shock acceleration of the frame and the traction force developed there is a linear function connection.** In case of active suspension if we increase the traction force the RMS values of the frame deviate around a horizontal line while in case of inactive suspension they deviate around a strictly monotonously increasing line.

This is true for all the cases examined and in generally speaking it means that RMS values of shock acceleration of the frame are constant in function of increasing traction force in case of active suspension. But in case of inactive suspension the RMS values increasing proportionately in function of traction force.

7. I defined the own static and dynamic frequencies of the front axle suspension with active and inactive suspension. **I defined that the difference between the dynamic frequency recorded during the experiment and the frequencies developing during bouncing was caused by the force excitation developed by the traction force.**

I also found that in case of inactive suspension the increasing traction force increases the frequency of the shocks of the frame and the frequency of the shocks of the traction force. This phenomena increases the statical 1,5 Hz to 2,4-2,5 Hz during bouncing.

8. **I found and showed that the power hop phenomena can occur in case of extreme big traction force and wheel slip. The big traction force significantly reduces the static front axle load, so the road excertation and fluctuation of traction force like force excertation can add to each other, which cause resonance.**

9. **Due to data recorded in case of bouncing I showed that acceleration with big amplitude appers on front axle, while side of traction force can occur force fluctuation with big amplitude too. I determined on the score of the process of data, when the power hop phenomena to come into being, between the amplitudes of acceleration and traction force has a linear connection. This connection describes the following formula:**

$$\Delta F_v = k_{emp} \cdot \Delta a \quad]kN],$$

where: ΔF_v – the rate of fluctuation of traction force [kN];

k_{emp} – emphirical constans, valid in case of bouncing, depending on the type of vehicle, its value is 3;
Δa – shock acceleration measured on the frame [m/s^2].

10. I introduced and defined the notion of “delay time” and defined its values.

Delay time: the time after which the amplitude of the traction force shock follows the amplitude of the shock acceleration of the frame.

I found that the 5-10% of the current shock time is the delay time. I defined the phase shift of the delay time which values were between 21-33^0.

4. SUMMARY

4.1. Summary of the research activity

The appearance of the universal tractors with mechanical front wheel drive in the agriculture generated several other technical improvements. Among these the solution for the front axle suspension can also be found. The practical usage of this raises some questions that can be answered only after appropriate examinations and analyses. In accordance with these my research aims focused on the followings:

- how the main traction features (traction force, speed, traction performance, slip) of the tractors and the connection between the slip and the traction force change?

- what consequencies can be drawn from the statistical analyses of the upright shock acceleration and of the main traction parameters and from the examination of the "energy absorbing" in case of active and inactive front axle suspension?

- studying and analysing the power hop phenomena. Defining the main parameters of the effect of upright shock acceleration on the traction performance and the traction force in case of active and inactive suspended front wheel axle.

For executing the aims set I made plough-land experiments at two locations for three occassions. In order to reveal the connection between the slip and the traction force the location of the experiments was a completely flat wasteland without any slopes after harvesting. For studying the wheel bouncing or as it is also refered to power hop an untouched, fallow land was chosen with slight slopes (3-5 %). The base of the tractor examinations on the lands was always a traction experiment.

4.2. The practical usage of the scientific results, conclusions, proposals

Based on the results of the research in accordance with the scientific aims described my conclusions and proposals are the following:

- The established measuring system can be used for comparing traction parameters during operation of the tractors with mechanical front wheel drive with active and inactive front axle suspension. The measuring method and the tools used may help the outworking and execution of similar researches.

- The examination of the important traction parameters show that active front axle suspension has a positive effect on the traction force mostly at the range of the upper 50% of the traction performance ability.

- The statical axle load has a discoverable effect on the slip-traction force connections in case of active and inactive suspension. From the results of the different experimental settings I drew the conclusion that in case of active front axle suspension with all wheel drive mode – mostly in case of the need of great traction force – the optimal share of the axle load is 40/60 % front/rear.

- After the statistical analyses of the traction parameters and the upright shock acceleration values the conclusion can be drawn that the active front axle suspension reduces the dynamics of both the traction parameters and the shock accelerations. Due to this the tractor can perform smoother traction force and traction performance.

- Under great stress the tractors with mechanical front wheel drive can produce a common problem, known as wheel bouncing. After executing tests and analysing the results I came to the conclusion that active front axle suspension withhold the above mentioned phenomena.

- My plough-land experiements justified that with inactive suspension a load can always be established where bouncing or performance hop appears. If the load does not change the dampings become periodical with almost the same frequency (2,13 – 2,50 Hz). The fluctuation of the traction force also becomes periodical with the same frequency and with a 5-10 % delay time in comparison with the shock time.

After analysing the results and the practical experiences of the test done by proposals are the following:

- When purchasing a new power machine it is advisable to choose a type which has a built-in suspended front axle or can be ordered as option. Since this would increase the price of the tractor with 5-8 % it would be advisable to make the decision makers and operators become familiar with its advantage.

- In the case of those tractors with mechanical front wheel drive that do not have suspended front axle I suggest to complete the subunit of the hydraulic suspension equipment with an upright shock acceleration sensor. If the upright shock acceleration sensor detects a shock acceleration value that is higher than the pre-defined so the tractor starts to bounce the power control can reduce the temporary traction force so the high ranged performance hop might be avoided.

- In order to establish the technical improvement described in the previous section further plough-land examinations are advised to clear up the necessary parameters details (e.g.: different soils and water content, different types of tyres and inner pressures, different soil covers, etc.)

PUBLICATIONS IN CONNECTION WITH THE THEME OF DOCTORAL DISSERTATION

Scientific articles in foreign language

1. **КОВАЧ, З.** – СЕВЛЫШИ, И. – КИШ, Ж . (2001): Справнение традициональной обработки почвы с минимальной обработкой с точки зрения уплотненности почвы. *Проблеми економичного и социального розвитку региону и практика наукового експерименту*. Науковый-техничный сборник, Выпуск 17, Киев-Ужгород, 2001.
2. SZŐLLŐSI, I. – KISS, ZS. P. – **KOVÁCS, Z.** – CZIRJÁK, T. (2001): Soil resistance and moisture content in sandy soil influenced by tillage and manuring. *Bulletin of Szent István University*, Gödöllő 2001-2002, pp. 109-118., ISSN 1586-4502
3. **KOVÁCS, Z.** – LAIB, L. – SZENTE, M. – KASSAI, ZS. (2003): Traction testing of agricultural power machines. *Annals of the University of Petrosani*, vol. (XXXII), Petrosani – ROMANIA, 2003. (Ed. ILIAS, N.), pp. 78 – 82. ISSN 1454-9166

Scientific articles in hungarian language

4. CZIRJÁK, T. - KISS, ZS. P. - **KOVÁCS, Z.** - SZŐLLŐSI, I. (2001): A penetrációs ellenállás változása különböző talajokon a tenyészidőszak alatt. *Agrokémia és Talajtan*, No. 3-4., 185-207. o., ISSN 0002-1873
5. SZŐLLŐSI, I. – KISS, ZS. P. – **KOVÁCS, Z.** (2002): Westsik-féle talajjavító vetésforgók hatása a talajtömörödöttségre. *Mezőgazdasági Technika*, XLIII. Évf. 2002. május, 38-39. o., ISSN 0026-1890
6. **KOVÁCS, Z.** (2007): A rugózott első híd hatása a traktor vonóerőkifejtésére. *GÉP*, LVIII. évf. (4), 35-39. o., ISSN 0016-8572
7. **KOVÁCS, Z.** (2007): A rugózott első hidas traktorok vontatási teljesítményének alakulása. *Mezőgazdasági Technika*, XLVIII. évf. (10), 2-4. o., HU ISSN 0026 1890
8. **KOVÁCS, Z.** (2008): A talajprofilmérés gyakorlati módszerei. *Talajvédelem különszám, Talajtani Vándorgyűlés 2008. május 28-29.* 609-616. o., ISSN: 1216-9560
9. **KOVÁCS, Z.** – LAIB, L. (2009): A rugózás hatása a traktor első kerekeinek és alvázának függőleges lengéseire. *GÉP*, LX. Évf. 2009/12., 39-42. o.

Conference proceedings in foreign language

10. CZIRJÁK, T. - KISS, ZS. P. - **KOVÁCS, Z**. - SZŐLLŐSI, I. (2001): The effect of uncultivation and green manuring on soil resistance and soil humidity. *Slovak Agricultural University in Nitra*, October 26th, 2001., pp. 80-89., ISBN 80-7137-931-X

11. KISS, ZS. P. - **KOVÁCS, Z.** - SZŐLLŐSI, I. (2001): Traditional Cultivation and Direct Sowing in Relation to Soil Compactness. *IV. International Multidisciplinary Conference 4th edition*, BAIA MARE, May 25-26, 2001., pp. 280-285., ISSN 1224-3264
12. **KOVÁCS, Z.** – LAIB, L. – SZENTE, M. (2003): Comparison of Energy Balance of Spring Supported and Not Spring Supported Front Bridged Tractors. *V. International Multidisciplinary Conference 5th edition*, BAIA MARE, May 23-24, 2003. (Ed. Dan C. P. et al) pp. 275-278. ISSN 1224-3264
13. SZŐLLŐSI, I. – **KOVÁCS, Z.** (2003): The Examination for Penetration Values for Clay Soils and their Conversion into an Indentical Humidity Level. *V. International Multidisciplinary Conference 5th edition*, BAIA MARE, May 23-24, 2003. (Ed. Dan C. P. et al) pp. 495-501. ISSN 1224-3264
14. **KOVÁCS, Z.** – LAIB, L. – SZENTE, M. – KASSAI, ZS. (2004): Vibration testing of agricultural power machines. *microCAD 2004 International Scientific Conference*, Miskolc, 18-19 March 2004, pp. 103 – 106., ISBN 963 661 618 3
15. **KOVÁCS, Z.** – LAIB, L. – SZENTE, M. (2004): Investigations of the pull force of spring supported front bridge tractors. *V. INTERNATIONAL STUDENTS CONFERENCE*, Faculty of Agriculture, University of South Bohemia České Budějovice, April 20th, 2004. pp. 155 – 159. ISBN 80-7040-677-1
16. **KOVÁCS, Z.** (2005): Spring supported front bridge as the possibilty of increasing in pulling force. *VI. INTERNATIONAL SCIENTIFIC-TECHNICAL CONFERENCE FOR V4 PhD STUDENTS,* Technical University of Kosice, Faculty of Manufacturing Technologies with a seat in Presov, Herlany, 2.-4. 5. 2005., pp. 126-129. ISSN 1335-3799
17. **KOVÁCS, Z.** (2005): Measurement of the terrain profile and the effect of it for pulling force of the tractors. *VI. International Multidisciplinary Conference 6th edition, 1st volume*, BAIA MARE, May 27-28, 2005. pp. 371-374. ISSN-1224-3264, ISBN 973-87237-1-X
18. **KOVÁCS, Z.** (2007): Influence of spring supported front bridge for the pulling force of JD6920S tractor. International Multidisciplinary Conference 7th edition, 2nd volume, BAIA MARE, May 17-18, 2007. pp. 363-368., ISSN-1224-3264

Conference proceedings in hungarian language

19. KISS, ZS. P. **– KOVÁCS, Z. –** SZŐLLŐSI, I. (2002): Különböző talajjavító vetésforgók hatása a talajtömörödöttségre. SZIE Gazd. és Mg.-i Főiskolai Kar, Gyöngyös, 345-351. o., ISBN 9639256889

20. **KOVÁCS, Z.** – SZŐLLŐSI, I. (2003): Mezőgazdasági erőgépek lengéstani viselkedésének vizsgálata. *MTA AMB Kutatási és Fejlesztési Tanácskozás*, Gödöllő, SZIE Gépészmérnöki Kar – FVM MGI, 2003. január, 2. kötet, 200 – 203. o.
21. SZŐLLŐSI, I. – **KOVÁCS, Z.** (2003): Homoktalajok penetrációs értékeinek vizsgálata azonos nedvességi szintre történő átszámítással. *MTA AMB Kutatási és Fejlesztési Tanácskozás*, Gödöllő, SZIE Gépészmérnöki Kar – FVM MGI, 2003. január, 2. kötet 204 – 208. o.
22. SZŐLLŐSI, I. – **KOVÁCS, Z.** (2004): A parlagoltatásos zöld, és gyökértrágyázásos vetésforgók összehasonlítása a talajtömörödöttség tükrében. *IX. Nemzetközi Agrárökonómiai Tudományos Napok*, Károly Róbert Főiskola, Gyöngyös, 2004. március 25-26., 133.o. (CD)
23. **KOVÁCS, Z.** (2007): A vontatási teljesítmény alakulása a rugózott első híddal szerelt traktoroknál. *„Versenyképes mezőgazdaság" konferencia kiadványa 2007. november 29.*, Nyíregyházi Főiskola, 101-104. o., ISBN 978-963-7336-80-5

Conference abstracts in hungarian language

24. **KOVÁCS, Z.** – LAIB, L. – SZENTE, M. – KASSAI, ZS. – SZŐLLŐSI, I. (2004): A rugózott mellsőhíddal szerelt traktorok vontatási vizsgálata. *MTA Agrár Műszaki Bizottság, XXVIII. Kutatási és Fejlesztési Tanácskozás*, Gödöllő, 2004. január 19-20., 56. o., ISBN 963 611 406 4
25. SZŐLLŐSI, I. – **KOVÁCS, Z.** – BÍRÓ, D. (2004): Agyagtalajok penetrációs értékei. MTA Agrár Műszaki Bizottság, *XXVIII. Kutatási és Fejlesztési Tanácskozás* (poszter, kiadvány), Gödöllő, 2004. január 19-20., 15. o., ISBN 963 611 406 4
26. **KOVÁCS, Z.** – LAIB, L. (2010): A rugózott elsőtengely hatása a traktor lengésjelenségeire és vonóerőkifejtésére segédelsőkerék-hajtás esetén. *XXXIV. Kutatási és fejlesztési tanácskozás*, MTA Agrárműszaki Bizottság, Gödöllő, 2010. február 2., 38. o., ISBN 978-963-269-165-7

Others

27. **KOVÁCS Z.** (2001): Terepen mozgó járművek mozgékonysági vizsgálata, különös tekintettel a szélsőséges terepviszonyokra. *A Magyar Tudomány Napja*, Nyíregyházi Főiskola – Debreceni Egyetem – MPKITK, 2001. október 29. Elhangzott előadás
28. **KOVÁCS, Z.** (2002): Mezőgazdasági erőgépek lengésvizsgálata. *Magyar Tudomány Napja*, Nyíregyháza, NYF – DOTE Egészségügyi Főiskolai Kar

– Megyei Pedagógiai, Közművelődési Intézet és Továbbképző Központ, 2002. november 11-én elhangzott előadás
29. SZŐLLŐSI, I. – **KOVÁCS, Z.** (2002): Különböző talajnedvességi értékeken mért ellenállás-értékek azonos nedvességi szintre történő átszámítása. *Magyar Tudomány Napja*, Nyíregyháza, NYF – DOTE Egészségügyi Főiskolai Kar – Megyei Pedagógiai, Közművelődési Intézet és Továbbképző Központ, 2002. november 11-én elhangzott előadás
30. SZŐLLŐSI, I. – **KOVÁCS, Z.** – BÍRÓ, D. (2003): Különböző talajnedvességi szintek mellett mért talajellenállás értékek átszámításának módszerei vályogtalajnál. *Magyar Tudomány Napja*, Nyíregyháza, NYF – DOTE Egészségügyi Főiskolai Kar – Szent Atanáz Görög Kat. Hittudományi Főiskola – Megyei Pedagógiai, Közművelődési Intézet és Továbbképző Központ, 2003. november 11-én elhangzott előadás
31. **KOVÁCS, Z.** (2009): Járműelektromos szerelő szakmai ismeretek. *FVM Vidékfejlesztési, Képzési és Szaktanácsadási Intézet.* Budapest, 2009. 9 A/5 nyomdai ív (111 o.)
32. KEREKES, B. et al. (2007): A környezetvédelem technikai alapjai. II. Vízkezelés és talajvédelem. Bessenyei György Könyvkiadó, Nyíregyháza 2007. 14 A/5 nyomdai ív (171 o.)
33. **KOVÁCS, Z.** – LUX, M. (2010): Könnyű ejtősúlyos berendezéses mérések és penetrométeres vizsgálatok összehasonlító elemzése a tömörödöttség, talajtípus és nedvességtartalom függvényében a DANHAUSER vizsgáló berendezésen. *SSTI Projekt*, Kutatási jelentés a MICHELIN Hungária Kft. részére.
34. KRISTON, S. – **KOVÁCS, Z.** (2011): Laza talajok viselkedésének meghatározása gumiabroncsokkal való kölcsönhatás során. *SSTI Projekt,* Kutatási jelentés a MICHELIN Hungária Kft. részére.