

# VEHICLE WHEEL POSITIONING INNOVATION ON A MACHINE FOR MEASURING THE CONTACT PARAMETERS BETWEEN A TYRE AND THE ROAD

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## Abstract

This article deals with the development and proposal to increase the application of equipment for measuring the adhesion force between the road and the road vehicle. The device enabling the adhesion force measurement of a rotating vehicle wheel is called a dynamic adhesor. The actual position of the vehicle wheel relative to the road-simulating cylinder is, in the present embodiment of the dynamic adhesor, by means of a bolted clamping joint and stops. This embodiment has limiting applications. The test must be aborted, if the position of the vehicle wheel relative to the rotating cylinder changes. Therefore, the adhesor has been upgraded. The upgrade consists of adding a hydraulic control to allow repositioning of the vehicle wheel online without interrupting the experiment, from a workstation outside the rotating parts area. Input load values were determined from the adherent desired operating parameters and from the load simulation in the MSC Adams software. Hydraulic cylinders were designed in accordance with the results of the maximum operating loads and installation space. The current construction was adapted and supplemented with new parts for the use of these hydraulic cylinders. The article only deals with the area of new key elements. The equipment's use has been expanded by improving the dynamic adhesor, for example, with the possibility of simulating the vehicle wheel's driving behaviour when the vehicle is propped up, and in this way, the equipment can also be used for any required research in regard to noise emitted from the tyre's road contact. As a result of innovation, the use of the unique tyre-testing equipment has been expanded, increasing the safety and environmental friendliness of road vehicles.

**Keywords:** adhesion force; vehicle testing; dynamic adhesor; electric car; FEM

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## 1. Introduction

A road vehicle's safety of movement is a key element in today's motorised era [16, 20], especially for achieving the goal of "The vision of zero casualties on roads" [18]. Today, in an effort to reduce the harmful effects of motoring, vehicle development is moving towards the electric mobility field [19] (Green Deal for Europe). It was the drive unit's electrification that achieved a significant reduction in pollutant emissions produced by the vehicle, redeemed by the limiting range per charge of the electric car. Therefore, current research is directed at the area of minimising the driving resistance of road vehicles [13]. For this reason, the electric car's body shape is optimised, and subsequently the vehicle wheel's rolling resistance is also addressed. Currently, the noise emitted by a vehicle wheels is a significant factor burdening the environment with modern electric cars. For vehicles with an internal combustion engine, this noise component was in the minority in relation to the noise emitted by the drive unit.

For the above reasons, tyre development must be carried out to ensure that tyre disadvantages are minimised while maintaining safety, maximum adhesion properties [1, 15] and acceptable vehicle handling characteristics. For this purpose, driving tests with real vehicles are used in a test corridor (pavement with a low coefficient of adhesion surface [4]), or a simulation numerical model [3, 5, 17]. In both cases, this involves simplification and difficult repeatability of the experiments. Therefore, research in the given area is increasingly directed towards laboratory measurements on test equipment [12]. One of these devices is the adhesion test rig. A static adhesion test rig is used to determine the tyre elasticity, while a dynamic adhesion test rig (DA) is used to determine the vehicle wheel's adhesion force and behaviour when rolling. The DA is a unique single-purpose device used to determine the dynamic properties of driven vehicle wheels; it is simultaneously possible to measure the forces transmitted in the contact between the tyre and the road as well as the noise emitted by the wheel rolling on the pad. A more detailed introduction to the dynamic adhesion test rig is given in [2]. The given dynamic adhesion test rig's design consists of a supporting frame, a rotating cylinder with a drive and a brake, a tilting head, a protective cover and a control panel. In the current design, the deflection and toe of the vehicle wheel are adjusted using a mechanical element. The DA modification achieves new possibilities for measuring the contact properties between the tyre and the road. The equipment upgrade enables tyre life testing compared to the current state. Due to the hydraulic control of the vehicle wheel position relative to the road (roller), it is possible to simulate the forces created by passing through a bend and subsequently the wheel position change due to the vehicle's suspension. Subsequently, it will also be possible to detect pollutant emissions that occur during tyre wear on the DA. It is not currently possible to determine these objectively in real operation. On the modified adhesion test rig it is possible to detect the following parameters with higher accuracy than at present:

- fatigue strength test,
- dynamic radius measurement,
- circumferential roughness test,
- dynamic radial stiffness and damping measurement,
- destruction rate test,
- noise level test,
- tyre heat test when the wheel is rolling,

- slip characteristics test,
- test for lateral guidance and resistance to bead deflection from the rim,
- durability and wear test,

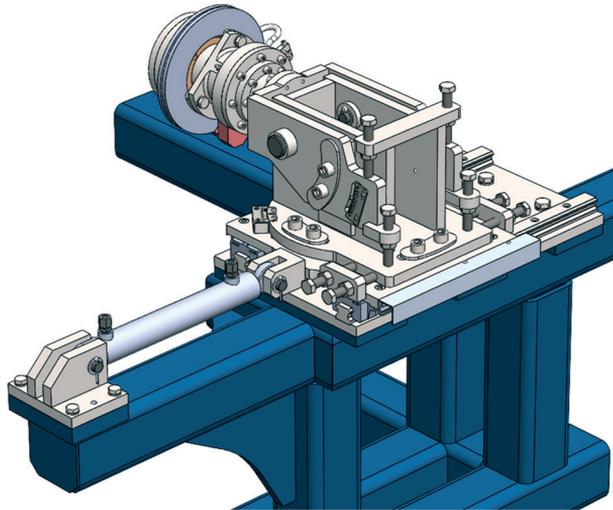
and these can be compared with the real conditions when measured on the vehicle, as detailed in [6]. At the same time, the measured DA data can be used to validate mathematical models [7, 9].

As part of the contribution, this mechanical adjustment in relation to the vehicle wheel position is replaced with hydraulic control. The result is the design of a unique device required to test vehicle wheels while ensuring the repeatability of measurements.

## 2. The adhesion's current design

The device is used to detect the contact parameters between the rolling vehicle wheels and the road. With regard to this device, the road surface is replaced by a cylinder with a 3.5 metre diameter. The cylinder is driven by a three-phase asynchronous motor. The main test parameters such as the cylinder's circumferential speed, the magnitude of the downforce, the speed of the downforce and the magnitude of the braking force on the wheel are implemented "by wire" from the control station. In addition to the above parts, the device consists of a measuring head, a support frame, cooling circuit, hydraulic generator and service brake. The vehicle wheel's actual position in relation to the cylinder is realised manually by changing the stops and the clamp on the measuring head. This part of the machine that is upgraded in the paper to allow the wheel position to be changed without interrupting the experimental measurement. The vehicle wheel's required radial reaction is induced by a hydraulic cylinder which presses the wheel against the cylinder at a speed defined by the user. This is the reason that the part of the DA described above is subjected to innovation. The innovation consists in the use of a hydraulic circuit and a pair of hydraulic cylinders. This modification requires the design and addition of an entire group of parts that allow the hydraulic cylinder assembly to be applied to the existing DA. The DA base is a bottom plate assembly with linear guide rails, which is fixed to the DA support frame (Figure 1).

Based on the existing hydraulic cylinder, this is used to move the vehicle wheel forward with the measuring head. This is how the wheel makes contacts with the cylinder and the derivation of the required pressure force is achieved. Next is the slide assembly, which contains all the tilting head components. The tilting head assembly is connected to the shift assembly via the main pivot pin, providing a change in the vehicle wheel's camber. The tilting assembly is also connected to the rotating head assembly via the tilting pins, which are also stored in the sliding bearings. This assembly is used to simulate convergence. A measuring head assembly is also attached to it and serves as a clamping element for the tested vehicle wheel.

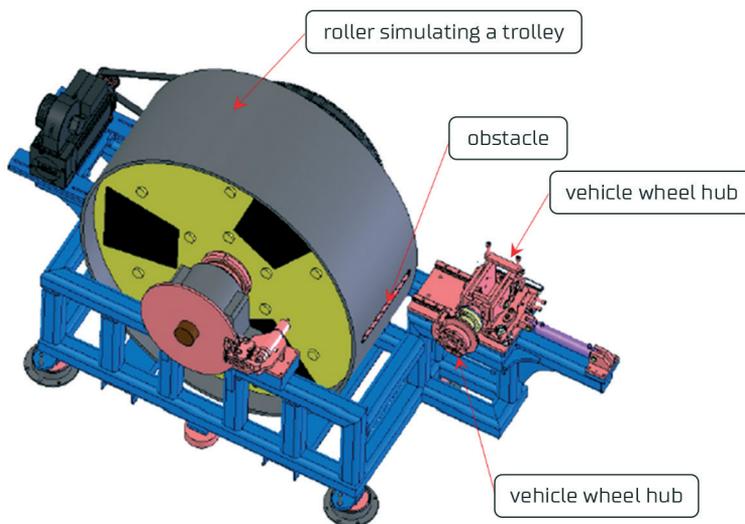


**Fig. 1. Initial DA version with mechanical wheel position adjustment**

The desired angle of deflection is set by using the screws located in two stops on the edge of the displacement assembly. These stops are connected by welds to the support plate assembly, which is part of the slide assembly. The angle of deflection and convergence of the vehicle wheel relative to the rotating cylinder (road) is done by unscrewing the stop bolts. The rotating head assembly is further supplemented by three deflection mounts, which create additional friction joints and therefore ensure that the head does not rotate during vehicle wheel testing. The toe-setting mechanism is solved in an analogous way. The disadvantage of the current arrangement is that changing the wheel position during testing is an impossibility.

### **3. Hydraulic cylinders design for vehicle wheel adjustment**

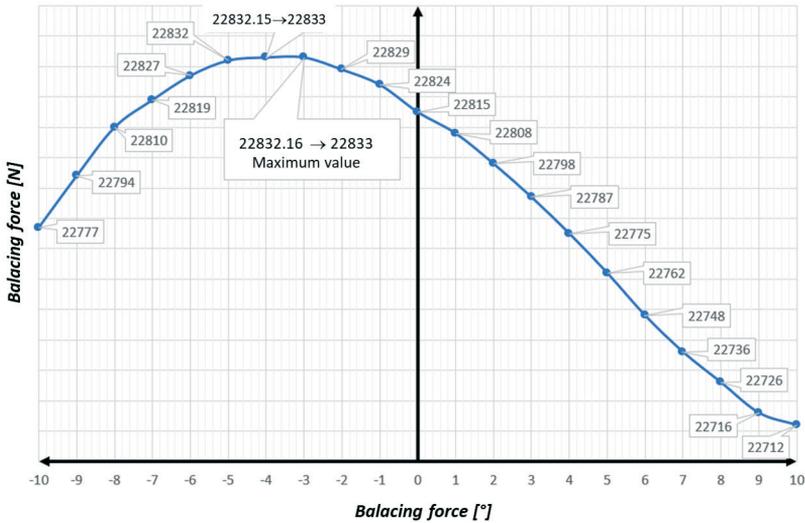
I proceeded to determine the dynamic effects after the initial orientation of the installation space for the hydraulic circuit. These are created by passing the wheel over obstacles placed around the perimeter of the road simulation cylinder. The input values for the design were obtained from the DA model in the MSC ADAMS SW (Figure 2).



**Fig. 2. Basis for multibody simulation**

The road shape is modelled in the form of a cylinder with obstacles on its surface. These obstacles are in 2 parts and are the same shape as in the real version. An adhesion coefficient  $\mu = 1.2$  was chosen to obtain the maximum values that can be applied to the DA. As a tyre model I used a validated tyre model – Pacejka, which I loaded with extreme parameter values, that the DA is able to achieve. These are the exact forces and stroke values that I use in the DA hydraulic control design. The maximum pressure that I designed the hydraulic cylinder is  $p_{MAX} = 25$  MPa. Because the maximum force required to change the toe of the wheel is lower than that of the camber ( $F_{R2} = 29200$  N); therefore, this amount of force was used for the design of both hydraulic cylinders.

Due to the limited installation space, I had to place the hydraulic cylinder to change the wheel camber outside the longitudinal axis of the DA. Due to misalignment, the force to change position is distributed vectorially. Therefore, the maximum control force is achieved at  $\varphi = -3^\circ$  from the starting position [Figure 3].



**Fig. 3. Force distribution for the hydraulic cylinder controlling the wheel camber**

I designed a specific version of the cylinders as HV 63-36-100 with integrated Temposonics RH-M-0100M-D60-1-A01 position measurement. The use of hydraulic cylinders required developing new components and modifying some existing DA components, where I only deal with newly designed key parts.

### Design and strength check of key head tilt components

The design and strength control of individual parts was implemented in the SolidWorks SW. For the non-linear analyses performed for the components [11], I chose the strategy of controlling the calculation using the arc length increment of the loading (equilibrium) curve. The load is proportionally increased so that balance is achieved at each step [8, 10].

The tilt handle is one of the key features. The calculation model includes the tilt mount itself and the parts of the tilt head sides where it is welded. I chose steel S355J2 [11 523] as the production material, and the strength control evaluation was carried out with elastic plastic behaviour according to von Mises plasticity.

I applied the load corresponding to the operating load value of the  $F_{RSMAX}$  component to half of the area of the holes for the pin at an angle that corresponds to the hydraulic cylinder tilting of toe-in when setting the angle value  $\varphi = -3^\circ$ . The data for the analysis and the model network are in Figure 4. The mesh size of the main elements is corresponding to 5 mm with a refinement in the area of the pin holes with a size of 2 mm.

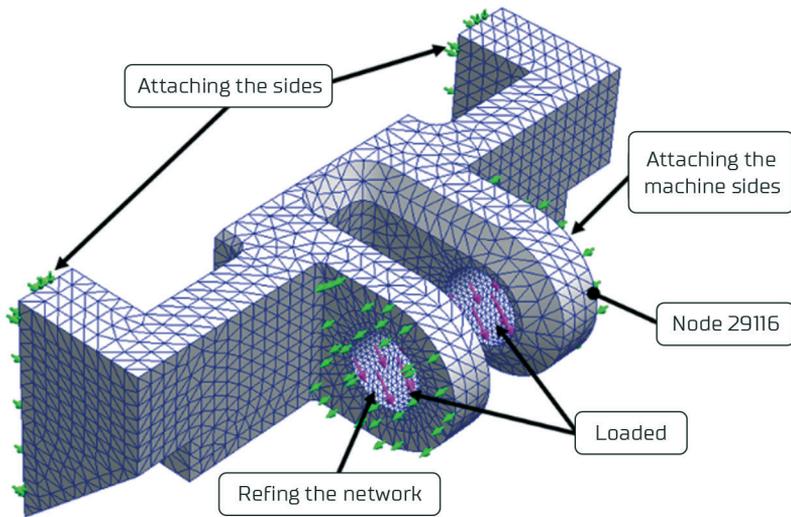


Fig. 4. Tilt-mount computational model

After performing a non-linear analysis, I found the load curve for the defined node 29116. At this node, the largest displacement occurs at the limit state of the load. The resulting load characteristics are shown in Figure 5.

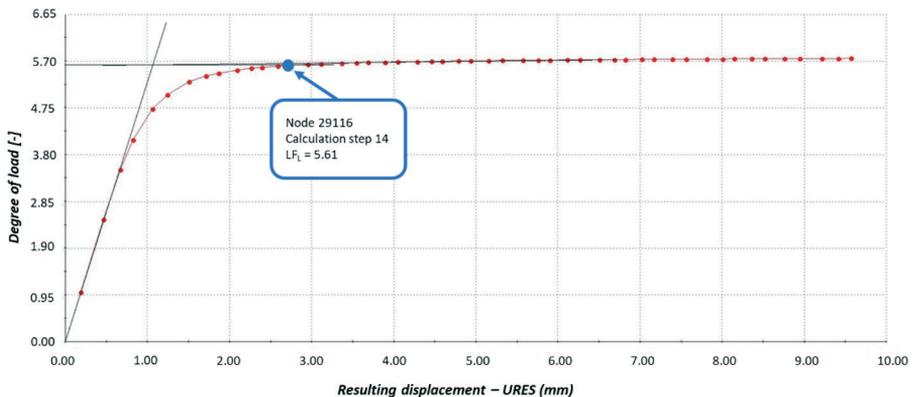


Fig 5. Load characteristics for the 29116 tilt mount assembly

Tangents were added to the curve in the elastic and plastic parts, when the corresponding value of the load limit factor was determined.  $LF_L = 5.61$ . In this case, the permissible limit load factor value  $LF_{LDOV}$  and the corresponding value of the permissible load  $F_{RSD}$  were

calculated in accordance with ČSN 690010 standard, which specifies the safety factor for the limit state of plasticity  $n_T = 1.5$ . Furthermore, it was necessary to take the type of joint weld joint into account with the  $\varphi_T = 0.7$  coefficient corresponding to the welds [14].

Calculating the permissible load limit factor value is done in accordance with the relationship [1]:

$$LF_{LDOV} = \frac{LF_L}{n_T} \cdot \varphi_T = 2.62 > 1 \rightarrow \text{OK} \quad (1)$$

$n_T$  – safety factor to the limit state of plasticity,

$\varphi_T$  – angle of deviation of the hydraulic cylinder from the longitudinal axis, °.

The calculating the permissible load value is in accordance with the relationship [2]:

$$F_{RSD} = F_{RSMAX} \cdot LF_{LDOV} = 59.8 \text{ kN} \quad (2)$$

$F_{RSMAX}$  – the given component's operating load, kN.

The permissible load limit factor value  $LF_{LDOV}$  is greater than 1; and therefore, the tilting handle is strong enough. Based on this value, the resulting permissible load value  $F_{RSD}$  was subsequently calculated.

The second key element that I subjected to strength analysis is the "L" brace, which is used to attach the hydraulic deflection cylinder to the DA support plate. The model material is S355J2 (11 523) with elastic plastic behaviour in accordance with von Mises plasticity. The load is placed on the half surfaces of the holes, where the deflection hydraulic cylinder is inserted through pivot pins, and its size corresponds to the maximum operating load force  $F_{R2}$  acting in the extreme position at the set angle. Furthermore, the boundary conditions are defined, where the plate's lower surface and its side surface created by cutting out of the unit are prevented from all displacements. The mesh element size was set to 10 mm. All the preparations of the computational model before starting the analysis are shown in Figure 6.

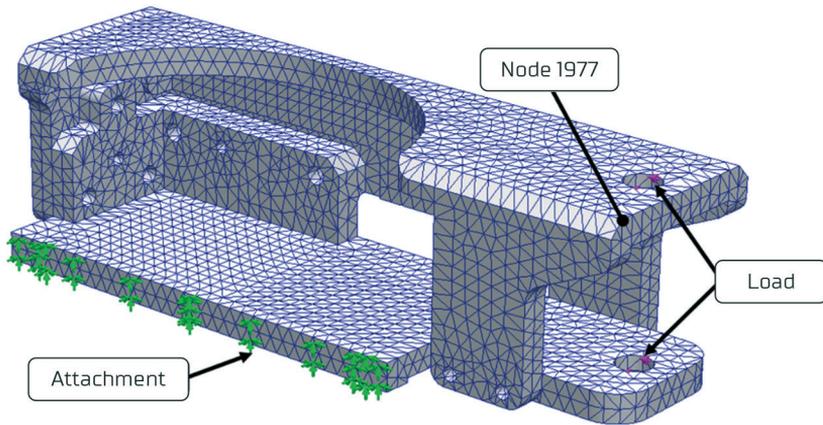


Fig. 6. Calculation model of the "L" brace

The analysis procedure is similar to the tilt mount. After carrying out a non-linear analysis, the load curve for node 1977 was found, where the largest displacement occurs at the load's limit state. The resulting load characteristic is shown in Figure 7.

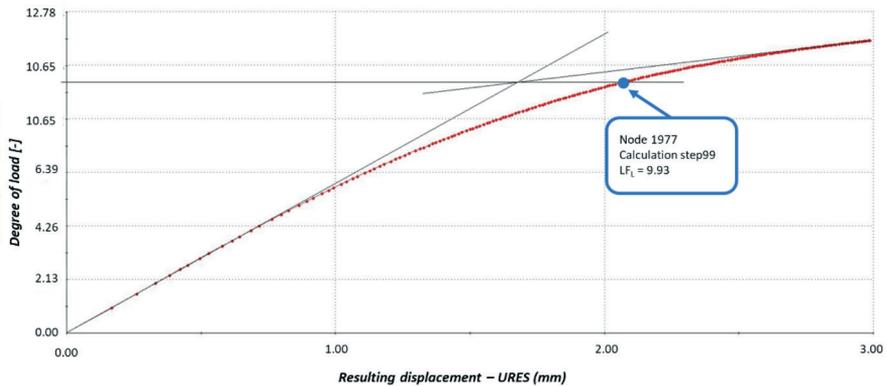


Fig. 7. The "L" brace loading characteristics

The "L" brace and the support plate assembly are a weld consisting of several parts, therefore the allowed limit factor value of load  $LF_{LDOV}$  is calculated in accordance with the ČSN 690010 standard with a safety factor for the limit state of plasticity  $n_T = 1.5$ , and the factor taking the weld joint  $\varphi_T = 0.7$  into account.

Furthermore, the distribution of the reduced von Mises stress [Figure 8] was determined in the load's limit state.

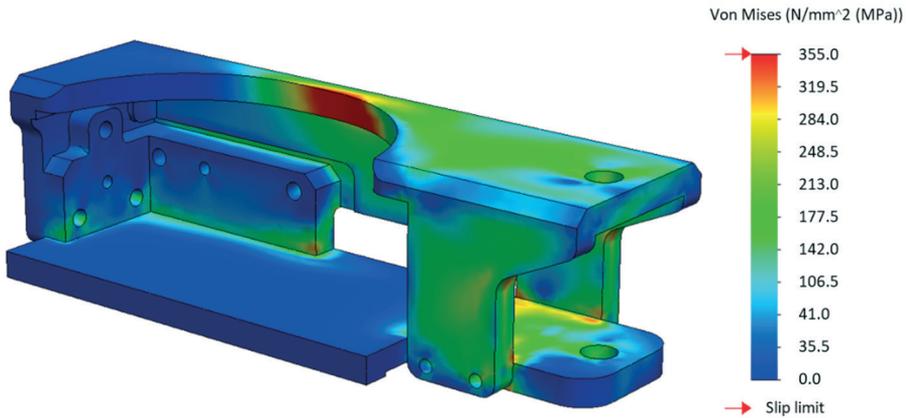


Fig. 8. Reduced von Mises stress for the limit state of the “L” brace load

Calculating the permissible load limit factor value is done in accordance with [3] and the satisfactory state in accordance with [5]:

$$LF_{LDOV} = \frac{LF_L}{n_T} \cdot \varphi_T = 4.63 > 1 \rightarrow \text{OK} \quad (3)$$

The permissible load limit factor value  $LF_{LDOV}$  is greater than 1; therefore, the “L” brace is suitable from a strength viewpoint. Based on this value, the resulting permissible load value  $F_{R2D}$  was subsequently calculated [4].

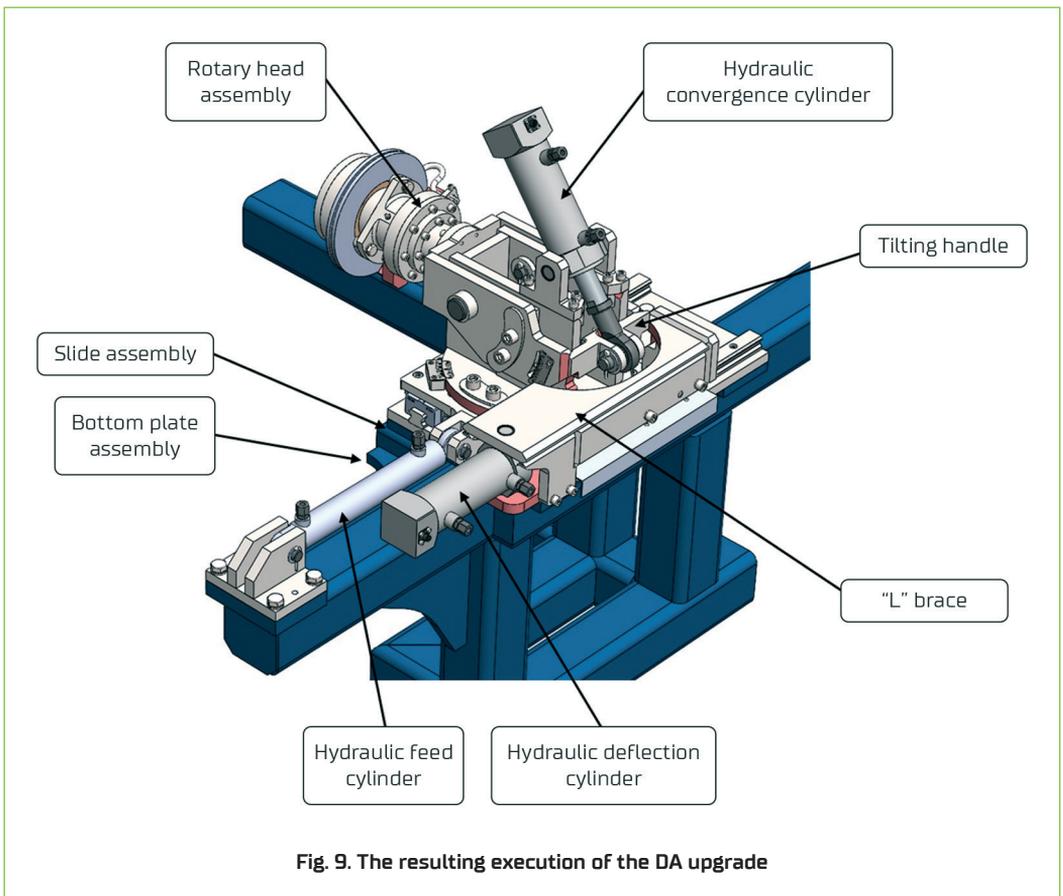
$$LF_{LDOV} = \frac{LF_L}{n_T} \cdot \varphi_T = 4.63 > 1 \rightarrow \text{OK} \quad (3)$$

Similarly, to the mounting side, a static analysis of the assembly was used to determine the loading forces that act on the individual bolts. This analysis used computational models of the “L” brace and parts of the base plate assembly that are bolted together. The material of both parts was established as linear elastic isotropic S355J2 [11 523]. The attachment is implemented in the same manner as the MNA the “L” brace analysis. Based on the obtained values, combinations of M10 and M12 bolt connections were chosen.

### Proposed concept assessment

All the designed components will now be combined as one. The rotary head assembly is connected to the feed assembly with a central pivot. This unit is also connected via ball guide carriages to the rails located on the bottom plate assembly. Subsequently, the hydraulic feed cylinder is connected via the existing pin assembly to the fork located on the feed assembly. Furthermore, the tilting assembly is connected to this unit via the tilting pins. Before connecting the hydraulic cylinder for toe-in, bronze washers are placed on its rocker pins as well as on the sides of the “L” and “P” mounts. The purpose of these washers is to define any

clearances, therefore ensuring smooth operation during the hydraulic cylinder movement by preventing the cylinder body from rubbing against the mounting sides. Next, the sides are set in a precise position using pins and connected to the swivel head assembly using bolts. The articulated head locks into the tilt bracket, which is part of the tilt assembly, through a locking pin. Bronze washers are fitted to the rocker pins before placing the tilting hydraulic cylinder. Subsequently, the cylinder is placed via one pin in the corresponding hole in the support plate of the slide assembly. Using a locking pin, the articulated head connects to the fork, which is part of the swivel head assembly. From the top side, the "L" brace assembly is placed on the second pivot pin, which is then anchored with pins to the support plate and secured with bolts. The last phase is to attach the headstock assembly to the tilt assembly's face plate, and this completes the tilt head. The resulting dynamic attached tilt head assembly is shown in Figure 9.



**Fig. 9. The resulting execution of the DA upgrade**

With the limited possibilities in terms of space and maximum use of the currently used DA elements, I designed the new components to make maximum use of the current layout concept. As the aim of this innovation was to ensure the vehicle wheel's adjustment from a place outside the attachments' working area, I used a hydraulic control system here which

can be connected to the existing hydraulic circuit, which is currently only used for the tilting head's forward movement.

## 4. Conclusions

The purpose of the contribution was to increase the use of the DA for road bikes. Many improvements have been made to make full use of the adhesor; this paper has discussed the design and development of the measuring head's hydraulic control for the purpose of being able to change the toe-in and camber of the vehicle wheel relative to the cylinder simulating the road. On the basis of the adhesion upgrade's implementation, there was an increase in the use of the device in experimental measurements. In the latest version, the adhesor can also be used for tests simulating real wheel-rolling conditions, including a vehicle wheel position change relative to the road due to the vehicle's suspension. However, this innovative design does take into account that it is possible to change these parameters continuously, that is, directly during the wheel test of the vehicle in question. This functional solution expands the range of possible simulations that can be performed on the DA.

A unique device was obtained based on the changes made; it is a powerful research tool regarding the contact of vehicle wheel properties and, simultaneously, a tool for research regarding the area of noise emitted by vehicle wheels, which is currently, with the advent of electromobility, one of the most significant factors in the negative impact of emissions on environmental impact.

## 5. Nomenclature

DA dynamic adhesor – a device for measuring the adhesion force between the rotating tyre and the road

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