Evaluation of different concepts for heavy tow-trucks

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1. Introduction

Tow-trucks with underlifts are used to recover and tow vehicles that are unable to drive independently due to an accident or a breakdown. Heavy tow-trucks are needed to tow broken down trucks and buses of different kinds and sizes. If the condition of the vehicle to be towed permits, it is picked up by its front or rear axle using the underlift of the tow-truck and is then towed away, fig. 1. The carrier which holds the vehicle, can be rotated around a vertical axis, the so called pivot pin, thereby allowing a steering motion between the two vehicles.

Fig. 1: towing process[1]

Technical evolution as well as stricter safety standards, such as an underride protection on the front and sides of a truck impose increased demands on both the underlift and the vehicle itself.

The standard carrier for heavy vehicles, the so called lift fork is available in combination with three fundamentally different crane systems mounted on one of two different types of trucks. Aside from the lifting motion, all three types of cranes also allow an adjustment of the carrier's angle of inclination as well as the extension of the underlift's arm.

The purpose of this study is to compare these three different systems regarding their technical principles and to evaluate and compare four heavy tow-trucks of the leading manufacturers regarding their load capacities.

[1] www.agefa-technik.de

2. Analysis of the system "tow-truck"

2.1. Mode of operation

In order to tow a vehicle, it is picked up on its front or rear end using an underlift. If permitted by the technical condition of the vehicle, it is fastened to the tow truck's crane using lift forks, see fig.2.

Fig. 2: underlift with mounted lift fork[1]

Depending on the circumstances of the recovery, the forks (1) are directed under the front or rear end of the vehicle so that the crossbeam (2) touches the tires. This can be done by moving the whole truck as well as using the extension mechanism (3) of the crane. The forks are then locked with the rear crossbars (4). When fastened, the towed vehicle is pulled towards the tow truck by retracting the crane's telescopic arm. The remaining minimal distance between the two trucks is determined by the overhang of the towed vehicle and the required space for the steering motion of the lift fork.

The axle located in the forks is then lifted off the ground using either the lifting or the pivoting mechanism of the crane. It is possible to use the pivoting mechanism, because in all three systems, its center of rotation (5) is located near the rear end of the tow truck. However, the main purpose of the pivoting mechanism is to adjust the lift fork's angle of inclination in order to avoid damage on the towed vehicle while it is lifted off the ground. The lift fork and the crane are connected by a joint (6) allowing a steering motion between the two vehicles. In order to provide proper function of the pivot pin and to minimize its wear, the lift fork should be kept horizontal during towing.

2.2. Legal Regulations

According to guideline 14 concerning § 18 Abs. 2 Nr. 1 of the German STVZO[2], tow trucks are recognized as self propelling work machines and therefore they can be exempted from the general regulations for the registration of vehicles[3]. Exceptions from these regulations can be made by the authorities in charge based on § 70 STVZO. In general, the regulations for the vehicle’s dimensions stay in effect, imposing the following requirements:

- The maximum width of the vehicle is not to exceed 2550 mm
- The maximum length of the vehicle is not to exceed 12000 mm
- The maximum height of the vehicle is not to exceed 4000 mm

If the vehicle is recognized as a work machine according to key no 1627 DA 53 or 1601 DA 1, its maximum permissible load on the rear axles is usually increased to 2x12 tons.

The STVZO also regulates the maximum permissible towing speed (equation 1) which is calculated based on the load distribution between the front and rear axles. It is necessary to keep a certain percentage of the total weight on the front axle to ensure the tractability of the tow-truck.

$$v_{zul} = \frac{160}{{A_h}/{A_v} - 2} km/h \tag{1}$$

v_{zul}: permissible maximum towing speed

A_h: weight on rear axle while towing

A_v: weight on front axle while towing

Furthermore, neither the maximum total weight, nor the maximum loads on each individual axle are to be exceeded in any mode of operation.

For the recognition as a towing company with a class I / B1 vehicle by the “Verband der Bergungs- und Abschleppunternehmen” a vehicle with a towing capacity of 6 tons at a speed of 80 km/h is required.

[2] Richtlinie für die Begutachtung von Abschleppwagen (Kranwagen) als Arbeitsmaschinen. BMV/StV 7-8033 B/67, Juni 1963, VkBl 1967, S. 394

[3] StVZO §18 Abs. 2

3. Comparison of different underlift principles

3.1. Requirements on the underlift

The design of an underlift can be evaluated under the following technical aspects:

Kinematics:

- For an easy and precise positioning of the underlift, the individual motions should not influence each other.
- With the exception of the extension and retraction of the telescopic arm, no motion should alter the distance between the rear axle of the tow truck and the lift fork. That way, restrictions of the steering motion between the two vehicles or an overload on the crane can be avoided.

Actuation:

- The actuation forces should be kept low in order to reduce the stress on the crane elements and to minimize the weight and wear of the relevant parts.
- The handling of the underlift should be precise. The higher the ratio between the stroke of the hydraulic cylinder and the resulting traverse path of the crane, the more accurately it can be positioned.

Rigidity and stiffness:

- To withstand the strain, the crane elements can be designed appropriately and dimensioned large enough or high-strength materials can be used.
- The elastic deformation of the crane occurring under the load should be kept relatively small to avoid excessive bouncing and swinging.

Weight:

- In order to keep the necessary hydraulic forces low and the load capacity of the tow truck high, the weight of the underlift should be as low as possible.

The following are the most important criteria for the evaluation of a crane principle:

1. The interference between the individual motions
2. The change of distance between the two vehicles caused by the crane motions
3. The ratio between the stroke of the hydraulic cylinder and the resulting traverse path of the crane

4. The ratio between hydraulic force and load

3.2. Evaluation of existing systems

Underlifts for tow trucks are built as cranes with a pivoting or a linear lifting motion. In cranes with a pivoting lifting motion, the joint for the mechanism to change the lift forks angle of inclination can be located up high between the horizontal and the vertical crane beam, or down low between the vertical beam and the telescopic arm. Fig. 3 shows the different crane types and the traverse paths resulting from the lifting motion (I) and the inclination of the lift fork (II).

Interference of the individual motions and alteration of the load distance

The amount by which the load distance is altered through the pivoting lifting motion depends on the position of the motion's center of rotation. The higher above the lift fork this center of rotation is located, the more the load distance is increased in the lower range of motion. A position of the center of rotation below the lift fork would result in a decrease of the load distance as the load is raised. These effects become stronger as the center of rotation is moved horizontally towards the lift fork.

The traverse paths of fig. 3 clearly show, that under the assumed geometrical conditions, the change of distance is very small compared to the total load distance. The change of the lift fork's inclination is also relatively small for a reasonable vertical motion.

The linear crane causes no alteration of the load distance or the inclination angle.

The change of the lift fork's inclination however, has a strong influence on the lifting height and a moderate influence on the load distance. The amount of alteration again depends on the position of the motions center of rotation. The further above or below the lift fork it is located, the higher the influence on the load distance becomes. A reduction of the inclination angle causes a decrease in the load distance if the pivoting motion's center is located above the lift fork and an increase in the load distance if it is located underneath. Increasing the angle of inclination has the opposite effect.

For the range of motion during regular use, however, a position in which the center of rotation is below the lift fork only occurs in model 1a.

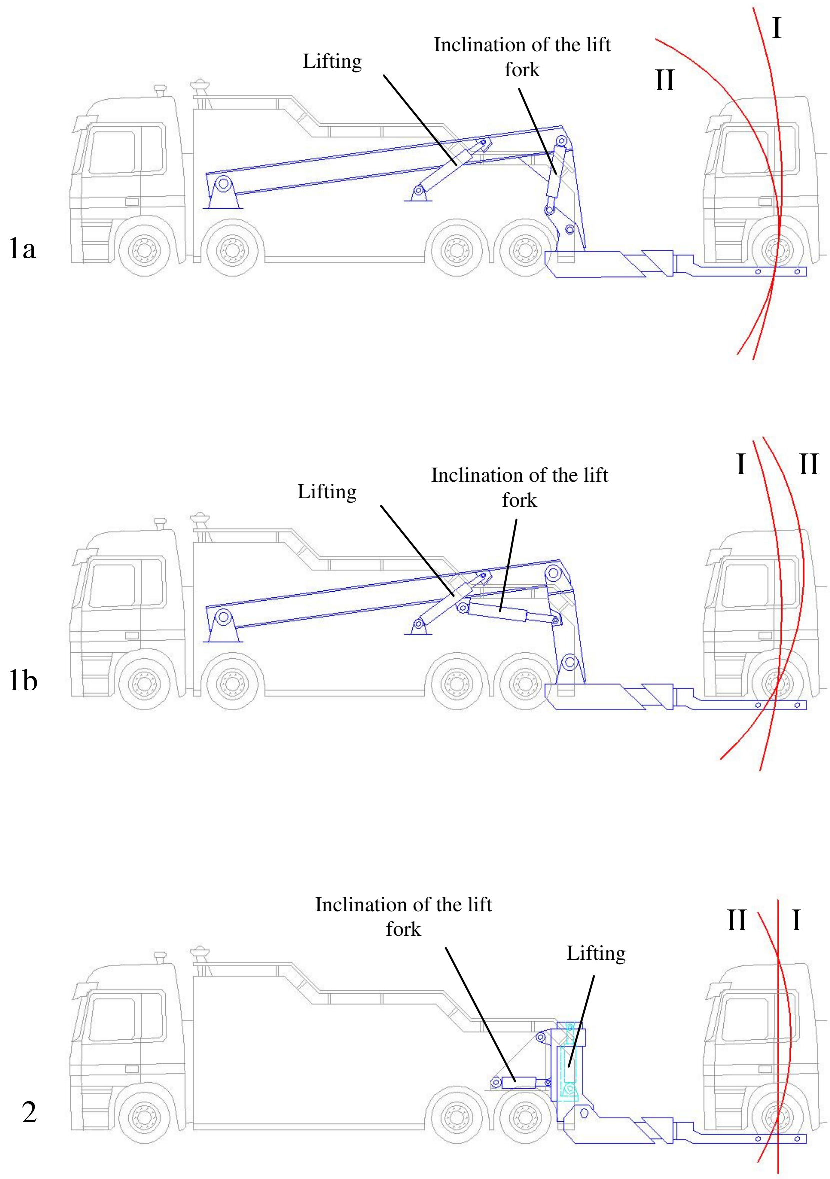

Fig. 3: Different crane concepts and their kinematics

<u>Handling precision</u>

The ratio between the stroke of the hydraulic cylinder and the resulting traverse path of the crane can be used as a measurement for the precision with which the crane can be positioned. However, this ratio can only be used to evaluate the kinematics, because the actual precision of handling also depends on other system parts such as the hydraulic pump, the valves, or the operating controls. Furthermore it has to be mentioned that the evaluation can only be done qualitatively, because all measurements are estimates.

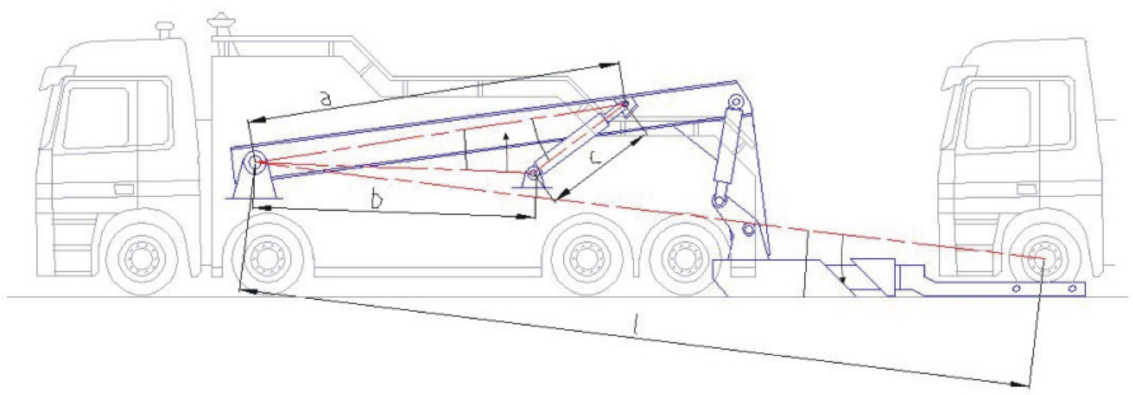

Fig. 4: Geometry of a pivoting underlift

According to fig. 4, the length c of the hydraulic cylinder can be calculated depending on the angle a using the following equation:

$$c = \sqrt{a^2 + b^2 - 2a \cdot b \cdot \cos\alpha} \tag{2}$$

The lifting height h can be determined as follows:

$$h = l \cdot (\sin\beta_0 - \sin\beta) \tag{3}$$

The correlation between the angles a and b is expressed in equation 4:

$$\beta = \beta_0 + \alpha_0 - \alpha \tag{4}$$

Using equations 2-4, the length c of the hydraulic cylinder can be expressed as a function of the lifting height h:

$$c = \sqrt{a^2 + b^2 - 2a \cdot b \cdot \cos(\beta_0 + \alpha_0) \cdot \sqrt{1 - \left(\sin\beta_0 - \frac{h}{l}\right)^2} - 2a \cdot b \cdot \sin(\beta_0 + \alpha_0) \cdot \left(\sin\beta_0 - \frac{h}{l}\right)} \tag{5}$$

The ratio between the stroke of the hydraulic cylinder and the resulting traverse path of the crane – or the achievable handling precision – is the derivative of equation 5.

$$\frac{\partial c}{\partial h} = \frac{1}{2}\left(a^2 + b^2 - 2a \cdot b \cdot \cos(\beta_0 + \alpha_0) \cdot \sqrt{1 - \left(\sin\beta_0 - \frac{h}{l}\right)^2} - 2a \cdot b \cdot \sin(\beta_0 + \alpha_0) \cdot \left(\sin\beta_0 - \frac{h}{l}\right)\right)^{-\frac{1}{2}}$$
$$\cdot \left[- a \cdot b \cdot \cos(\beta_0 + \alpha_0) \cdot \left(1 - \left(\sin\beta_0 - \frac{h}{l}\right)^2\right)^{-\frac{1}{2}} \cdot \frac{2}{l} \cdot \left(\sin\beta_0 - \frac{h}{l}\right) + 2 \cdot \frac{a \cdot b}{l} \cdot \sin(\beta_0 + \alpha_0)\right] \tag{6}$$

For the crane with a linear lifting motion, the changes in length in the hydraulic cylinder and in the lifting height are equal.

The curves for the handling precision that were derived from the dimensions in fig. 4 are shown in fig. 5. It clearly shows that a linear crane allows for much more precise adjustment of the lifting height.

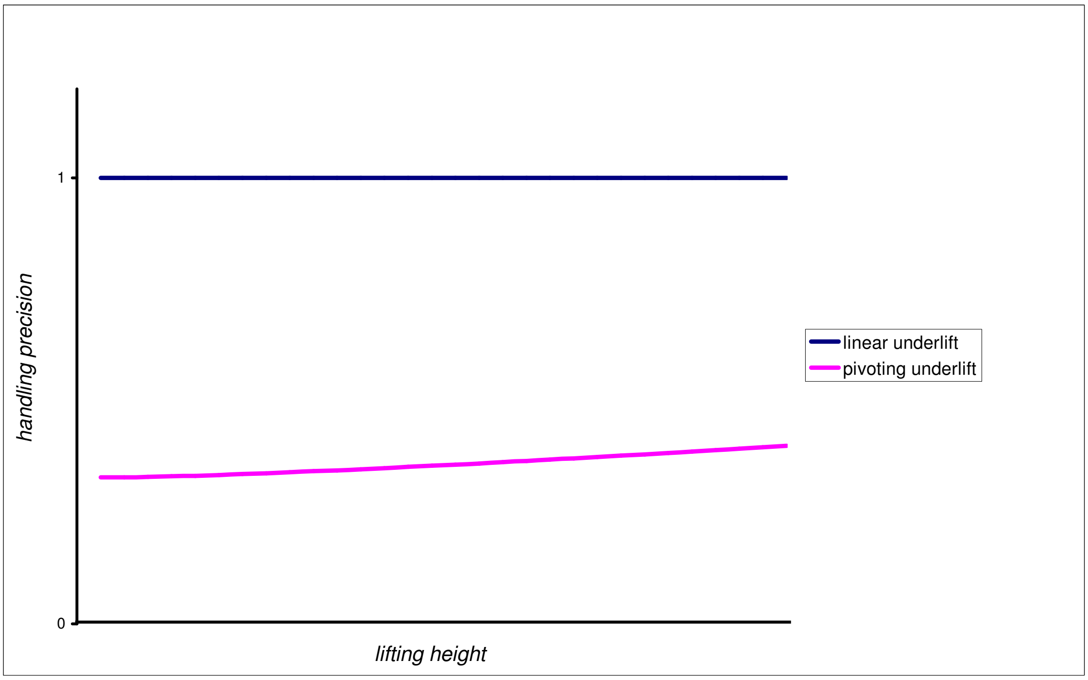

Fig. 5: Handling precision

The mechanisms to adjust the inclination angle of the lift fork are very similar as far as their kinematics are concerned. Therefore a qualitative comparison would not produce a reliable result, even though it can be expected, that model 1a will have the lowest achievable handling precision due to the least favorable leverage.

Necessary hydraulic force

In order to reduce weight and wear on the underlift, the hydraulic forces necessary to move the load should be kept as low as possible.

According to the first law of thermodynamics, the following expression can be derived from fig. 4:

$$W_{hydr} = W_{Last} \Rightarrow F_Z \cdot \partial c = F_L \cdot \partial h \Leftrightarrow \frac{F_Z}{F_L} = \left(\frac{\partial c}{\partial h}\right)^{-1} \qquad (7)$$

In the linear crane, the ratio of forces is 1. For the pivoting crane, this ratio can be calculated using equation 6. The curves resulting from these calculations show that the pivoting crane requires much higher hydraulic forces than the linear crane to lift the same weight (see fig. 6).

The similarity of the different pivoting mechanisms for the lift fork makes it impossible to compare them without exact technical data.

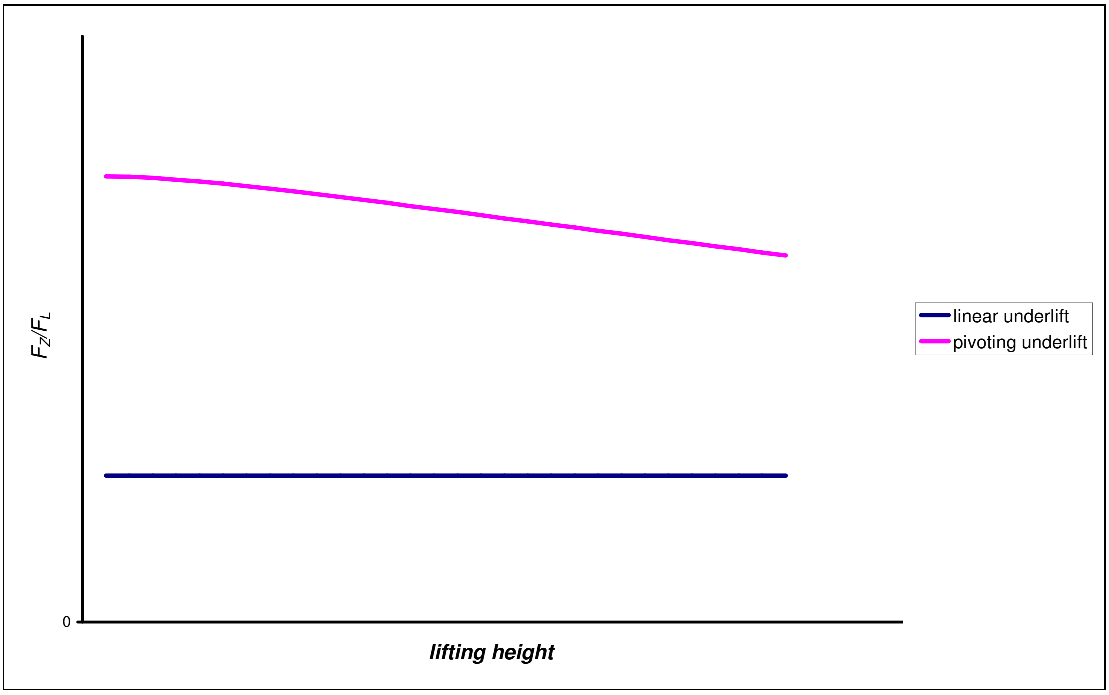

Fig. 6: Ratio of forces over lifting height

In conclusion it can be stated that the linear crane is superior to the pivoting crane as far as achievable handling precision and necessary hydraulic forces are concerned. On the other hand, it requires a longer hydraulic cylinder and a linear bearing for the lifting motion. The interference between the individual motions of the cranes are similar in all three models and do not show any significant advantages for a specific crane type.

4. Load capacity

4.1. Requirements on the tow-truck

Towing a vehicle does not only increase the total weight of the tow-truck, it also shifts the center of gravity towards the rear axle. This leads to a disproportionately high increase in the load on the rear axle while simultaneously decreasing the load on the front axle. Neither the maximum total weight, nor the maximum permissible load on the rear axle is to be exceeded through this process. In addition, the ratio between the loads on the rear and front axle has to be observed since it is the limiting factor for the towing speed. A ratio of 4:1 or less results in a maximum towing speed of 80 km/h.

On the other hand, the maximum permissible load on the front axle is not to be exceeded while the crane is load-free. A balanced distribution of the weight between the axles should be aimed at since it contributes to the vehicle handling.

4.2. Calculation of the load capacity

The geometry and the distribution of forces on the truck while towing are depicted in fig. 7.

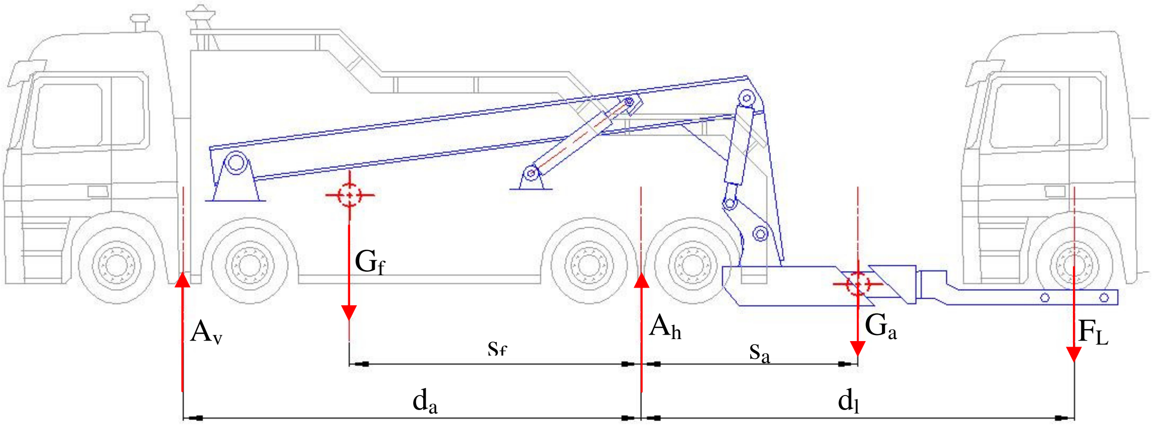

Fig. 7: Geometry and distribution of forces during towing

G_f:	*weight of the vehicle*	A_h:	*load on rear axle*
G_a:	*weight of the telesdopic arm*	F_L:	*towing load*
A_v:	*load on front axle*		

The conditions on the load-free truck are shown in fig. 8.

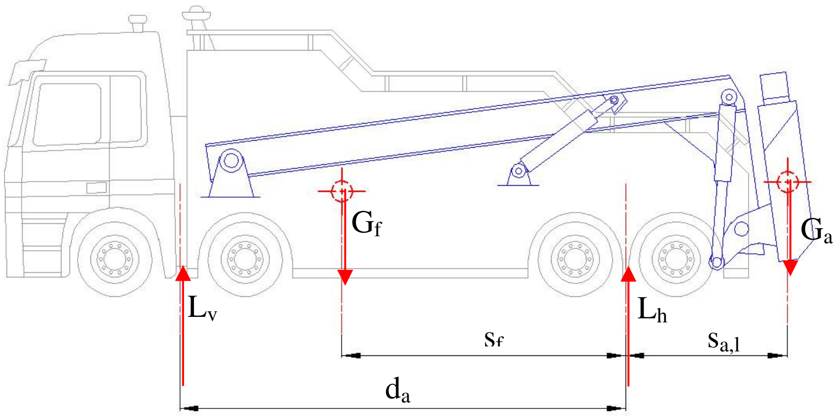

Fig. 8: Geometry and forces on the load-free truck

The German TÜV uses a simplified method to calculate the load capacity and neglects the shift of the telescopic arm's center of gravity. The load capacity can be deducted from the equilibrium conditions (equations 8 and 9) as a function of the load distance and the ratio between the load on the rear and front axle (equation 10).

A comparison of the moment around the rear axle in the two loading cases leads to the following equation:

$$(L_v - A_v) \cdot d_a = F_L \cdot d_l \tag{8}$$

The sum of the axle loads while towing is increased by the weight resting on the lift fork:

$$L_h + L_v + F_L = A_h + A_v = A_v \cdot \left({}^{A_h}\!/_{A_v} + 1 \right) \tag{9}$$

A combination of these two equations leads to the formula that is used by the TÜV to calculate the towing capacity.

$$\begin{aligned} & L_h + L_v + F_L = \left(L_v - \frac{F_L \cdot d_l}{d_a} \right) \cdot \left({}^{A_h}\!/_{A_v} + 1 \right) \Leftrightarrow L_h + F_L \cdot \left[1 + \frac{d_l}{d_a} \cdot \left({}^{A_h}\!/_{A_v} + 1 \right) \right] = L_v \cdot {}^{A_h}\!/_{A_v} \\ & \Leftrightarrow F_L = d_a \cdot \frac{{}^{A_h}\!/_{A_v} \cdot L_v - L_h}{d_a + d_l \cdot \left({}^{A_h}\!/_{A_v} + 1 \right)} \end{aligned} \tag{10}$$

For a tow-truck with a movable counterweight (see fig.9), the calculation has to be slightly adapted. The weights Lv and Lh on the axles of the load-free truck that are needed for the calculation are the theoretical values that would be obtained if the counterweight was moved in the towing position but without any load on the crane. That way, the theoretical load on the front axle can exceed the maximum permissible load giving the tow-truck extra tractability. Another advantage is that the counterweight creates a well-balanced weight distribution and thereby good vehicle handling when it is moved to the back of the truck.

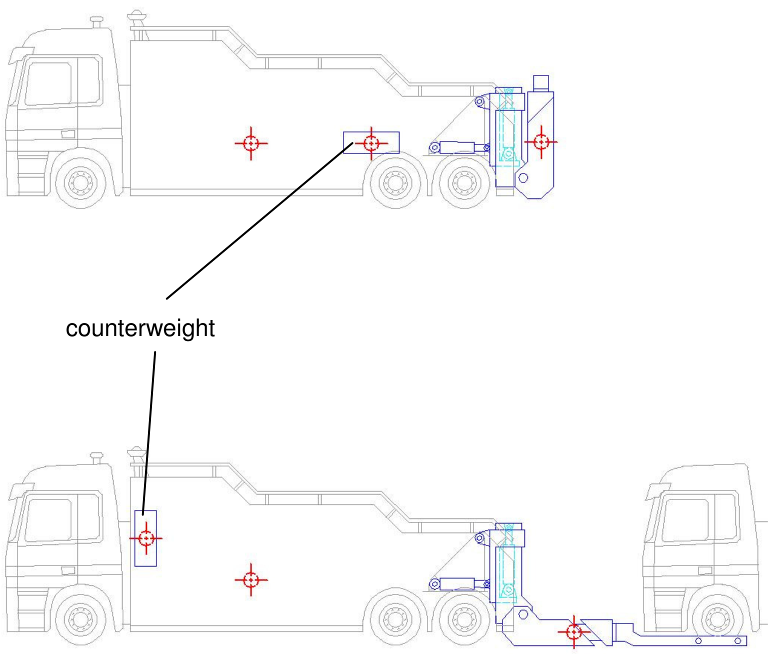

Fig. 9: tow-truck with movable counterweight

The load capacity is not only limited by the unloading of the front axle, but also by the maximum permissible load on the rear axles. The calculation is based on the equilibrium of moments around the rear axle according to fig. 7:

$$G_{ges} \cdot s_f = A_v \cdot d_a + F_L \cdot d_l \tag{11}$$

The distance between the center of gravity and the rear axle can be calculated as follows:

$$s_f = \frac{L_v \cdot d_a}{L_h + L_v} \tag{12}$$

The load on the front axle can be expressed according to equation 13:

$$A_v = G_{ges} + F_L - A_h \tag{13}$$

Combining equations 11-13, the maximum load can be expressed as a function of the maximum permissible weight on the rear axle and the load distance:

$$F_{L,\max} = d_a \cdot \frac{L_v - G_{ges} + A_{h,\max}}{d_l + d_a} \tag{14}$$

4.3. Comparison of different heavy tow-trucks

In the following chapter, 4 different heavy tow-trucks are evaluated and compared regarding their load capacities. Three of these systems are based on platforms with 4 axles, have a stationary counterweight and pivoting cranes, one of them is based on a platform with three axles, has a movable counterweight and a linear underlift. The relevant technical data is listed below.

MAN TGA 35.480 8x4 BB; by E

Wheel base[4] d_a:	5102.5 mm
Max. permissible total weight:	35000 kg[5]
Unloaded Weight:	27000 kg
Max. permissible weight on front axle:	2 x 8000 kg
Unloaded weight on front axle:	13000 kg
Max. permissible weight on rear axle:	2 x 12000 kg[5]
Unloaded weight on rear axle:	14000 kg
Weight distribution rear axle : front axle:	52:48

MAN TGA 41.460 8x4 BB; by N

wheel base d_a:	5102.5 mm
Max. permissible total weight:	35000 kg
Unloaded Weight t:	23600 kg
Max. permissible weight on front axle:	2 x 8000 kg
Unloaded weight on front axle:	12400 kg
Max. permissible weight on rear axle:	2 x 12000 kg[5]
Unloaded weight on rear axle:	11200 kg
Weight distribution rear axle : front axle:	47.5 : 52.5

[4] The wheel base of a 4 axle vehicle is the distance between the suspensions (= midpoints) of the front and rear double axles, in a 3 axle vehicle it is he distance between the front axle and the suspension of the rear double axle, which results in a longer wheelbase at the same total length of the vehicle.

[5] Limited by legal regulations rather than technical possibility

MAN TGA 41.460 8x4 BB; by B

Wheel base d_a:	5702.5 mm
Max. permissible total weight:	35000 kg
Unloaded Weight:	27400 kg
Max. permissible weight on front axle:	2 x 7500 kg
Unloaded weight on front axle:	13600 kg
Max. permissible weight on rear axle:	2 x 12000 kg[5]
Unloaded weight on rear axle:	13800 kg
Weight distribution rear axle : front axle:	50 : 50

Mercedes Benz ACTROS 3346 6x4; Aufbau: AGEFA

Wheel base d_a:	6075 mm
Max. permissible total weight:	33000 kg
Unloaded Weight:	21840
Max. permissible weight on front axle:	9000 kg
Unloaded weight on front axle:	9000 kg
Unloaded weight on front axle, theor.:	11480 kg
Max. permissible weight on rear axle:	2 x 12000 kg
Unloaded weight on rear axle:	12840 kg
Unloaded weight on rear axle, theor.:	10520 kg
Weight distribution rear axle : front axle:	58 : 42
Weight distribution rear : front, theor.:	48 : 52

The load capacities resulting from the restrictions on the tractability and the maximum load on the rear axle are shown in fig. 10.

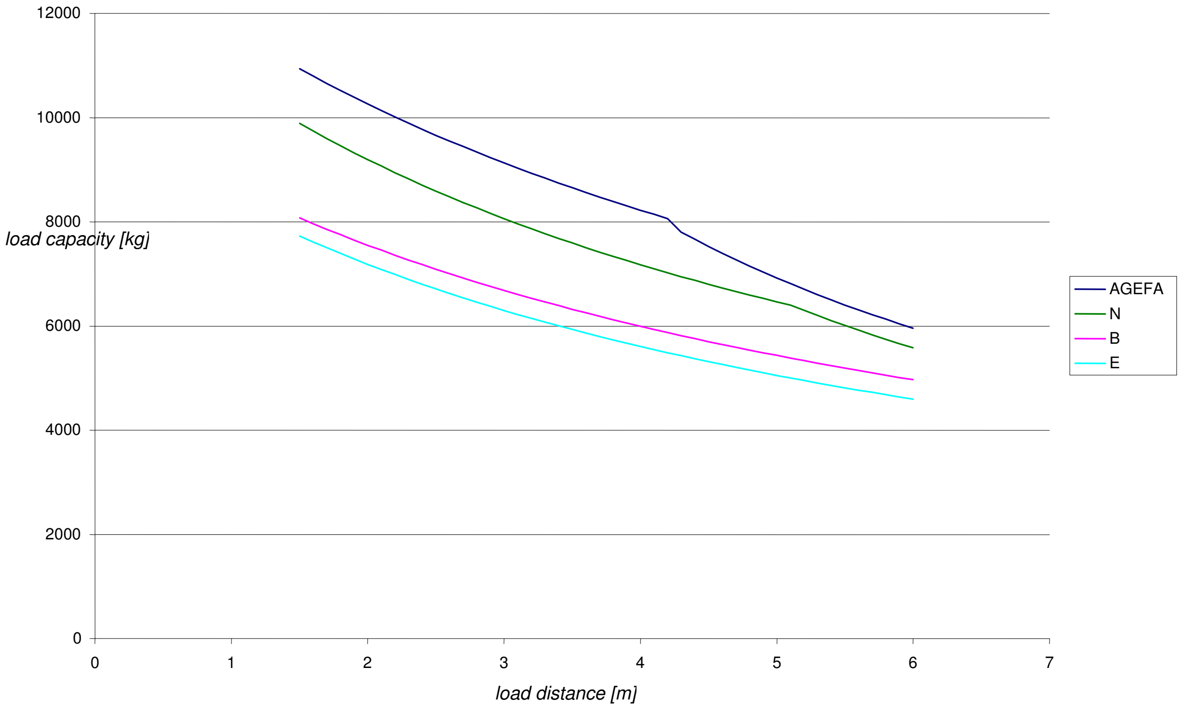

Fig. 10: Load capacities of the tow-trucks

It shows that the lighter 3-axle vehicle by AGEFA has a load capacity which averages between 1 and 3 tons above those of its competitors with 4-axle platforms. The reason for the higher load capacity is the movable counterweight of 3000 kg which gives the tow-truck a better weight distribution both when towing and when load-free.

Another important aspect in the evaluation of the tow trucks is the difference in the maximum load capacity limited by the tractability compared to the maximum load capacity limited by the permissible weight on the rear axle. The corresponding curves are shown in fig. 11. As in fig. 10, the tractability is calculated for a maximum towing speed of 80 km/h.

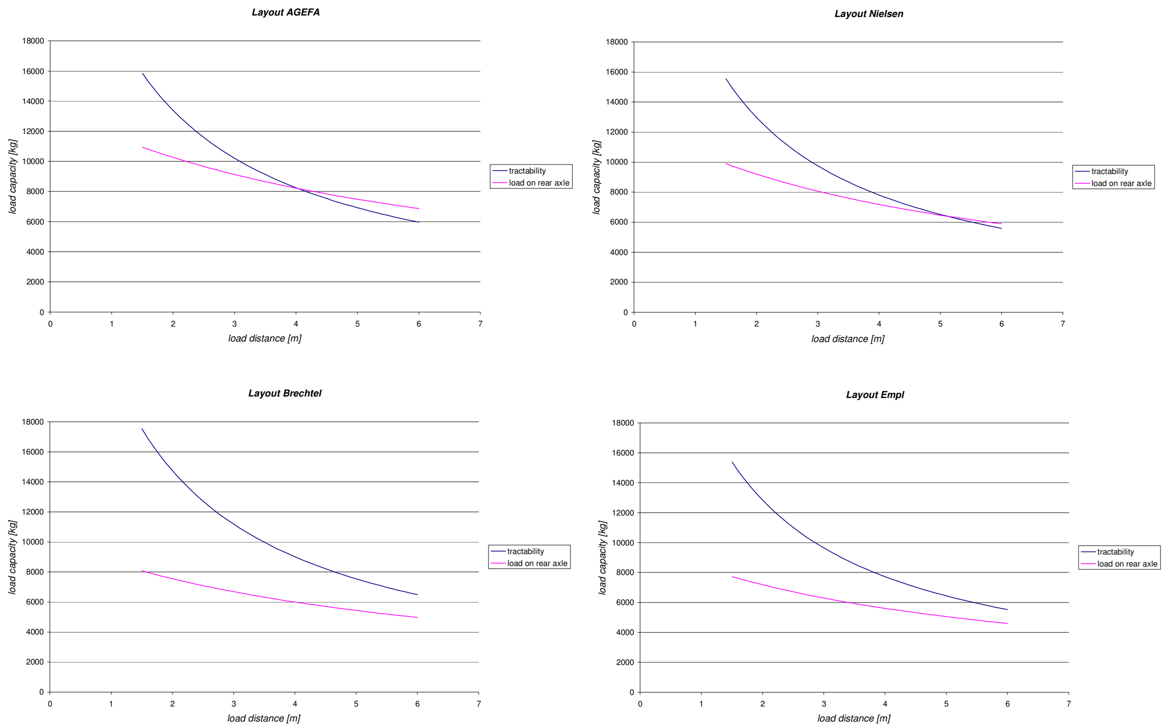

Fig. 11: layout of the tow-trucks

The graphs show that the models E and especially B are limited in their load capacity by the maximum permissible weight on the rear axle long before the limit of the tractability is reached. Their high overall weight causes good tractability on the one hand, but also leads to a rather low load capacity, because too much of this weight rests on the rear axle.

The N tow-truck shows a more balanced layout design, the two curves are much closer together. At a load distance of more than 5 m, the load capacity is even limited by the tractability rather than the maximum weight on the rear axle.

The truck by AGEFA shows the best balanced layout, the curves are very close to each other. At a load distance of more than 4 m, the tractability becomes the limiting factor for the load capacity. In addition to having the highest towing capacity, a reduction of the towing speed would lead to a further increase in the permissible weight to be towed, so that a weight of 7 tons could be towed at a load distance of 6m.

5. Conclusion

Heavy tow trucks do not only have to be equipped with an underlift that allows for easy and precise positioning, they also need to have a high load capacity and good vehicle handling when they are load-free.

Different concepts for heavy tow-trucks were evaluated based on their crane principles as well as their load capacities and layout designs:

- A tow-truck with four axles, equipped with a static counterweight and a conventional pivoting crane with an inclination mechanism for the lift fork between the vertical crane beam and the telescopic arm.
- A tow-truck with four axles, equipped with a static counterweight and a conventional pivoting crane with and an inclination mechanism for the fork lift between the horizontal and the vertical crane beam.
- A tow-truck with three axles, equipped with a movable counterweight and a linear underlift.

The comparison of the different crane systems showed that the linear crane requires lower hydraulic lifting forces and can be positioned more precisely. Regarding the kinematics and the pivoting mechanism, no significant advantages or disadvantages for one of the three systems could be found.

Four tow-trucks of different manufacturers were compared regarding their load capacities. Again, the lighter vehicle with three axles and the linear crane showed the best performance. The crucial factor for this result is the truck’s movable counterweight of 3000 kg which results in the following advantages:

- A better balanced weight distribution between the axles when the truck is load free and therefore better vehicle handling
- A better layout design as far as the limitation of the load capacity by tractability and maximum weight on the rear axle is concerned. This results in a higher overall load capacity
- A lower unloaded weight
- The crane can be mounted on a truck with three axles which reduces both acquisition costs and maintenance expenses.