VAKOLA
03450 OLKKALA
913-46211
VALTION MAATALOUSKONEIDEN TUTKIMUSLAITOS
FINNISH RESEARCH INSTITUTE OF ENGINEERING IN AGRICULTURE AND FORESTRY

STUDY REPORT 26
TUTKIMUSSELOSTUS

JUKKA AHOKAS - RISTO SALMINEN

AGRICULTURAL TRACTOR HITCH-HOOK LOADING AND LOCATION

TRAKTORIN VETOKOUKUN KUORMITUKSET JA SIJAINTI

VIHTI 1981

NKJ 42
REPORT 7

STUDY REPORT 26
Tutkimusselostus

Jukka Ahokas - Risto Salminen

AGRICULTURAL TRACTOR HITCH-HOOK LOADING AND LOCATION

Traktorin vetokoukun kuormitukset ja sijainti

Vihti 1981

NKJ 42
Report 7

FOREWORD

This study is a part of the Nordic project NKJ-42 (Nordiska Kontaktorgan för Jordbruksforskare) which deals with a connection between tractor and its implements. This report is based on a diploma-work "A study of tractor hitch hook" made by Risto Salminen.

The study has been made in VAKOLA, Finnish Research Institute of Engineering in Agriculture and Forestry and is financed by The Academy of Finland and partly by VAKOLA.

Ylö-tehtaat Oy has given some help by constructing a special hitch hook for this study.

The English version of the report has been revised by Pekka Olkinuora.

Vihti, December 1981

Jukka Ahokas Risto Salminen

FOREWORD

CONTENTS

1. QUICK-COUPLERS IN FINLAND

1.1 Number of users

It has been estimated that there are some thousands of quick-couplers in Finland. The amount of them is continuously increasing because some new tractors have quick-couplers as standard equipments.

1.2 Experiences of use

To evaluate the experiences of quick-coupler users 17 of them were interviewed. Seven of those used Normet-system, nine Walterscheid and one LTN-system.

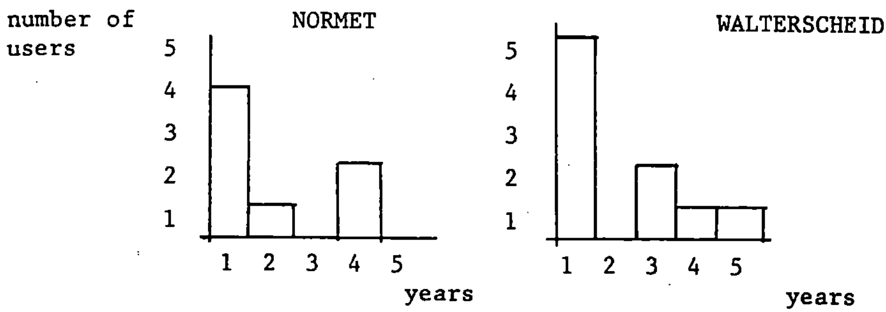
Quick-couplers are mainly bought to big, well-profiting farms. So were also these farms, whose fieldareas varied from 40 to 210 hectar. The Walterscheid users, nine farmers, were all in southwestern Finland and Normet-users in eastern Finland.

system	average area in total	average field-area
LTN		100 ha
Normet	161 ha	34 ha
Walterscheid	74 ha	39 ha

Table 1. The average field and total areas of the farms interviewed.

Most farms had more than one tractor. The quick-couplers were usually mounted on the newest tractors, but more than half of the "old" tractors were also equipped with the same kind of quick-couplers.

The period for which quick-couplers had been in use varied as follows. LTN-system had been in use only for half a year.



As reasons for bying quick-couplers were mentioned neighbouring's good experiences of use and also own wish for easier coupling. Also arguments of salesmen on farmshows had led into acquiring a quick-coupler.

1.2.1 LTN-system

LTN-system consists normally of quick-coupling devices on the top link and on the lower links. Additionally to those there was also hydraulic top link length control and hydraulic control on right lifting rod length. The amount of true experiences of use was very small because a system had been used only for one autumn and winter. No misfunctions of coupling devices had occurred. Some implements did not provide enough room around connection pins due to not meeting the respective standard. A hydraulic implement control had proved to be easy to use and quite accurate.

1.2.2 Normet

As advantages of the system it was mentioned that coupling events become safer and faster to do. The main disadvantages were the triangles which had to be fixed to the implements and which move the implements' center of gravity rearwards. A coupling does not work well if the centerlines of the tractor and implement do not coincide.

Users were of the opinion that function capabilities of implements were not reduced when using Normet-couplings. Only in one case the power lift capacity was not sufficient for lifting the implement with Normet-couplers.

As a method of improvement many users mentioned a need of a hydraulically controlled top link. No remarks were done concerning the locking devices.

1.2.3 Walterscheid

Like Normet-users the Walterscheid-users valued the ease of use of the coupling and it's safety. By using the Walterscheid-couplers the coupling can usually be done very well. Only if the centerlines of the tractor and the implement are not parallel some problems occur. Very few difficulties in opening of the locks and in some cases also in locking were observations of many users. Part of these were caused by disconnecting cables, which did not work satisfactorally. In some implements there was not enough room around connecting pins for Walterscheid claws.

About a half of the users were of the opinion that it is necessary to clean the coupling devices during wintertime. Some of them stated that the cleaning is necessary also in the summertime. One user had observed a fast wear of implements connection pins.

2. CONNECTION OF TOWED IMPLEMENTS TO A TRACTOR

2.1 Requirements for the coupling

To guarantee a safe and function capabilities at the connection some requirements for the coupling must be stated:

1. The coupling may not limit the use of the tractor or the implements
   - a turning angle between tractor and implement must be sufficient, stated in ISO 500

2. The coupling must be safe
   - strength of coupling devices must be sufficient
   - steerability of the tractor must remain in all conditions
   - a safety of working must be secured when coupling implements.

Only the mechanical part of the connection has been examined in this study. Possible hydraulic, electric and remote control connections are excluded. There are mainly four different systems in a general use, fig. 2:

a. upper clevis
b. cross bar
c. draw bar
c. hitch hook

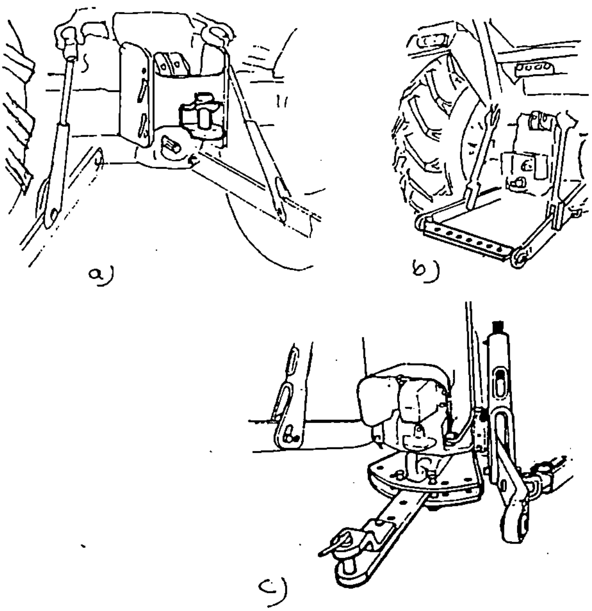


Fig. 2. Coupling systems

In Finland the cross bar and the hitch hook are used generally. Only a few farmers use a draw bar instead of a cross bar. It depends mainly on the weight of the implement whether it is coupled to the hitch hook or to the cross bar. Also constructional factors effect on this.

Light implements, as balers and towed drills are connected to a cross bar, but heavier ones, as trailers to a hitch hook. The upper clevis is not legal in Finland and therefore not used.

In other countries the use of the coupling systems vary from the Finnish way of use. The drawbar is used very widely so that the cross bar and also the hitch hook are not as common. In Central Europe two axle trailers are common and they are connected usually to the upper clevis.

From wide and very varying way of use of the different coupling systems a conclusion that none of them is superior to the others or not able to fulfil its demands can be made.

Following advantages and disadvantages can be found when comparing systems:

Cross bar

+ easy hydraulic height control
+ no problems on yoke angles of PTO drive shafts
+ allows wide turning angles
- a danger of breaking the PTO drive shaft exists when lifting the power lift
- the pin has to be inserted manually.

Draw bar

+ no problems with PTO drive shafts
+ allows wide turning angles

- the height of the drawbar is fixed, so that the height correction, when coupling implements, must be done by changing the height of implements draw bar
- the accuracy need when coupling is quite high
- the pin has to be inserted manually.

Hitch hook

+ coupling is safe and partly automatic, a driver can stay in the cab
+ because of hook's firm construction, heavy loadings can be permitted

- a coupling event can not be seen due to hook's location so near the rear axle and due to invisible master
  quard of PTO
- turning angles between the tractor and implements are smaller than with the other systems
- there are problems on yoke angles of PTO drive shafts in some cases.

Upper clevis

+ a good vision from the operator's seat
+ a good weight transfer from tractor's front axle to rear axle when traction force is increasing

- because of good weight transfer a steerability of the tractor can be lost
- some problems on yoke angles of PTO shafts like with hitch hook
- a controlled coupling event requests easy change of the height of implements drawbar
- PTO drive shaft coupling is difficult.

There are no great differences between various systems. The only essential difference is the adjustement of heights of the coupling devices. To control the height of heavy implement's drawbar is quite a troublesome task.

2.2 The use of PTO drive shafts

The output angular acceleration of Cardan universal joint can be calculated as follows /19/:

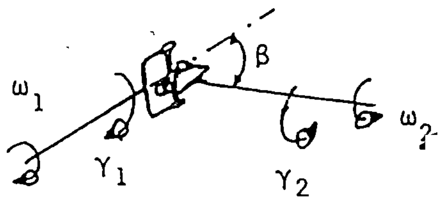


Fig. 3. Cardan joint

$$\dot{w}_2 = \frac{w_1^2 \cdot \cos\beta \cdot \sin^2\beta \cdot \sin 2\gamma}{(1 - \sin^2\beta \cdot \sin^2\gamma)^2} \qquad (1)$$

where

$\dot{w}_2$	output angular acceleration (rad/s^2)
w_1	input angular velocity (rad/s)
β	joint angle
γ	angle of rotation

The output angular acceleration reaches its maximum values when angle $\gamma = 45°$.

A shaft with two Cardan yokes can be reduced to be one equivalent yoke with joint angle /19/:

$$\psi_{ekv} = \sqrt{|\beta^2 - \psi^2|} \tag{2}$$

where β = angle of the first yoke (rad)
ψ = angle of the second yoke (rad)

For agricultural machines the practical limits of maximum angular accelerations are at implement input shaft (PIC) 1400 rad/s^2 and at the connecting shaft between PTO and PIC 3000 rad/s^2. /19/.

Figure 5 shows a dependence between ψ_{eq} and turning angle β with various l, l_2 relations.

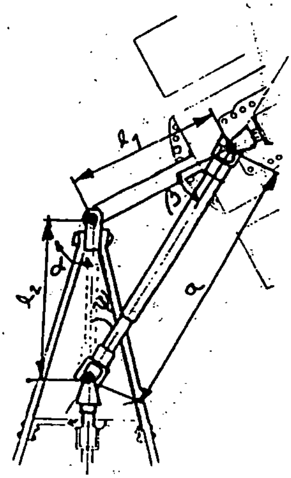


Fig. 4 PTO driven shaft

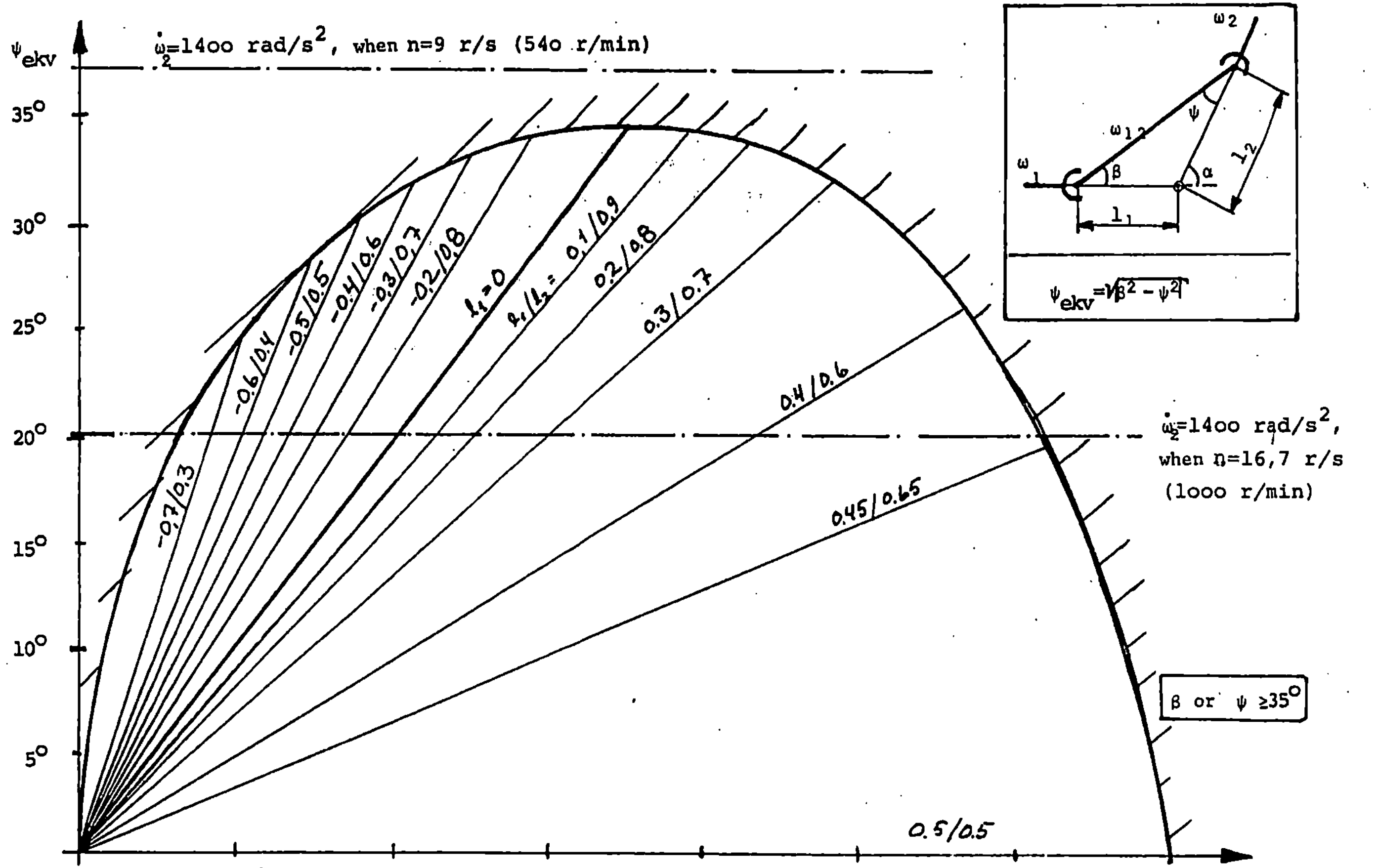


Fig. 5. A dependence between ψ_{eq} and turning angle β

In the same figure the equivalent joint angles on which the maximum accelerations at PIC are not exceeded are shown. Using rotation speed of 9 r/s (540 r/min) the maximum acceleration appears at equivalent angle of about 37 ° and at higher 16,7 r/s (1000 r/min) speed at 20,2 ° angle.

The maximum angular acceleration 3000 rad/s^2 at the connecting shaft limits the angle of the first yoke less than 29,3 ° when using 16,7 r/s rotation speed. At 9 r/s speed a theoretical limit is higher than 35 °, which is the widest joint angle that can be used with a normal Cardan joint.

If we want a 60 ° turning angle between a tractor and implements, as stated in hitch hook standard, the pivot point must be within 16 % accuracy in the middle of the connecting shaft if normal Cardan joints are used. Otherwise maximum joint angles 35 ° are exceeded. In addition to this limitation, the maximum acceleration of the connecting shaft limits the first joint angle to be less than 29,8 °, when using 16,7 r/s (1000 r/min) rotation speed.

The limitations mentioned above can be reduced much by using wide angle joints as in fig. 6. Both the output and connecting shaft angular velocities of wide angle axle are constant.

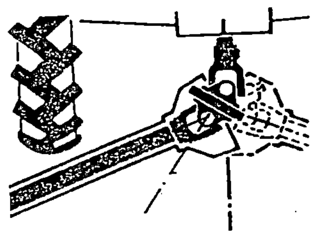


Fig. 6 Wide angle joint

Wide angle joints can be used nearly in all connections if the pivot point between the tractor and the implement is in the middle of the first yoke or to the rear of it. Only a bigger size of the yoke and smaller static turning angle may cause problems.

3. TRACTOR-TRAILER COMBINATION

3.1 Static case

3.1.1 Free body diagrams

For a tractor and its trailer a free body diagrams as shown in fig. 7 can be done.

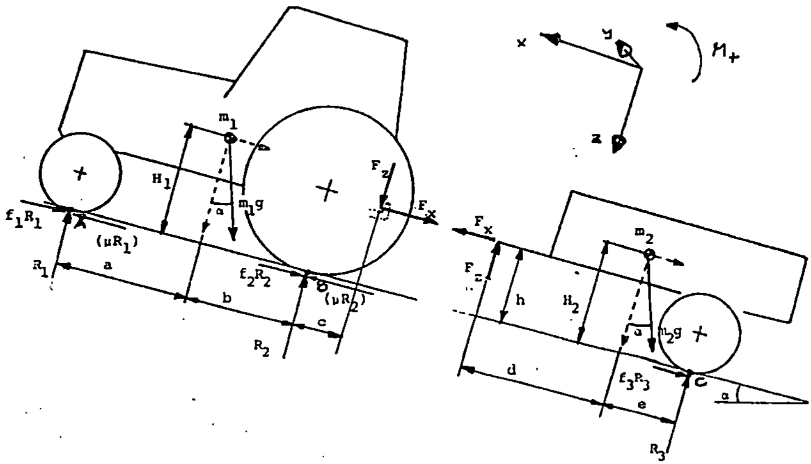


Fig. 7. Free body diagrams

If $(d+e) = l_z$ and $e/l_2 = \gamma_2$, can F_x and F_z be solved from equilibrium equations to the following mode:

$$F_x = m_2 g \cdot \{\sin\alpha + \cos\alpha \cdot f_3 \cdot (1 - \psi_2)\} \quad (3)$$

$$F_z = m_2 g \cdot \{\sin\alpha \cdot (\frac{h}{l_2} - \frac{H_2}{l_2}) + \cos\alpha \cdot \psi_2\} \quad (4)$$

Terms $f_3 \cdot h/l_2$ and $f_3 \cdot H_2/l_2$ have not been taken into consideration because of their minor importance.

The tractor's front and rear axle loads can be solved from its equilibrium equations as follows:

$$R_1 = m_1 g \cdot \cos\alpha \cdot \psi_1 - m_1 g \cdot \sin\alpha \cdot \frac{H_1}{l_1} - F_z \cdot \frac{c}{l_1} - F_x \cdot \frac{h}{l_1} \qquad (5)$$

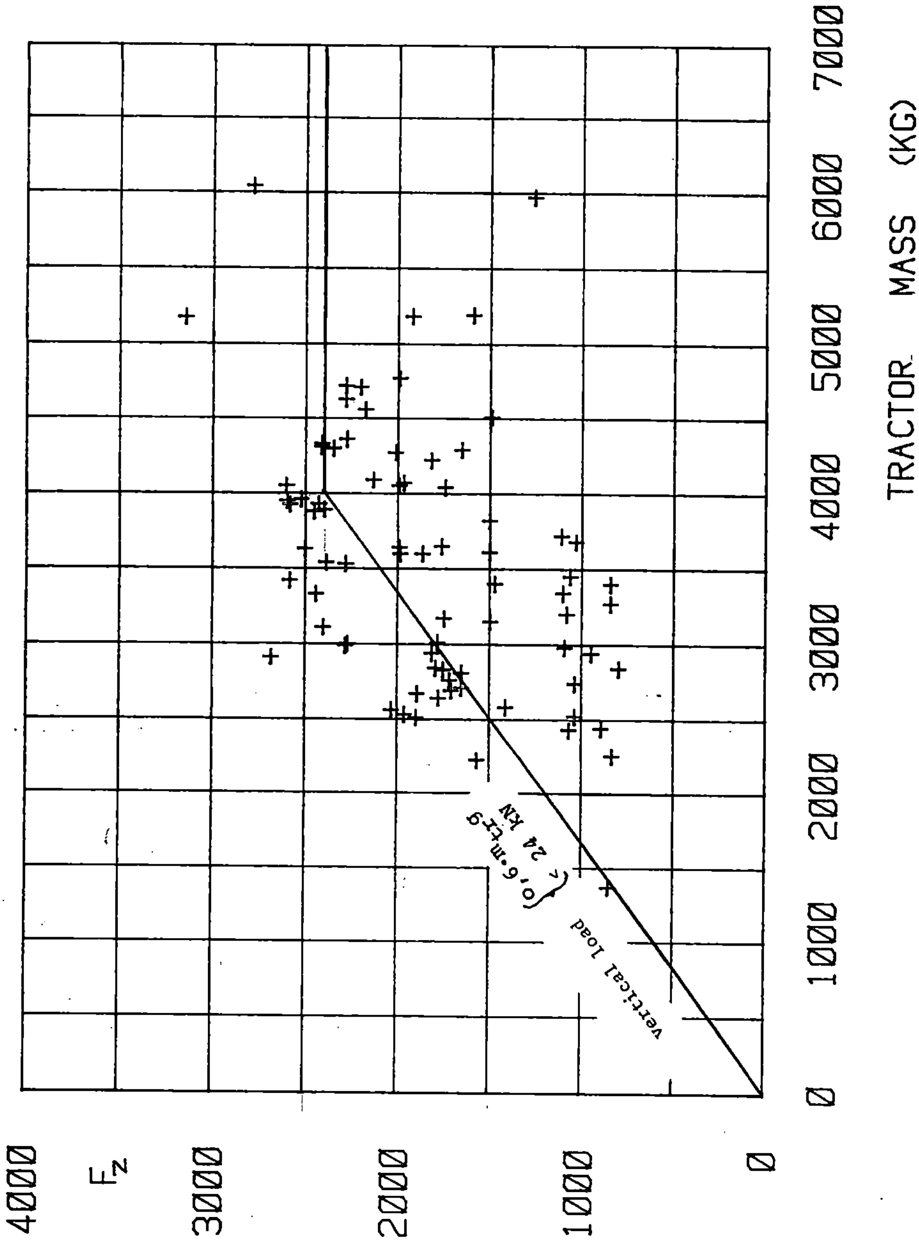
$$R_2 = m_1 g \cdot \cos\alpha \cdot (1 - \psi_1) + m_1 g \cdot \sin\alpha \cdot \frac{H_1}{l_1} + F_x \cdot \frac{h}{l} + F_z \left(1 + \frac{c}{l_1}\right) = 0 \qquad (6)$$

With equations (3) - (6) it can be calculated how changes in a trailer and a tractor do effect on the hitch hooks loadings and on the axle loads of the tractor.

3.1.2 Carrying capacity of tractor's rear axle

The load carrying capacity of tractor's rear axle sets a limit to the vertical load on the hitch hook. This limit is determined either by the strength of the rear axle or by rear tyre load carrying capacities. In fig. 8 the maximum permitted vertical loadings on the hitch hook of 78 tractors are shown.

In calculation the tractor rear tyres were regarded as 8 PR-tyres.


TRACTOR MASS (KG)
0
1000
2000
3000
4000
5000
6000
7000
F_N
0
1000
2000
3000
4000
vertical load
$0,6 \cdot m_{tr} \cdot g$
< 24 kN

In Finland weight distributions of one axle trailers are such that the static vertical load on the drawbar is from 15 to 22 percent of the trailer mass. However the maximum loads are less than 23 kN. When the mass of the trailer is three times the tractor mass, which mass is the heaviest legal trailer mass in Finland, the static vertical load on the hitch hook is from 4,5 to 6,6 times tractor mass (in kilonewtons, tractor mass in kilograms).

3.1.3 Traction force of the tractor

When a better mobility of the tractor-trailer combination is desired, a traction force of the tractor can be increased with weight transfer to tractor rear axle. Extra weight to rear axle can be transferred either from the trailer or from the front axle of the tractor. The first alternative is possible, when using one axle trailers and the second when using a two axle trailer.

3.2 Dynamic situation

3.2.1 Vertical and pitch vibrations

A tractor-trailer combination can be analyzed with a model on which the tyres have been replaced by springs and viscous dampers, fig. 9.

If the model is used to estimate the loadings on the connection devices, a clearance on them brings problems that are very difficult to solve.

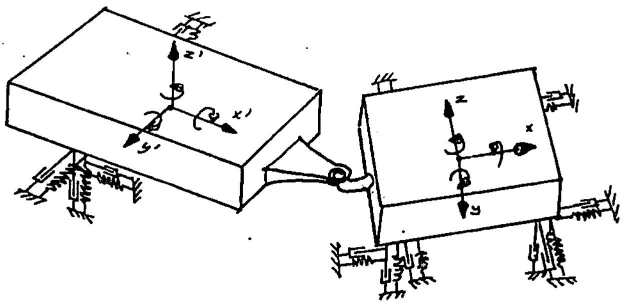
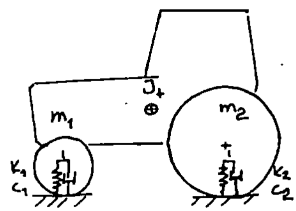


Fig. 8. A theoretical model for tractor and trailer vibrations

The natural frequencies of the tractor axle vibrations can be calculated with equation (7):



$$f_n = \frac{1}{2\pi}\sqrt{\frac{c_n}{m_n}}$$

Fig. 9 A simplified model for tractor vibrations

where c = spring constant
m = axle load
n = index of axle (1, 2)

The natural frequency of the pitch vibration can be calculated with following equation:

$$f_p = \frac{1}{2\pi}\sqrt{\frac{a^2c_1 + b^2c_2}{J_t}} \tag{8}$$

where a = distance between front axle and c · g.
b = distance between rear axle and c · g.
I_{tr} = moment of inertia at the tractor

For instance the calculated natural frequencies of Volvo 2654 -tractor are:

f_1 = 2,7 Hz
f_2 = 2,0 Hz
f_p = 3,1 Hz

3.2.2 Longitudal vibrations

A simplified model in fig. 10 describes a tractor-trailer combination in longitudinal vibration. A natural frequency of it is /3/:

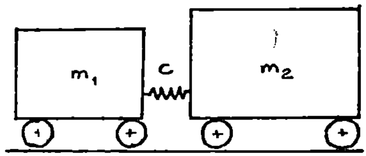


Fig. 10. A model for longitudinal vibration

$$f_p = \frac{1}{2\pi}\sqrt{c \cdot \frac{m_1 + m_2}{m_1 m_2}} \qquad (9)$$

If a spring constant c is $\frac{m_1 \cdot m_2}{m_1 + m_2} \cdot g / X_o$

(X_o is the maximum deviation), the maximum connection force is /3/:

$$F_{max} = \frac{m_1 m_2}{m_1 + m_2} \cdot g \qquad (10)$$

The equation (10) is used in road traffic legislations of many countries to dimension coupling devices. In ISO the French have proposed it to be the testing force of the tractor hitch hook.

3.3 Driving safety of the combination

3.3.1 Load on the front axle

A driving stability depends strongly on the instantaneous front axle load. In most cases steerability is estimated by using examination of static front axle load. Based on this there are regulations in traffic legislations of many countries concerning the minimum permitted front axle loadings.

Another method for evaluating a steering ability has been proposed by MERTINS and ULRICH. /14/.

$$n = \frac{R_{eff}}{R_{st}} \qquad (11)$$

where R_{eff} = the effective value of wheel load vibration
R_{st} = a static wheel load

When a wheel load coefficient n is greater than 1/3, the steering ability of the tractor is not good enough to guarantee a safe drive.

In this study the front axle loadings of MF 575 - tractor were measured in different driving situations in order to evaluate driving stability with help of a dynamic wheel-load coefficient. In table 3 are the data of various drives.

Drive	Trailer mass m_2	Static load on hitch hook F_z(kN)	Static load on front axle R_f(kN)	R_f/G_{tr}1) (%)	$R_f/(G_{tr}+F_z)$(%)
1[2]	8,7	20,6	8,4	27,7	16,5
2	8,4	24,5	8,7	28,7	15,8
3	8,7	20,6	9,3	30,6	18,5
4	8,4	15,4	9,9	32,7	21,7
5	5,4	10,5	11,1	36,6	27,4

Table 3. The data of measurements

1) Tractor MF-575, G_{tr} = 31 kN, static front axle load 12,5 kN

2) a distance between the hitch hook and the tractor rear axle is 100 mm greater than in other trials. A horizontal distance between the hook and PTO is 100 mm.

A mass of the trailer, its weight distrubution and a distance from the hook to rear axle did vary in different trials. The main test track was a gravel road but also some measurements were done on ISO 5008-track. The speeds used on the road were normal transporting speeds, 10-20 km/h.

The results are shown in table 4.

Drive	Static front axle load R_f(kN)	Frequency of vibration (Hz)	Effective value of oscillation X_{rms}(kN)	Coefficient of dynamic wheel load $n = X_{rms}/R_t$
1	8,4	3,3	1,93	0,23
2	8,7	4,6	2,29	0,26
3	9,3	3,4	2,29	0,24
4	9,9	3,5	3,07	0,31
5	11,1	3,5	2,94	0,26

Table 4. Effective values and dynamic coefficienties of front axle oscillations of MF 575.

In all cases the dynamic coefficient n was smaller than 1/3 and so the driving security was obtained in all drives, when making measurements, there occurred no difficulties to change or hold the driving direction.

Based on the results measured a steering ability of the tractor-trailer combination was good enough if approximately over 15 % of the actual weight of the tractor (the tractor weight includes a load transfer from a trailer) remains on the front axle.

3.3.2 Distance from tractor rear axle to hitch hook

CROLLA and HALES /5/ have in their study delt with lateral stability of the tractor-trailer combination by developing a mathematical model for a combination and simulating it by a computer. From the results it can be found out that in low speeds, below 32 km/h (9 m/s) the placing of the pivot point between a tractor and a trailer has no great effect on the lateral stability of the combination. A tractor-trailer combination stays quite well stable if the distance from a tractor rear axle to the hitch hook is smaller than one meter.

3.3.4 Braking the combination

Very few tractor trailers in Finland are equipped with brakes. Braking can be done only with the tractor rear wheel brakes and so the braking forces are equal to the actual tractor rear axle weights. That is why a weight transfer from a trailer to a tractor ought to be as great as possible.

Although braking forces and decelerations of the combination are quite satisfactory when braking only with tractor a jack-knifing of the combination exists until the trailer is braked. The only way to prevent dangerous jack-knifing is to equip a trailer with brakes.

4. MEASUREMENTS OF THE LOADINGS ON HITCH HOOK

Loadings on the hitch hook were measured in normal driving situations with three different tractors - Ford 7700-4, Massey Ferguson 575 and Volvo 2654 - with various trailer weights. Measurements were carried out on several tracks: on a gravel road, on asphalt road, on field and on "rough" ISO 5008-track. Driving speeds were such that they are hardly exceeded in normal working conditions. Force transducers were built on Volvo's hitch hook and also on a coupling ring of the test trailer. When using the first transducers all forces (x, y, z) were measured but with a trailer transducer only those loadings directed in drive-line.

4.1 Vertical force F_z

When using one-axle trailers, a static vertical load on the hitch hook depends on the weight of the trailer and on the location of its axle. A dynamic loading depends, besides these, also on the terrain roughness, driving speed and on the vibrational properties of the tractor-trailer combination.

When a static vertical load on the hitch hook increases, the relative amplitude of the vibration decreases of course, the real peak values get higher.

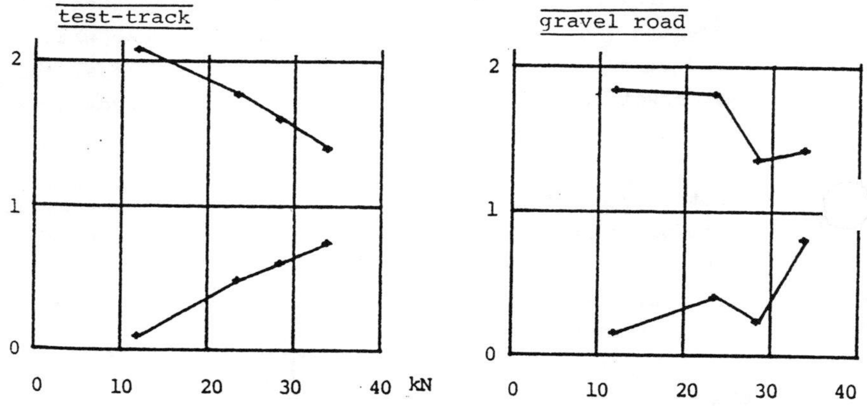


Fig. 11. A dynamic coefficient of the vertical loading as a function of static loading

Increasing the driving speed increases the peak values nearly in a linear dependence of the speed.

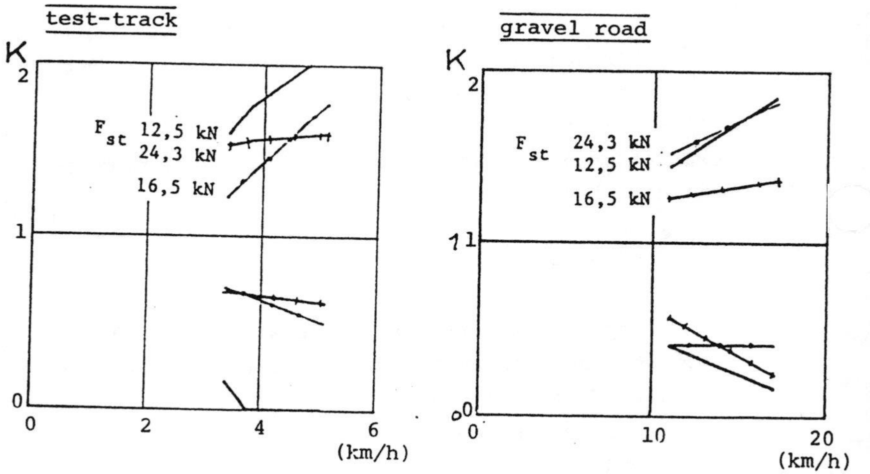


Fig. 12. A dynamic coefficient of the vertical loadings as a function of speed.

No significant differencies on the results measured on various tracks were found. This can be explained so that despite different profiles of the tracks, the input to the vibration system was of the same magnitude on all tracks because of different driving speeds. A value of 1,8 can be used well as a dynamic coefficient.

4.2 Lateral forces F_y

The values of lateral forces were equal to 0,5–0,6 times the longitudal forces when driving on a circle ISO 5008-track. The angle between a tractor and a trailer was about 30 degrees. Then the resultant of lateral and longitudal forces was almost in a direction of a center line of the trailer.

On the road lateral loadings of the hook are caused by side slopes of the road, by turns and by different rolling resistances of the trailer wheels. The values of lateral forces on the road driving are quite small. The measured peak values of the lateral forces were as follows.

Track, speed	Lateral force F_y (kN)	Longitudinal force F_x (kN)	Resultant R (kN)	Angle
Circle ISO 5008				
,4	10,4	20,0	22,5	27,5°
3,8	12,0	21,6	24,7	29°
5,1	9,0	15,0	17,5	31°
Gravel road				
15 - 20	5,5	39	39,4	8°

Table 5. The values of lateral forces on different speeds and tracks

4.3 Longitudinal force F_x

The dependence between a longitudinal force F_x and a mass of the trailer is shown in fig. 13.

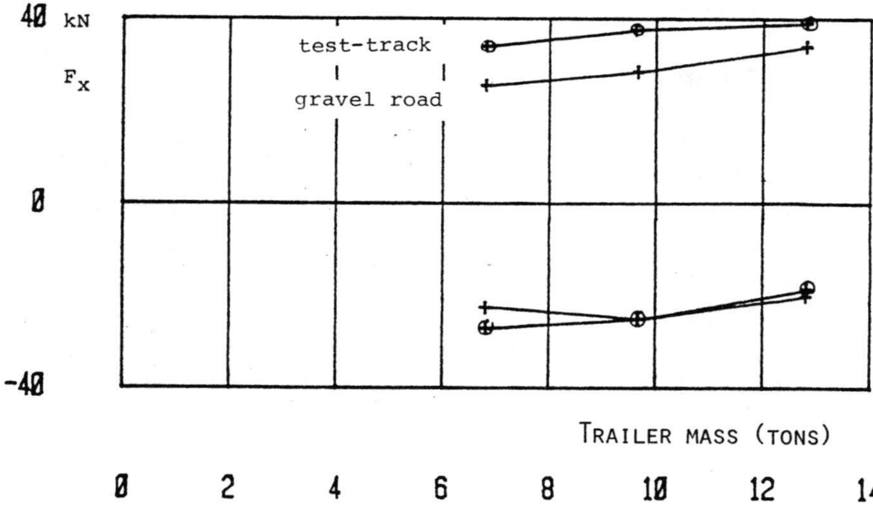


Fig. 13. The peak values of the traction force as a function of trailer mass.

A slight increasing of peak values can be found out when trailer mass is increasing. A varying weight distribution of the trailer had no effect on longitudinal forces.

Also a mass of the tractor has effects on the longitudinal loading of the hitch hook. In fig. 14 are shown the maximum values of the longitudinal force with different tractors. Whether the tractor was 2-wheel or 4-wheel driven no effect on the loadings of the hitch hook was observed.

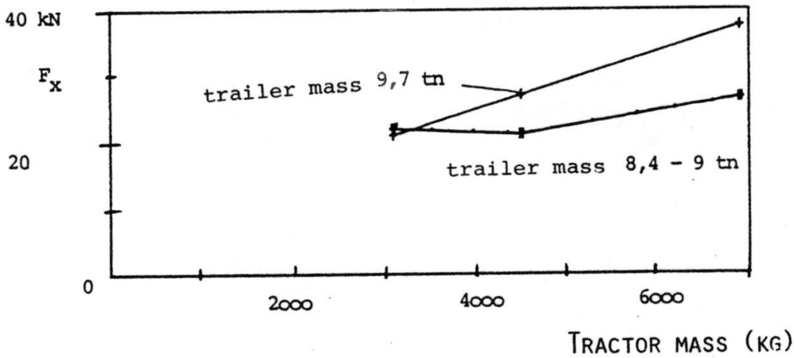


Fig. 14. The effect of the tractor mass on the maximum longitudinal loadings.

The maximum values of longitudinal forces F_x can be calculated with equation (11) in quite good accuracy

$$F_{max} = \frac{m_1 m_2}{m_1 + m_2} \cdot g \qquad (11)$$

where m_1 = tractor mass

m_2 = trailer mass

Both the calculated and measured values of longitudinal forces are shown in fig. 15 as a function of trailer mass. Any significant differences can not be seen.

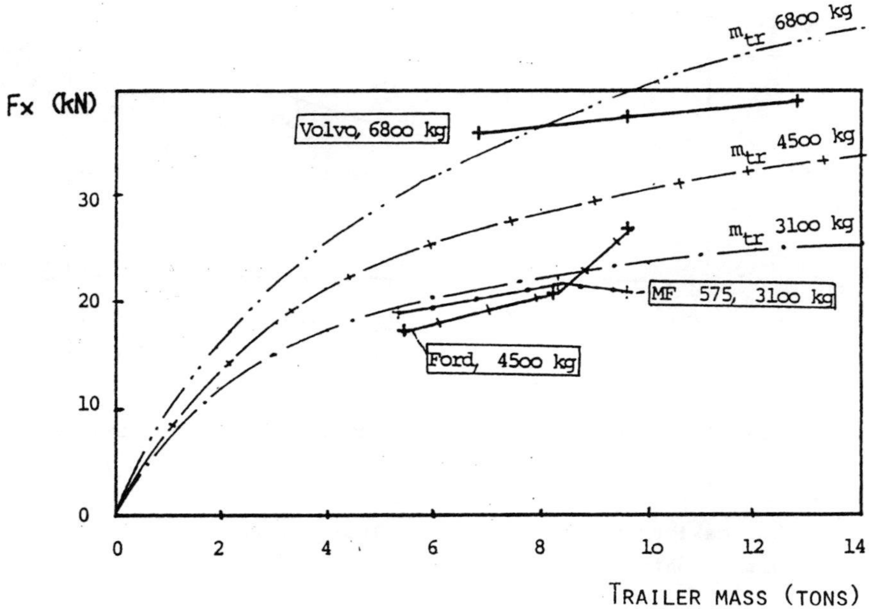


Fig. 15. Measured and calculated values of longitudinal loadings.

The clearances on coupling devices ought to be smaller than 10 mm in order to avoid rapid and high impulsive forces. On a clearance of about 15 mm were peak values nearly twice as h as on a 7 mm clearance.

The power spectra of longitudinal and vertical forces are shown in fig. 16.

The standardized hook and drawbar eye only allow minimum free play to avoid wear.

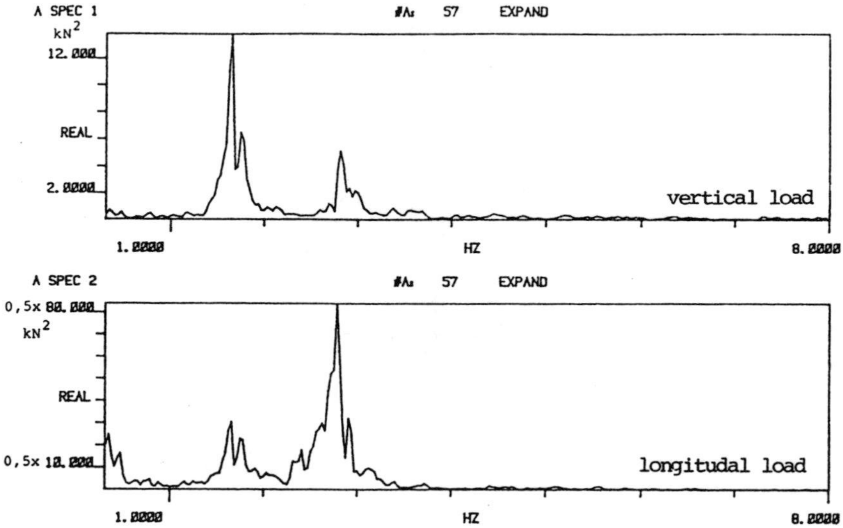


Fig. 16. a) Power spectrum of vertical force
b) Power spectrum of longitudal force.

Two specific frequencies can be found. On a vertical force a dominating frequency is about 1,7 Hz and on a longitudinal force it is about 2,8 Hz. An estimated value of the frequency of rear axle vertical oscillations was in this case 1,6 Hz, calculated with help of equation (9).

A correlation between longitudinal forces and tractor pitching seems to be obvious. The value of the natural frequency of pitching was estimated to be 3 Hz in chapter 3, which freq- vency is quite near to the now measured 2,8 Hz. COENENBERG /4/ has also found out a dependence of a longitudinal force and pitching.

4.4 The resultant of forces

The maximum values of longitudinal and vertical forces do not exist simultaneously. This feature can be explained with the dynamic behavior of the trailer. When to the coupling point of the trailer is directed a rapid impulse of a force, causes this decreasing of the support force F_Z.

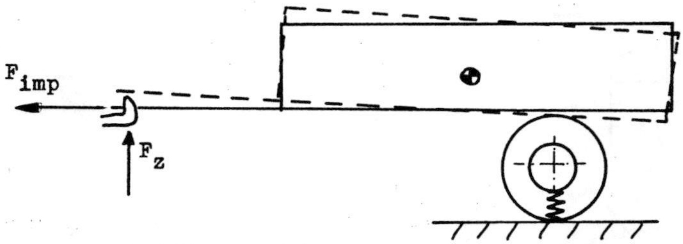


Fig. 17. Trailer in a transient case.

A typical area of the resultant is shown in fig. 18. On a form and a location of the area effect the factors as told above. Especially a static vertical load of the drawbar has great effects.

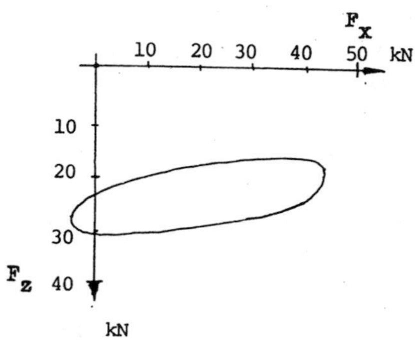


Fig. 18. The resultant of forces

5. THE STRENGTH OF COUPLING DEVICES

5.1 Hitch hook

Calculation of the stresses on a hitch hook are done by using the standard dimensioning shown in fig. 23. The Finnish and Swedish standards of the coupling devices state the maximum permitted load in a vertical direction to be 30 kN and in a longitudinal direction 60 kN. On these loadings the maximum stress on a point y in fig. 23 is 520 N/mm^2. If such a stress could be tolerated in a static case, ought the strength of the material to be as good as Fe 60. If a firmness against fatigue is wanted, the material must be hard, tempered steel. In the standards there are however not any demands for materials.

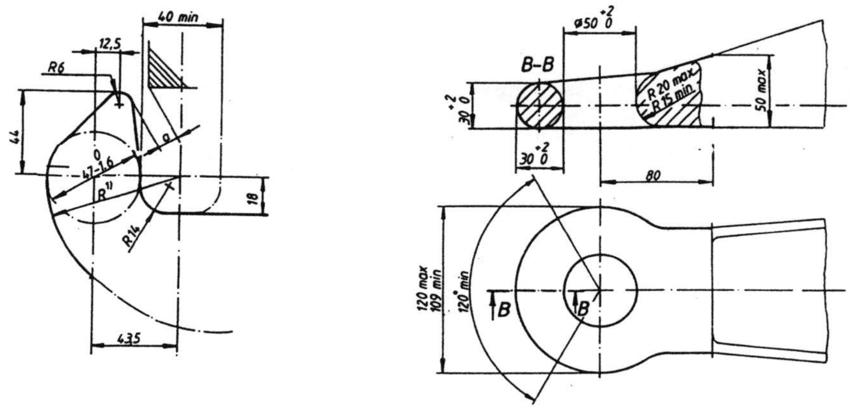


Fig. 23. Hook and a coupling ring.

If the stresses are calculated with maximum loads measured in this study F_x = 39 kN and F_z = 36 kN (static load 24 kN and a dynamic coefficient 1,5) the calculated stress is 340 N/mm^2. The firmness against fatigue is guaranteed with use of Fe 42.

5.5 Hitch ring

A point of the maximum stress of a hitch ring is shown in fig. 24. The maximum stress on a point A is, with loads F_x = 60 kN, F_z = 30 kN, 270 N/mm^2. The value of the stress is then just a little higher than a yield point of Fe 37. If a firmness against fatigue breaking is wanted under those loadings, a material must be Fe 60 or better. ISO standard needs a forged hitch ring.

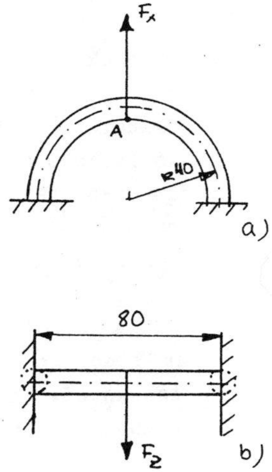


Fig. 24. Hitch ring

When stresses are calculated with maximum loadings measured in this study, a maximum stress on the coupling ring is 300 N/mm^2. The value is higher than the one calculated with standard loads. That can be explained by higher vertical loading that causes higher bending moment and also a higher stress. In practice the stresses are smaller because the maximum forces F_x and F_z do not exist simultaneously and also because the contact points of the forces do change because of wear.

5.3 Wear resistance of materials

When a particle slides on another, there occurs wearing on sliding surfaces. The wear rate depends deeply on the contact pressure of particles. Fig. 27 shows the correlation between contact pressure and a wear rate.

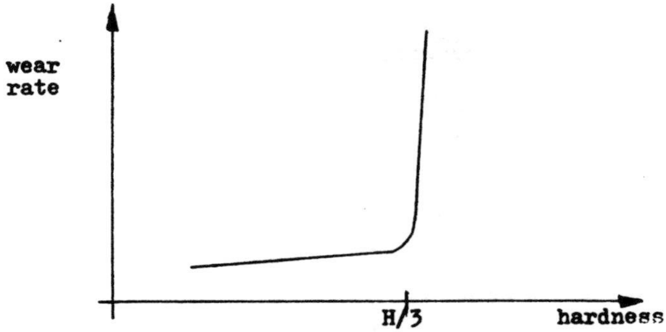


Fig. 27. Wear rate as a function of contact pressure

Rapid increasing of the wear can be seen when the contact pressure is higher than H/3, where H is the hardness of surfaces. A rough estimation $H = 3 \cdot \sigma$, where σ = yield point of the material, can be done /7/.

A theoretical contact pressure between a hook and a hitch ring is about 22,5 kN/mm^2 on 24 kN loading. That exceeds the strengths of the usual materials 5 - 6 times. In the beginning the parts wear rapidly. The contact area is however growing at the same time as wear occurs and the wearing speed is decreasing. A wear of 0,25 mm on a hitch hook and 0,5 mm on a ring is sufficient for area on which the contact pressure does not exceed the value of Fe 37 yield point. Fig. 25.

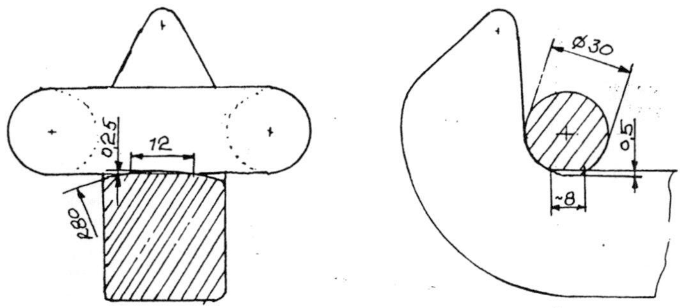


Fig. 25. Wear in coupling devices.

A condition for so equal wear on a hook and on a ring is that the hardnesses of both surfaces are quite similar.

To make wear resistances of the coupling devices better is difficult. In searching best price - safety - ratio it is wise to allow such a wear that occurs in a standard construction.

6. COUPLING EVENT

When connecting a trailer to a tractor are there several phases:

1. unlocking the hitch hook and lowering it

2. backing up a tractor so that the hitch hook is just under the coupling ring

3. lifting the hook until it is locked

4. driver is getting off the cab

5. connecting electricity, hydraulics and possible PTO and lifting a standing support

6. driver is getting back to the cab.

The accidents that occur when coupling implements to tractors represent about 10 to 30 percent of all tractor accidents. Almost a half of those coupling accidents occur when coupling a trailer. In many cases a lack of the good standing support has caused accidents.

A standing support ought to be easy to use. In addition it must be strong enough so that it can be used as a parking brake of the trailer.

Going on and off the cab creates many possibilities for accidents. According to some estimates, a half of a tractor usage is transporting and so coupling, and also getting on and off the cab, must be done often. A coupling that is made completely automatic decreases this danger.

An unsatisfactory visibility from the cab to the hitch hook is a problem of many tractors. However a driver should see the hitch hook when doing the connection. This demands also very difficult working positions. In addition to ergonomical factors, difficult working positions cause immediate dangers on using a clutch and brakes.

The location of the hitch hook so near the tractor rear axle and especially non-transparent master guards of PTO are the greatest obstacles for good visiblity. A sufficient visiblity is in most cases obtained if the hitch hook were moved 100 mm rearwards from PTO and the master guard of PTO is to some degree transparent, grille or net. This kind of transparent master shield is allowed in Finnish standards.

A succesfull coupling needs quite good accuracy in backing up the tractor. A radius of the top of the hook along with a round material of ring allows that the top of the hook may be within 40 mm accuracy in a center point of the ring and a coupling still can be done.

A wider guidance can be obtained by using separate guides for a ring. Guides shown in figures 26 and 27 were tested on the hitch hook of MF 575 tractor. Small, less than 50 mm, lateral errors were eliminated well by moving a hook within its clearances.

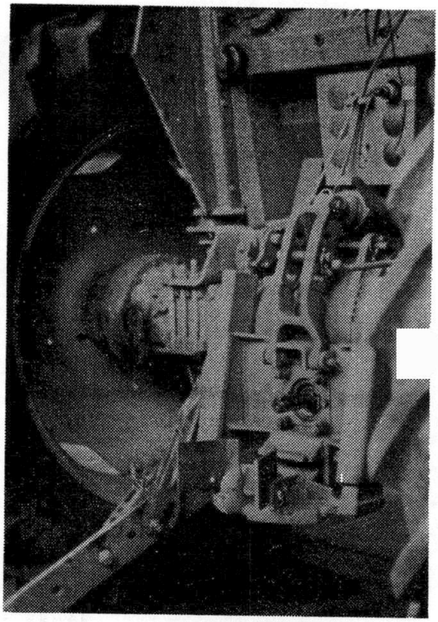


Fig. 26.

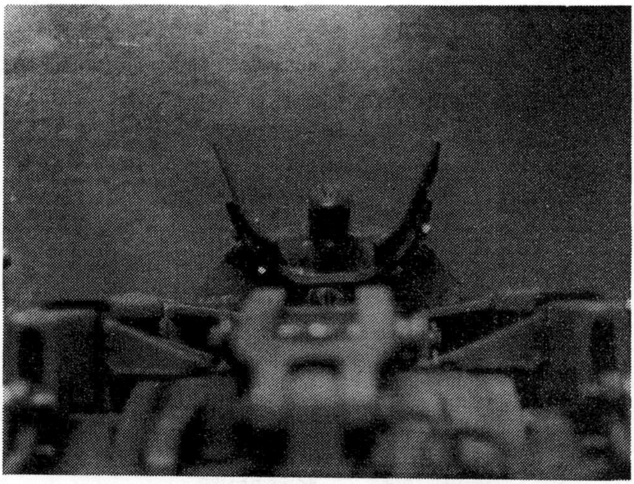


Fig. 27.

A height of the guides from a top of the hook must be 60-80 mm. The the use of guides is limited by the free space of PTO, definied in standard ISO 500 and shown in fig. 28.

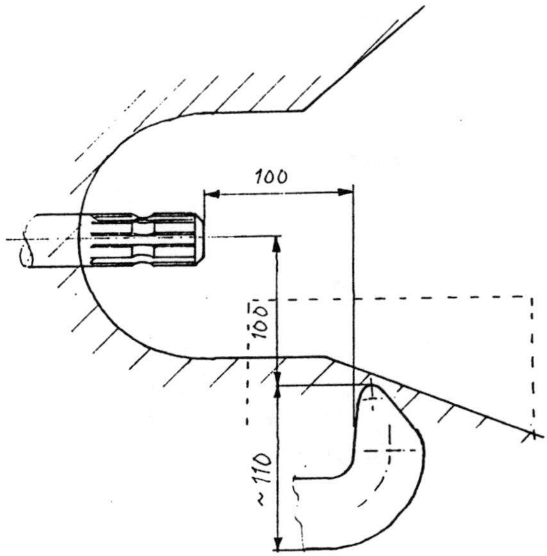


Fig. 28. A free space of PTO and a location of the hitch hook

If the hook is placed 100 mm rearwards of PTO and the free space with guides is obtained, the ground clearances of the tractors are 260-490 mm (the height of the hook from its top to the lowest point is approximated to be 110 mm and the height of PTO 575-775 mm). So small ground clearances can not be accepted.

The possibilities to use guide supports effectively without limiting the function abilities of tractors are quite small.

7. TEST METHOD AND PROPOSALS

7.1 Loadings on the coupling devices

Longitudinal forces

In quite a good accuracy can a longitudinal force F_x be calculated with help of the following equation

$$F_{max} = \frac{m_1 m_2}{m_1 + m_2} \cdot g$$

where m_1 = tractor mass (kg)
m_2 = trailer mass (kg)

If the trailer mass is equal to three times the tractor mass, which is the highest legal weight of a trailer in Finland, the equation (12) can be expressed in a form:

$$F_x = 7{,}5 \cdot m_1 \qquad (13)$$

Lateral forces

Magnitudes of lateral forces in a normal use are below 15 pe cent of actual longitudinal forces.

Vertical loadings

When constructing a trailer, can a static loading on its hitch ring be choosen easily by moving a location of trailer's axle. With no essential alterations on modern constructions a maximum static load on the hitch hook / coupling ring can be stated to be

$$F_{zmax} = \begin{cases} 6 \cdot m_1 \quad [kN] & , \text{ when } m_1 < 4000 \text{ kg} \\ 24 \text{ kN} & , \text{ when } m_1 \geq 4000 \text{ kg} \end{cases}$$

where m_1 = tractor mass (kg)

For the trailers that means (trailer mass is three times a tractor mass)

$$F_{zmax} = \begin{cases} 2 \cdot m_2 \quad [kN] & , \text{ when } m_2 < 12000 \text{ kg} \\ 24 \text{ kN} & , \text{ when } m_2 \geq 12000 \text{ kg} \end{cases}$$

where m_2 = trailer mass (kg)

7.2 Materials of the coupling devices

Based on chapter 5.5 the material of the coupling ring must be in any cases of Fe 42 or better. A sufficient strength against fatigue needs a use of Fe 52 on light trailers and stronger materials on heavy trailers.

The strength of hitch hooks is to be tested dynamically and so there is no need for regulations concerning materials of them.

7.3 Drawbar

Taking into account the driving security of average tractors, the maximum loadings on the drawbar can be about 15 kN on 3 ton tractors and about 10 kN on 4 ton tractors. Under such big loadings the drawbar must be very strong and the size of it becomes quite big. For instance on 15 kN loading the square profiled beam must be stronger than 130 x 50 mm.

With a maximum static loading of 5 kN it can be reached with most implements. Only trailers and heavy harvesting machines can not then be coupled on the drawbar.

7.4 Test method of hitch hook

A hitch hook testing must be done dynamically because the actual loadings are cyclic. According to chapters 4 and 7.1 the maximum loadings are

- on a vertical direction 1,8 times a static loading, which is 60 percent of tractor weight, maximum 24 kN
- on lateral direction ± 0,15 times longitudinal loading
- on longitudinal direction 7,5 times tractor mass.

The resultant of the maximum forces is shown in fig. 29. A magnitude of it is 1,32 · m_{tr} · g. Angles α and β are about 4,9 ° and 55,2 °.

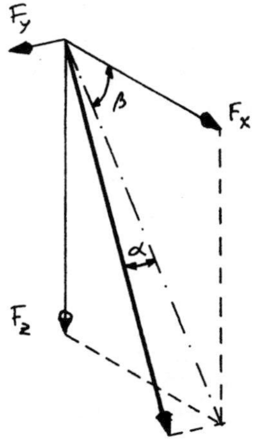


Fig. 29. Resultant of test force.

A test is possible to be done with one loading direction if angles α and β are constant. An amplitude of the resultant can be choosen so that the average vertical load is the same as static load on hitch hook, 6 m_{tr}. Then do the forces vary as follows

$$F_x = 0,8 - 7,5 \ m_{tr}$$
$$F_y = 0,1 - 1,1 \ m_{tr}$$
$$F_z = 1,2 - 10,8 \ m_{tr}$$

$$\bar{R} = 1,47 - 13,2 \ m_{tr}$$

A variation of the vertical load is quite the same as in normal use.

The loading must be applied 2 million times in order to ensure that the fatigue strength is sufficient.

7.5 Location of the hook

Taking into consideration the use of PTO driven implements, a visiblity of coupling event, a steering ability of a tractor and other things mentioned before, the optimum placing of the hitch hook would be 100 mm rearwards of the PTO and 100 mm below its center line. This dimensioning is shown in figure 28, on page 35. Using this dimensioning a drawbar can in many cases be replaced by the hitch hook. When testing the hitch hook, it must be tested so that load on the front axle of the tractor is 25 percent of tractor moss when a loading of 60 % of the tractor moss is on the hook. If needed this weight distribution must be secured with fixed front ballast.

SUMMARY

In the beginning of this study the experiences of the use of quick-couplers has been reported. As advantages were considered to be the ease and safety of coupling work. The disadvantages were very few, some malfunctions of coupling devices had occured and also in some implements the free space around the connecting pins did not meet the standards and then was not sufficient.

The second part of this work was to study the connection between a trailer and a tractor. Changing the hitch hook location could eliminate the need of the drawbar and crossbar.

A place of the hitch hook should be 100 mm rearwards of PTO in order to guarantee a sufficient visiblity to the hitch hook. This dimensioning makes also the coupling of PTO driven implements to the hitch hook possible.

A vertical loading on the hitch hook causes weight transfer from tractor's front axle to rear axle and so reduces the steering ability of the tractor. In measurements carried out in this study it was found out that the steerability was sufficient if over 15 percent of actual weight of the tractor lies on the front axle. The maximum vertical load on the hitch hook is proposed to be 20 % of the trailer weight to secure the lateral stability of the tractor-trailer combination.

Testing of hitch hook must be done dynamically. The resultant of the testing force is proposed to be $R = (0,73 \pm 0,59)$ $m_{tr} \cdot g$. A direction of it is 55 ^{o} downwards and 5 ^{o} to the side.

Many accidents that have occurred in coupling work have been caused by a poor standing support of the trailer. That is why it ought to be done firm, immovable and easy enough to use. Also the face area of it should be large enough to prevent sinking into the ground.

TIIVISTELMÄ

Tämän tutkimuksen alussa on käsitelty pikakytkentälaitteiden käyttöä Suomessa. Laitteiden parhaina puolina pidettiin kytkennän helppoutta ja turvallisuutta. Haittoina huomautettiin useimmiten työkoneiden liian ahtaasta rakenteesta kytkentätappien ympärillä, mutta myös joitakin huomautuksia tehtiin kiinnityselimien huonosta lukkiutumisesta tai avaamisesta.

Toinen osa tutkimusta käsittelee perävaunun ja traktorin kytkentää. Tietyin muutoksin voidaan vetotanko ja vetopuomi korvata vetokoukulla.

Vetokoukkuun perävaunusta kohdistuva pystykuormitus aiheuttaa painon siirtymistä traktorin etuakselilta sen taka-akselille ja näin heikentää traktorin ohjattavuutta. Tässä työssä tehtyjen mittausten perusteella traktorin ohjattavuus on riittävä kun sen etuakselilla on yli 15 prosenttia traktorin senhetkisestä painosta. Traktoriperävaunu -yhdistelmän ajoturvallisuuden takaamiseksi esitetään tutkimuksessa suurimmaksi sallituksi vetokoukun pystykuormitukseksi 20 % perävaunun painosta.

Vetokoukun koetus on tehtävä dynaamisesti. Koetusvoiman resultantiksi ehdotetaan $R = (0,73 \pm 0,59) \cdot m_{tr} \cdot g$, ja sen suunnaksi 55 ° taakse alas ja 5 ° sivulle.

Perävaunun kaatuva seisontatuki on aiheuttanut useita onnettomuuksia. Tämän vuoksi se on tehtävä lujaksi ja helpoksi käyttää sekä asennettava kiinteästi perävaunuun. Lisäksi sen pinta-alan tulee olla riittävän suuri.

REFERENCES

1. Barger, E., Tractors and their power units. 2 ed. New York, 1967, 524 s.

2. Bekker, M.G., Introduction to terrain-vehicle systems. The University of Michigan Press, Michigan 1969, 846 s.

3. Coenenberg, H.H., Das "äussere" Schwingungsverhalt von Ackerschleppern, inbesondere ihre dynamischen Achslasten. Landtechnische Forschung 12 (1962) 6, s. 157-165.

4. Coenenberg, H.H., Dynamische Beanspruchungen bei Ackerschleppern. Landtechnische Forschung 11 (1961) 6, s. 145--150, 12 (1962) 1, s. 7--11 and 2, s. 33--39.

5. Crolla, D.A., Hales, F.D., The lateral stability of tractor and trailer combinations. Journal of Terramechanics 16 (1970) 1, s. 1-22.

6. Habarta, F., Determination in relation to safety of operation of the minimal load on the front steering axle of a tractor with implements attached. Journal of Agricultural Engineering Research, 16 (1971) 2, s. 126-140.

7. Halling, J., Introduction to tribology. Wykeham Pu... cations ltd, Basinstoke 1976. 157 s.

8. ISO 500 Agricultural tractors - Power take-off and drawbar - specifications. International Organization for Standardization, 1976.

9. Kozin, F., Bogandoff, J.L., Cote, L.J., Introduction to a statistical theory of land locomotions - III. Vehicle dynamics. Journal of Terramechanics 3 (1966) 3, s. 69-81.

10. Kupr, J., Silové účinky v závěsném oku návesů. Zémédélska technika 22 (1976) 11. s. 675-686.

11. Kutzbach, Heinz, Sy Can Nguyen, Dynamische Belastung von Schlepper Anhängerkupplung durch Einachs- bzw. Doppelachsanhänger hoher Nutzmasse. Grundlager der landtechnik 28 (1978) 6, s. 209-248.

12. Lantbruksvagnar. Lantmännen, 1980: 11-12, s. 19-23.

13. Larsson, R., Lantbrukets transporter. Lantmännen 1979: 15, s. 21-38.

14. Mertins, K-H., Ulrich, A., Zur Frage der Lenksicherheit von Ackerschleppern mit Anbaugeräten bei Strassenfahrt. Grundlagen der Landtechnik, 28 (1978) 3, s. 99-107.

15. Pershing, R.L., Yoerger, R.R., Simulation on tractors for transient response. Transaction of the ASAE 1969 s. 715-719.

16. SFS 4532 Traktorit ja maatalouskoneet. Vetokoukku ja vetosilmukka. Suomen Standardisoimisliitto, 1980. 4 s.

17. Stieglitz, E., Der Traktor beim Strassentransport. Deutsche Agrartechnik 21 (1971) 2, s. 59-63.

18. Timoschenko, Strength of materials.

19. Universal joint and driveshaft design manual. Publ. The Society of Automotive Engineerings, Inc. Warrendale 1979, 440 s.

Helsinki 1981. Government Printing Centre