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KINEMATICS AND COMPLIANCE ANALYSIS OF A 3.5 TONNE LOAD CAPACITY INDEPENDENT FRONT SUSPENSION FOR LCV

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ABSTRACT

This paper deals with the development of a 3.5 tonne carrying double wishbone front suspension for a low floor LCV. It is a novelty in this class of vehicles. It has a track width of 1810 mm and it has a recirculating ball steering system. The steering mechanism has been arranged so that the steering angle could reach to 48° that is a very effective angle in that vehicle range. This results as a lower turning radius which indicates a better handling for the vehicle.

The steering and the front suspension system here have been optimized in terms of comfort and handling by using DOE (design of experiments) based on sequential programming technique. In order to achieve better suspension and steering system geometry, this technique has been applied. The results have been compared with the benchmark vehicle.

Keywords: Light commercial vehicles, independent suspension, steering, vehicle dynamics.

OBJECTIVE

In this study, the development process of the front suspension system for CV7(Commercial Vehicle 7 tonnes) vehicle has been mentioned. The front suspension of CV7 is considered as a double wishbone type suspension system. For those vehicles which have above than 3.5 tonnes axle weight, double wishbone type of suspension is much more suitable than McPherson type of suspension. On the other hand, it is more convenient to adjust camber change for the double wishbone type of suspension. Above 3 tonnes, it is not suitable to use rack and pinion steering system. Thus, a recirculating ball steering system has been adapted. In this study, three different vehicles have been compared to each other as being CV5(Commercial Vehicle 5 tonnes), CV7, CV10 (Commerical Vehicle 10 tonnes).

ANALYSIS METHOD

In order to satisfy project's requirements; first of all, a suspension system has been modeled in Adams/View and programmed parametrically for the adaption of the components most of suspension parameters. Some macros have been programmed in order to do DOE (design of experiments) with the sequential approach.

Suspension requirements;
- carrying 3.5 tonnes,
- satisfying acceptable comfort level
- wheelbase 4400 mm
- track width 1810 mm
- satisfy ground clearance and step height criteria
- allowing 48° steering angle

Steering system requirements;

- inner steer angle should be turn 48°
- supplying adequate steering torque in order to steer 48°,
- Ackermann error should be lower than 3°

During development of the suspension system, advanced suspension system analyses have been done.

There are

- Parallel wheel travel
- Roll & vertical force analysis

Parallel wheel travel analysis shows the K&C properties of the suspension. Some critical suspension setups have been given as below [1-8]:

- Camber Angle Change: Camber change shows the characteristics how wheels support the vehicle in cornering condition. Negative camber in positive wheel travel has positive effects on vehicle when cornering occurs.
- Caster Angle Change: Positive caster creates aligning torque along positive z direction, positive caster creates aligning torque and it straightens the steering wheel, it improves straight line stability and also increases negative camber gain when turning.
- Toe Change: It represents wheels' toe angle change during suspension systems travel along vertical axis. This should be kept minimum, in order to avoid

bump steer effect and it also protects against tire wear.
- Ride Frequency: Ride frequency shows comfort level of the vehicle. Lower ride frequency indicates better comfort of the vehicle. This should be kept below 1.5 Hz.

Roll &vertical force analysis shows K&C (kinematics and compliance) properties of the suspension system in roll action. Some critical suspension results have been shown:
- Roll steer: It represents steer angle change with respect to roll angle.
- Roll center vertical distance: Roll center vertical distance shows that vehicles' moment point during cornering events. When cornering occurs, lateral force acting on CoG (center of gravity) of the vehicle creates moment along the distance between CoG and roll center, thus the vehicle tends to roll. Closer moment arm, better roll stability.

RESULTS

Parallel Wheel Travel Analysis

Table 1: *Axle Load, Ride Frequency, Spring Displacement table with respect to suspension travel along vertical axis.*

	Wheel Travel [mm]	Load [kg]	Ride Frequency [Hz]	Spring Displacement [mm]
Full Bump	80	12232	1.3065	141.2
2.5g Bump	73.469	8000	1.5788	145.26
Overload	15.7	3500	1.2983	188.32
GVW	5.1	3200	1.3528	196.21
Design	0	3053.3	1.3806	200
Bumpstop (@Contact)	-26.4	2323.5	1.3328	219.46
Kerb	-40.4	2075	1.406	229.73
Full Rebound	-110	884.17	2.2	280.07

Table 1 summarizes the axle load capacity (kg), wheel travel (mm), ride frequency (Hz) and spring displacement (mm) for the loading conditions of full bump, 2.5g bump, overload, GVW (gross vehicle weight), design, bumpstop at contact, kerb and full rebound consecutively.

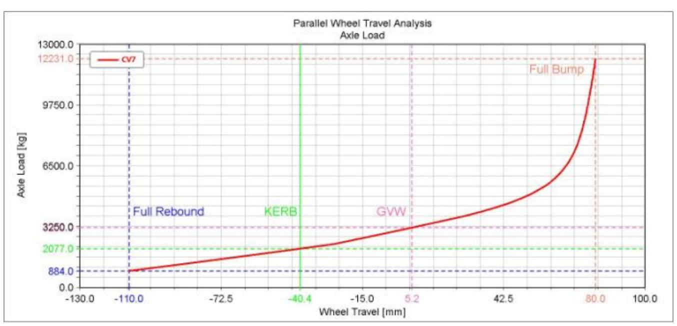

Figure 1: *Axle load with respect to suspension travel along vertical axis.*

Figure 1 shows the front suspension travel and axle load in different loading condition. Kerb and GVW position is adjusted in order to have better ride comfort level, ground clearence and step height conditions.

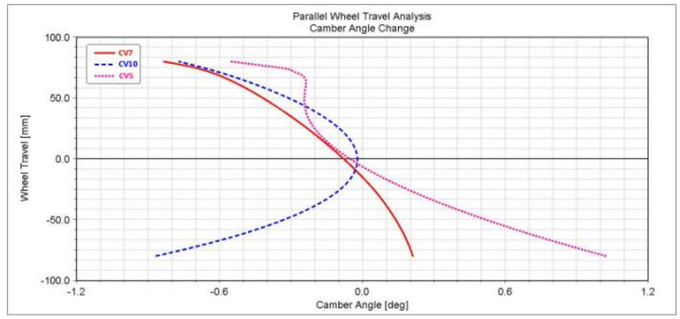

Figure 2: *Camber angle change comparison with respect to suspension travel along vertical axis.*

Camber angle ideally is adjusted by double wishbones upper and lower control arms according to dynamics criteria of the vehicle. Figure 2 illustrates the camber angle change comparison with respect to suspension travel along vertical axis within the benchmark vehicles.

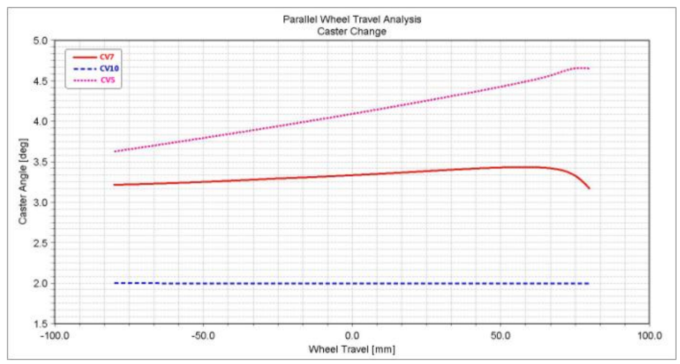

Figure 3: *Caster angle change comparison with respect to suspension travel along vertical axis.*

Caster angle is ideally set also taking into account steering wheel moment demands and also straight and cornering stability of the vehicle. Figure 3 shows the caster angle change comparison with respect to suspension travel along vertical axis within the benchmark vehicles.

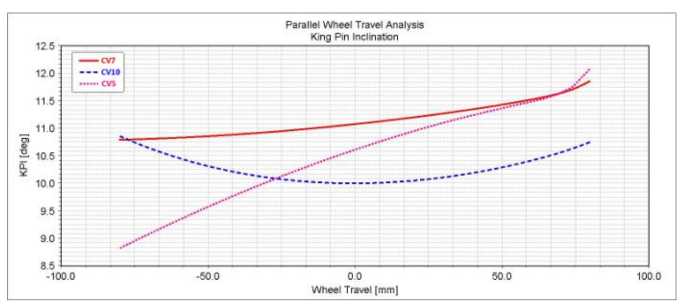

Figure 4: *King Pin inclination with respect to suspension travel along vertical axis.*

Figure 4 gives the king pin inclination with respect to suspension travel along vertical axis within the benchmark vehicles.

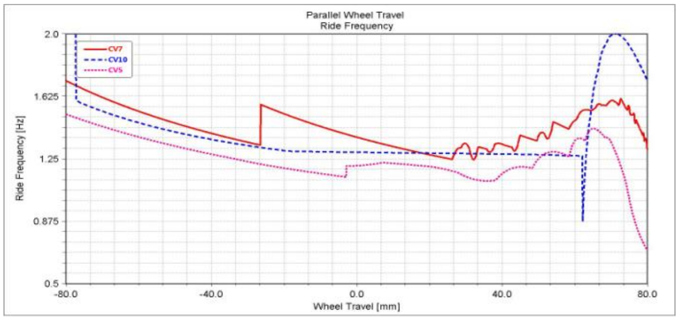

Figure 5: *Ride Frequency with respect to suspension travel along vertical axis.*

For comfortable driving conditions, kerb and GVW ride frequency has been kept below 1.5 Hz, shown in Figure 5.

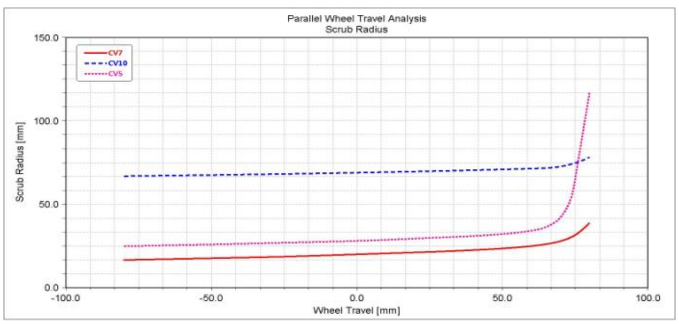

Figure 6: *Scrub Radius with respect to suspension travel along vertical axis.*

Figure 6 shows the scrub radius with respect to suspension travel along vertical axis within the benchmark vehicles.

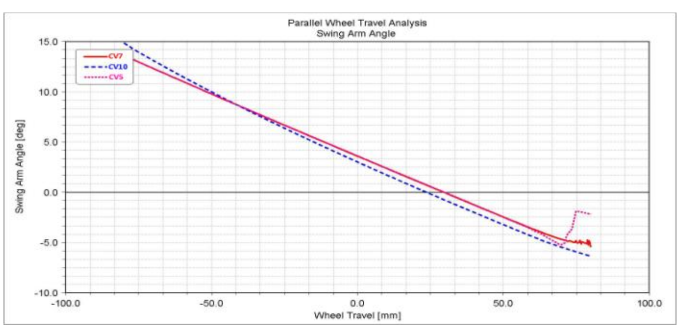

Figure 7: *Swing Arm Angle with respect to suspension travel along vertical axis.*

Swing arm angle should be in the range of (-8°) - (+ 30°). In all vehicles, these values are in the range, as given in Figure 7.

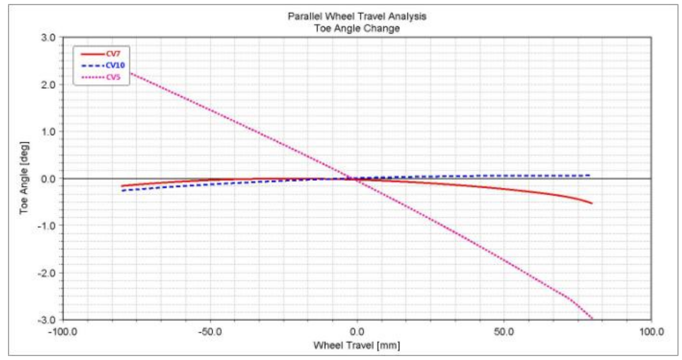

Figure 8: *Toe Angle Change wih respect to suspension travel along vertical axis.*

Toe angle change shows that how the wheels are kept in straight line position during all position of the suspension vertical travel as shown in Figure 8.

<u>Roll Vertical Analysis:</u>

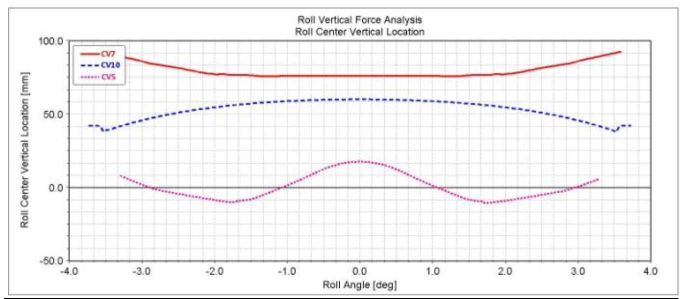

Figure 9: *Roll Center Vertical Location with respect to roll angle of the suspension system.*

Roll center vertical location values are kept in acceptable ranges (lower than 100 mm) as given in Figure 9.

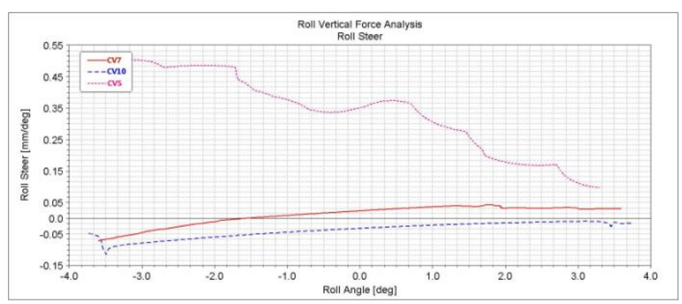

Figure 10: *Roll Steer with respect to rolll angle of the suspension system.*

Roll steer values are kept in acceptable ranges as shown in Figure 10.

Steering Analysis

Table 2: *Kerb to kerb and wall to wall turning radius calculation for CV7.*

Inner	δi (deg)	48
Outer	δo (deg)	37.3193647
Kerb to Kerb	Rkk	7257.645549
Wall to Wall	Rww	8041.978545

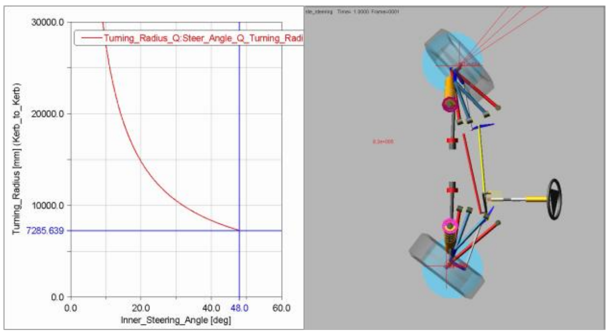

Figure 11: Turning radius Adams simulation.

Kerb to kerb and wall to wall are calculated by hand (Table 2) and simulated Adams/Car software taking into account turning wheel along King Pin axis as indicated in Figure 11.

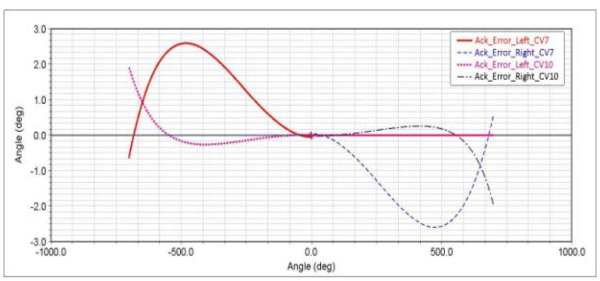

Figure 12*:* *Ackermann error comparison for CV7 and CV10 vehicles.*

Ackermann error should be lower than 3° in order to prevent excessive tire wear[9] (see Figure 12).

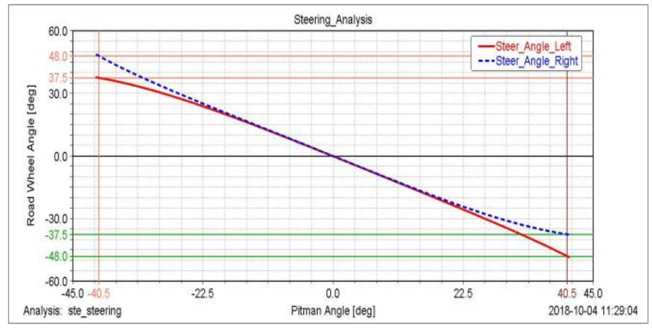

Figure 13: *Inner and outer steering angle with respect to pitman angle.*

Taking into account turning radius, package constraints and movement capabilities of the drive shafts, inner steer angle is limited to 48° (see Figure 13). Steering geometry is set to symmetrical both left steer and right steer (see Figure 14 and Figure 15).

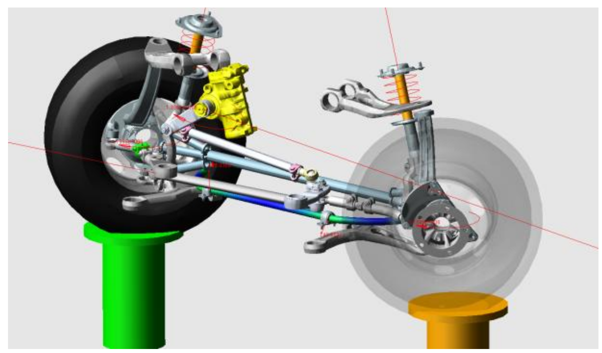

Figure 14: *Right steer position in GVW(Gross Vehicle Weight) condition*

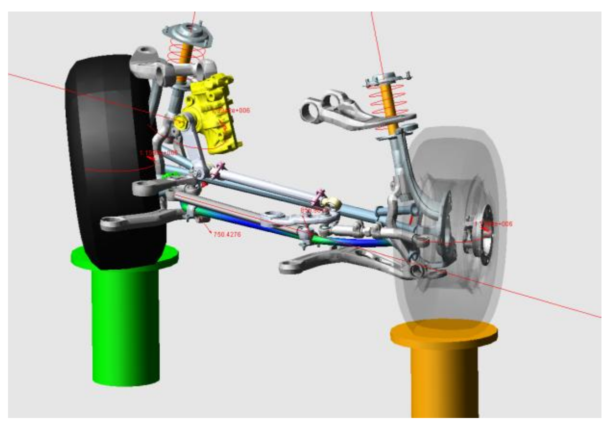

Figure 15: *Left steer position in GVW condition*

In order to achieve turning radius according to given target wheelbase, inner steer angle should be 48° to satisfy

Ackerman Angle. One of the main issue to satisfy such large steer angle is to avoid singularity between tie rod and kinpin axis. When toggle angle starts to increase, singularity may be occured around larger toggle angle. Figure 16 shows CV7 toggle angle when wheel turns 48°. Target value of toggle angle is set below 150° to be kept in safe side for singularity.

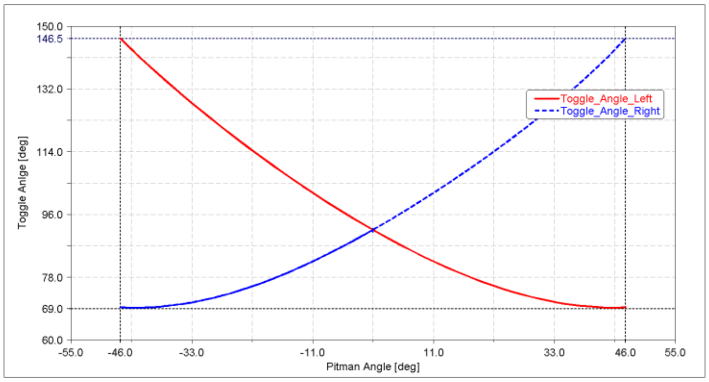

Figure 16: *Toggle angle with respect to pitman angle*

Step Steer Analysis

Total Understeer Gradient:

By definition the total understeer gradient is calculated by the following expression in Eq. 1[10]:

$$U = \frac{d\left(\frac{SWA}{NG}\right)}{d(a_y)} - \frac{d\left(\frac{3.6 * YAW * WB}{V}\right)}{d(a_y)} \qquad (1)$$

where SWA is the steering wheel angle in degrees, ay is the lateral acceleration in g's, NG is the overall steering ratio in degrees/degrees, YAW is the yaw rate in degrees/second, WB is the wheelbase in meters, V is the forward speed of the vehicle in km/h.

The linearized single-track model is valid for tire slip angle ranges up to 2.5°. Therefore, the range of lateral acceleration to reach steady-state 2.5 degrees of side slip angle on any of the tires is inspected from the simulation results of the ramp steer maneuver.

The valid lateral acceleration range of linearized expression is determined as 0.265 g where the slip angles are below the 2.5° degree target. (Figure 17).

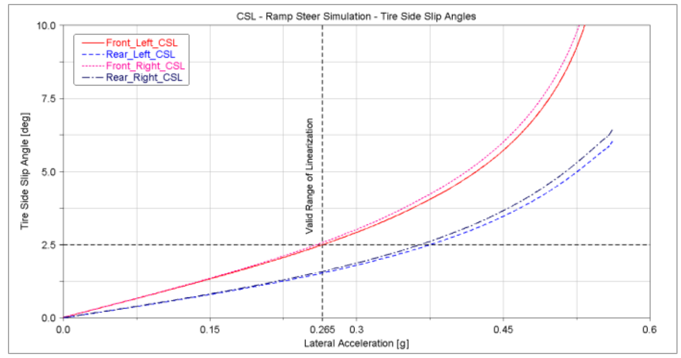

Figure 17: *Tire side slip angle with respect to lateral acceleration*

According to the calculated understeer gradient on Figure 18, the average understeer gradient inside the range of validity is 2.92 deg/g.

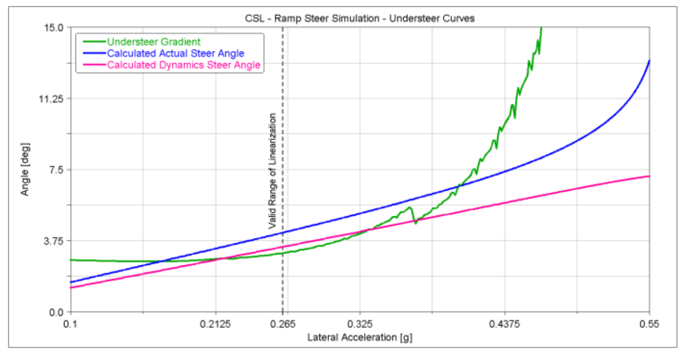

Figure 18: *Understeer gradient with respect to lateral acceleration*

CONCLUSION

In the future studies, full vehicle analysis is investigated by comparing the benchmark vehicles to each other by taking into account the target settings of Vehicle Dynamics characteristics.

Kinematics and compliance tests and steering analyses have been completed in ADAMS environment. They have been compared to the ones of CV10 and CV5 vehicles.

Finally a comfortable and better handling have been maintained for the developed vehicle.

REFERENCES

[1] Song, C., & Cai, Z. (2004). Modeling and simulation of double wishbone suspension based on ADAMS/CAR. *Journal of Jilin University of Technology (Natural Science Edition)*, *4*, 007.

[2] Gillespie T (2005) Using vehicle dynamics simulation as a teaching tool in automotive engineering courses, SAE Technical Paper Series, 2005

[3] Sanville, A. J., & Clark, W. (2008). *U.S. Patent Application No. 11/532,217*.

[4] Sancibrian, R., Garcia, P., Viadero, F., Fernandez, A., & De-Juan, A. (2010). Kinematic design of double-wishbone suspension systems using a multiobjective optimisation approach. *Vehicle System Dynamics*, *48*(7), 793-813.

[5] Hwang, J. S., Kim, S. R., & Han, S. Y. (2007, January). Kinematic design of a double wishbone type front suspension mechanism using multi-objective optimization. In *5th Australasian Congress on Applied Mechanics (ACAM 2007)*(Vol. 1, pp. 788-793). Engineers Australia.

[6] Gillespie T (1992) Fundamentals of vehicle dynamics, SAE International, 1992

[7] SAE Publications (2008) Vehicle dynamics terminology, SAE International, No. J670

[8] Apel, A., & Mitschke, M. (1997). Adjusting vehicle characteristics by means of driver models. *International journal of vehicle design*, *18*(6), 583-596.

[9] Mitchell, W. C., Staniforth, A., & Scott, I. (2006). *Analysis of ackermann steering geometry* (No. 2006-01-3638). SAE Technical Paper.

[10] ISO, D. (2013). 8855. Road vehicles-Vehicle dynamics and road-holding ability-Vocabulary. DIN Deutsches Institut für Normung e, 2013.

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