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SELECTED ISSUES RELATED TO THE RESEARCH ON A HYBRID THREE-WHEELER FOR THE DISABLED

WYBRANE ZAGADNIENIA BADAŃ TRÓJKOŁOWEGO POJAZDU HYBRYDOWEGO DLA OSÓB NIEPEŁNOSPRAWNYCH

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Summary

The article concerns the performing of selected experimental and simulation tests carried out when engineering a motor vehicle for the disabled, provisionally named PIMOTEK. The project was subsidized by the State Fund for Rehabilitation of Disabled Persons (PFRON) within a Research and Analysis Programme. The vehicle, engineered at the Automotive Industry Institute, has a three-wheeled running gear and a load-bearing structure that enables tilting of the vehicle together with its driver in relation to the road surface and rear vehicle axle. The solution that enables the tilting of vehicle body is well known from the Gyro series of Honda scooters. However, it was modified with respect to the needs of users with impaired body balance abilities. A unique solution incorporated in PIMOTEK is a hybrid internal combustion and electric powertrain mounted on the rear vehicle axle.

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The electric drive system simultaneously solves the problem of reversing, which is indispensable for disabled persons. The three wheeler is provided with a lift-and-swivel seat, which makes the transfer between the seat and the wheelchair easier for a disabled person and, at the same time, enables the disabled person to put the wheelchair in the luggage compartment by himself/herself. The innovativeness of the vehicle also manifests itself in the modern lines of the vehicle body, masking the features of the vehicle as an invalid carriage.

Keywords: three-wheeler, disabled persons, functionality, stability

Streszczenie

Praca dotyczy realizacji wybranych badań doświadczalnych i symulacyjnych zrealizowanych podczas projektowania pojazdu dla osób niepełnosprawnych o roboczej nazwie „Pimotek". Projekt został dofinansowany ze środków Państwowego Funduszu Rehabilitacji Osób Niepełnosprawnych w ramach programu Badań i Analiz. Zaprojektowany w Przemysłowym Instytucie Motoryzacji pojazd posiada trójkołowy układ jezdny oraz konstrukcję nośną umożliwiającą pochylanie nadwozia pojazdu wraz z jego kierowcą względem płaszczyzny drogi i tylnej osi pojazdu. Zastosowanie rozwiązania konstrukcyjnego umożliwiającego pochylanie nadwozia jest doskonale znane z serii skuterów Gyro marki Honda. Zostało jednak zmodyfikowane pod kątem potrzeb użytkowników o słabym balansie ciałem. Unikalnym rozwiązaniem konstrukcyjnym pojazdu „Pimotek" jest zastosowanie hybrydowego układu napędowego, spalinowo – elektrycznego, zabudowanego na tylnej osi pojazdu. Jednocześnie napęd elektryczny rozwiązuje problem niezbędnego dla osób niepełnosprawnych rewersu. Trójkołowiec jest wyposażony w windowany fotel obrotowy, ułatwiający osobie niepełnosprawnej przesiadanie się z jednoczesną możliwością samodzielnego umieszczania wózka inwalidzkiego w przestrzeni bagażowej. Innowacyjność pojazdu przejawia się również nowoczesną linią nadwozia o zamaskowanych cechach pojazdu inwalidzkiego.

Słowa kluczowe: trójkołowiec, niepełnosprawni, funkcjonalność, stabilność

1. Object under test

The object subjected to the tests was a three-wheeled vehicle with a hybrid drive system, intended for disabled persons. The hybrid powertrain includes an electric motor and an internal combustion (IC) engine, which can operate both simultaneously and alternately. Each of the prime movers drives one of the two wheels of the rear vehicle axle. The vehicle is intended for the disabled to travel shorter or longer distances on their own, without any assistance of third parties. Thanks to its construction and functionality, the vehicle may also be used by able-bodied persons as well.

The IC engine is chiefly intended for operation in non-built-up areas; in urban areas (such as urban roads, walkways, parks, overpasses, roofed shopping areas, etc.), primarily the electric motor should be used. The electric motor used in towns reduces pollutant emissions, enables the vehicle to move in roofed areas, where the emission of exhaust gases is forbidden, and may also assist the IC engine e.g. on uphill drives or unpaved roads. The electric motor's capability of bidirectional operation enables reversing, which is required from vehicles for the disabled. Furthermore, thanks to the controllable height of the swivel driver's seat, the disabled person may be temporarily lowered to the wheelchair level where, when sitting sideways to the direction of travel, driver will be able to put the wheelchair to the specially designed luggage compartment in the vehicle and to fix the chair in place. Then, after the seat is lifted and turned to the direction of travel, the vehicle will be ready to drive.

In the PIMOTEK design, the IC engine drives the rear right wheel through a variable-speed belt transmission, forwards only. The rear left vehicle wheel is driven by an electric motor incorporated in the wheel hub, both forwards and rearwards. The rear wheels do not move in relation to each other, i.e. a dependent rear suspension system of the vehicle is thus formed. Rear suspension movements in relation to the vehicle body are enabled by the swingarm type system of fastening the rear axle to the vehicle body

Characterization of the construction of three-wheelers

Predominantly, three-wheelers are characterized by low unladen mass and functionality differing from that of four-wheeled vehicles. First, the three-wheelers are in most cases type-approved as vehicles of category L. Their classification in the group of motorcycle-type vehicles can also be noticed in the design solutions typical for motorcycles.

The idea of creating a three-wheeled mechanical vehicle dates back to the renaissance epoch: it was at that time when Leonardo da Vinci designed a vehicle driven by clockwork. That three-wheeled vehicle looking like a cart was provided with a driving system resembling a clockwork mechanism; for the vehicle wheels to be set in motion, the springs of the mechanism had to be tightened. Regardless of the spring tension, the vehicle moved with a constant speed and could change the direction of its motion, which could be pre-programmed. Although that invention was rather not intended for the transportation of people, it is now considered as the first prototype of a motor vehicle.

A mechanical road vehicle closer to the present-day solutions was the "Fardier a vapeur" three-wheeler designed by Nicolas-Joseph Cugnot and launched in 1769. However, the actual beginning of motorization is identified with the first vehicle provided with an IC engine, i.e. a three-wheeler patented and presented by Karl Benz in Mannheim in 1886. That vehicle had two rear wheels and a single steered front wheel. Such a configuration solved the problem with vehicle steering.

As regards PIMOTEK, its three-wheeled running gear has made it possible to attain low vehicle mass and to avoid the necessity of using any additional support (prop) when its linear speed is zero. Three-wheelers are an excellent compromise between motorcycles and cars. However, they also have some drawbacks, which can be eliminated by special solutions; in the vehicle under consideration, a solution of this kind is the tilting body.

This three-wheeled invalid carriage has been chiefly designed for the disabled. Therefore, it has been provided with solutions enabling a disabled person to travel in it on his/her own, without any assistance of third parties. According to the assumptions made, the IC engine is to power the vehicle in road traffic and the electric motor is chiefly intended for driving the vehicle in closed areas, such as e.g. shopping malls. In this connection, the vehicle should be characterized by relatively small overall dimensions and, simultaneously, adequate stability. At previous project stages (already completed), vehicle versions with a single steered front wheel and an independent or dependent rear wheels suspension system were considered.

1.1. Vehicle with an independent rear suspension system

When designing the vehicle with independent suspension of the rear axle, an assumption was made that the electric motor would drive the front wheel and the IC engine would drive the two rear wheels. The front wheel suspension system was adapted as a complete unit from a scooter. The rear wheels were mounted on swingarms, thanks to which the vehicle body height could be adjusted so that the body could be lowered to improve vehicle stability when the disabled person moved from the wheelchair to the vehicle seat. An example of the independent rear wheels suspension system has been shown in Fig. 1.

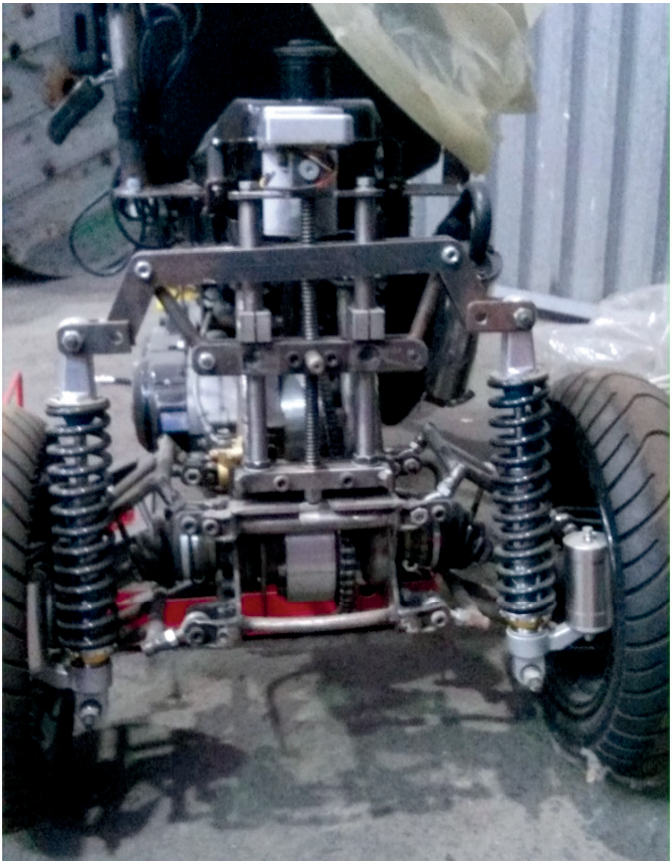

Fig. 1. Three-wheeler with the independent rear wheels suspension system

A CAD model of the vehicle, with the independent rear wheels suspension system, has been presented in Fig. 2. The significant degree of complication of the solution with the independent rear wheels suspension system was a reason for other solutions to be sought [source: authors' materials].

Fig. 2. CAD model of the three-wheeled invalid carriage with an independent rear wheels suspension system [source: authors' materials]

1.2. Vehicle with a dependent rear suspension system

In consideration of complicated design of the three-wheeled invalid carriage with the independent rear wheels suspension system, a decision was made to modify and simplify the vehicle construction. In consequence, the rear suspension system was made similar to Honda Gyro three-wheeled tilting motorcycle [27]. The construction of the rear suspension system of the said motorcycle was safe enough for the motorcycle to be approved for use in road traffic; therefore, after some modifications, it could be adapted for an invalid carriage.

To build the second physical prototype (Fig. 3), the Romet Ride Design 50 scooter was used, with its rear suspension system having been replaced with a system of new design [source: author's materials]. The main component, of the new suspension system, is a tilting swingarm (Fig. 4), which was originally used in Honda Gyro scooters. The said swingarm, when assembled, enables limited rotation of its two parts in relation to each other, thanks to which the vehicle body can tilt.

Fig. 3. The second prototype of the three-wheeled invalid carriage, with the dependent rear wheels suspension system [source: authors' materials]

Fig. 5 shows the prototype with the tilting mechanism in the position of maximum tilt of the vehicle body to the right

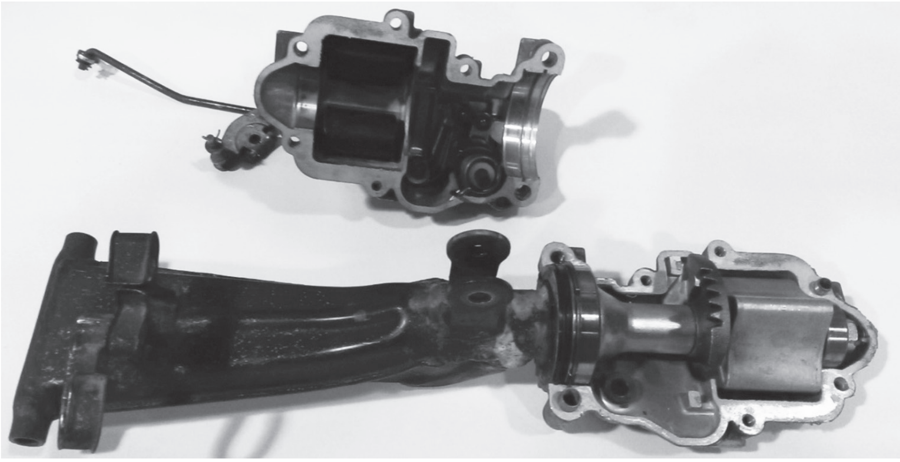

Fig. 4. Tilting swingarm used in the Honda Gyro motorcycle [source: authors' materials]

Fig. 5. The second prototype of the three-wheeler in the position of maximum tilt of the vehicle body [source: authors' materials]

2. Models of vehicle loads

Model of the forces and moments

Vehicles, whether four- or three-wheeled, undergo the action of dynamic forces. The average values of the dynamic forces have a cyclic nature; they may be represented by a sinusoid with constant amplitude. Conversely, their maximum values appear as single force pulses caused by local road irregularities. In practice, we know that if a specific structure can withstand, with no damage, the maximum forces acting sporadically then it is likely to have adequate fatigue strength, too.

The computational loads of a structure may be defined, by determining dynamic coefficients to multiply the static forces. For the three-wheeler, they are based on the gross vehicle mass (GVM). This method has been described in [2].

The dynamic forces acting on a vehicle may be expressed as follows:

$$F_d=\frac{F_{st}}{g}\cdot a=F_{st}\cdot m \tag{1}$$

where:

F_{st} – static force acting on the vehicle, defined by vehicle weights;

g – acceleration of gravity;

a – acceleration acting on the three-wheeler;

m – dimensionless coefficient of mass forces.

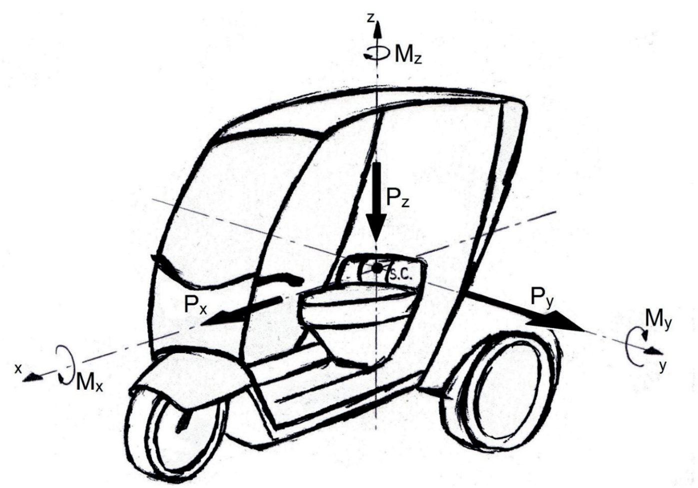

Fig. 6. The forces and moments applied to the three-wheeler as loads
(s.c. = cg = centre of gravity – translator's note)

To summarize, the following forces act on the vehicle body:

$$P_x = m_x \cdot (F_{st} - F_{nr}) \quad (2)$$

$$P_y = m_y \cdot (F_{st} - F_{nr}) \quad (3)$$

$$P_z = m_z \cdot (F_{st} - F_{nr}) \quad (4)$$

where:

F_{nr} – gravity force of unsprung masses;

M_x – twisting moment;

M_y – bending moment in the x-z plane;

M_z – bending moment in the x-y plane.

(The moments have only been shown in Fig. 6).

Model of the road

The road model, presented in Fig. 7, shows the basic types of excitations most frequently received by vehicles moving on roads. The longitudinal, lateral, and vertical forces assumed in the computations are set on the grounds of simulation tests carried out on the said road model. The model has been divided into sections, which characterize normal road conditions.

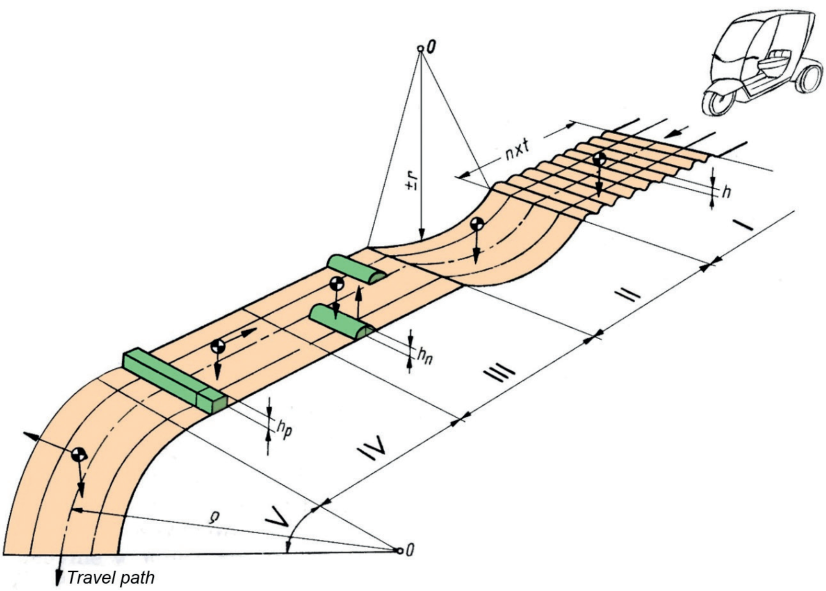

Fig. 7. Model of the road, which represents the basic events occurring on roads

Sections I and II simulate symmetrical loads. The road sections represent, in succession, symmetrical irregularities and vertical curvature.

Section III represents asymmetrical irregularities, which cause asymmetrical vertical loads.

Section IV simulates longitudinal loads. The road has a transverse bump with a specific height, causing a load to be applied to the vehicle structure in the direction of vehicle motion.

Section V simulates a lateral load, acting on a vehicle moving along a curvilinear path with a radius of ρ.

The acceleration values on the specific road sections depend on vehicle speed, shape of the road, and vehicle motion mode.

Symmetrical vertical loads

At the instant when the vehicle runs onto an obstacle, of this kind and the front wheel, and both rear wheels simultaneously run onto the irregularity, a force described by equation (5) is applied to the vehicle body:

$$P_{zs} = m_{zs} \cdot (F_{st} - F_{nr}) \tag{5}$$

where:

m_{zs} – dimensionless coefficient of symmetrical mass forces.

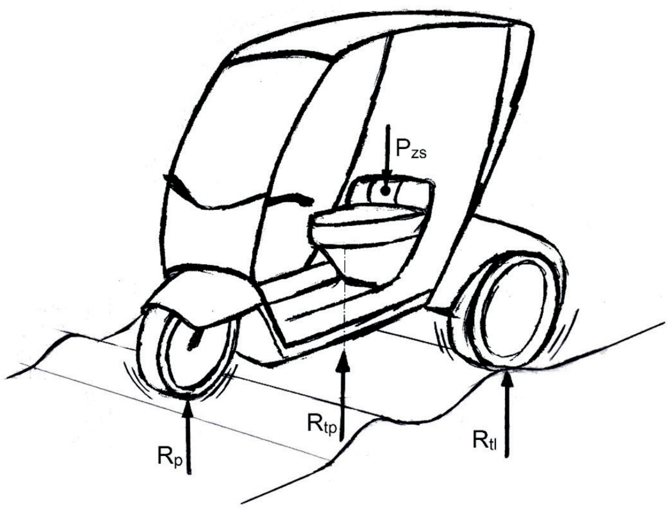

Fig. 8. Schematic diagram of the action of symmetrical vertical loads
R_p – road reaction acting on the front wheel;
R_{tp}, R_{tl} – road reactions, equal to each other, acting on the rear right and left wheels, respectively;
P_{zs} – force acting on the vehicle body and bending the load-bearing structure

Tests carried out on real objects have shown that for passenger cars and buses, the maximum symmetrical vertical accelerations are within a range of 15-25 m/s^2. For three-wheelers, the value of the dimensionless coefficient of symmetrical mass forces was assumed for the calculations as m_{zs} = 2.0 [2].

Asymmetrical vertical loads

At the instant when one of the rear vehicle wheels runs onto an obstacle provided in section III (Fig. 7), asymmetrical vertical accelerations are generated. The wheels of the same axle undergo different road reactions, which produce not only a bending moment but also a twisting moment. The force acting on the vehicle body structure, in result of the accelerations is:

$$P_{zns} = m_{zns} \cdot (F_{st} - F_{nr}) \tag{6}$$

where:

m_{zns} – dimensionless coefficient of asymmetrical mass forces.

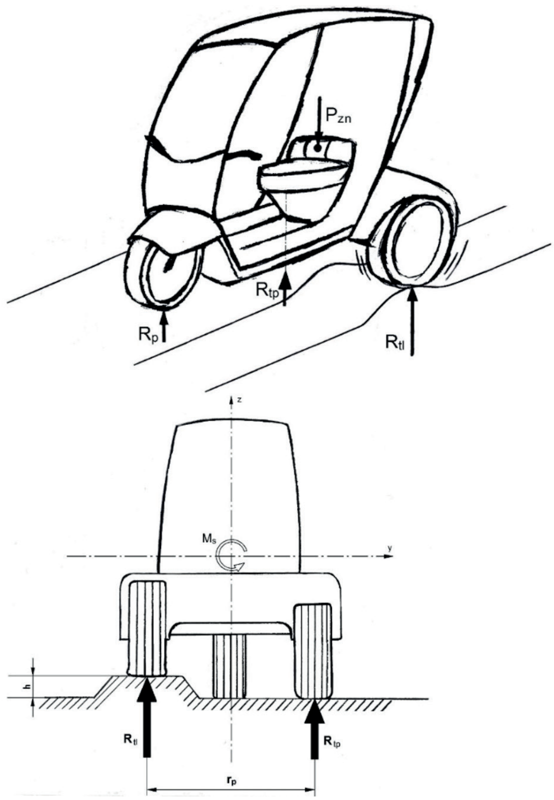

Fig. 9. Schematic diagram of the action of asymmetrical vertical loads

According to the experiments, described in the literature [2], the dimensionless coefficient of asymmetrical mass forces is m_{zns} = 1.3.

When the twisting moment reaches its maximum value, the right wheel will be lifted off (separated from the road). Of course, such a situation is quite unlikely in the case of a three-wheeler; for safety reasons, however, the vehicle body has been so designed that it will be capable of withstanding the maximum possible loads.

For a real road to be considered passable for three-wheelers, the maximum acceptable irregularities are assumed as ±75 mm, with positive and negative values corresponding to bumps and pits, respectively.

Stiffness of vehicle structure

In vehicle testing, vehicle body stiffness is one of the first and basic parameters that define the quality of the load-bearing structure having been developed. Depending on vehicle type, the importance of either torsional or bending stiffness predominates.

Torsional stiffness

The torsional stiffness of a vehicle body is a crucial parameter in the designing the vehicle's load-bearing structure. It describes the resistance offered by the vehicle body to deformation, when subjected to moments twisting the structure. Fig. 10 presents a schematic diagram of loading the vehicle's load-bearing structure with a concentrated twisting force.

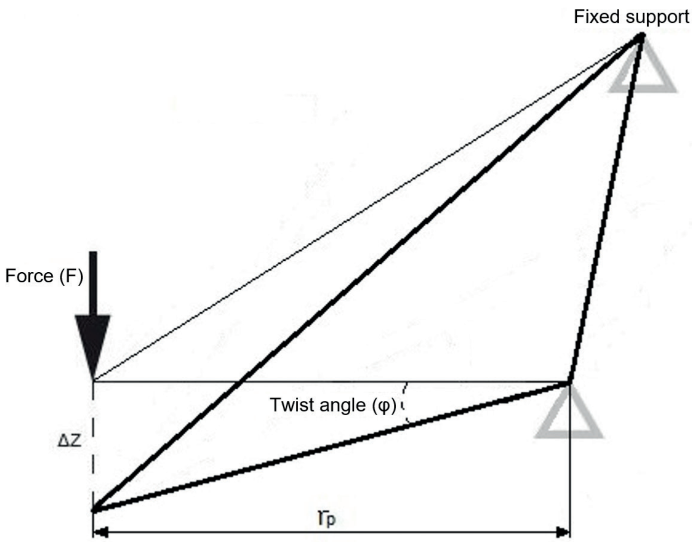

Fig. 10. Schematic diagram of loading a twisted vehicle's load-bearing structure with a concentrated force

$$\varphi = arctan\left(\frac{\Delta Z}{r_p}\right)[deg]; \tag{7}$$

$$M_s = r_p \cdot F\ [Nm]; \tag{8}$$

$$K_s = \frac{M_s}{\varphi}\left[\frac{Nm}{deg}\right]; \tag{9}$$

φ – vehicle body twist angle;

M_s – vehicle body twisting moment;

K_s – torsional stiffness.

Bending stiffness

The bending stiffness refers to the difference between the pitch angles of the front and rear parts of the vehicle body. The vehicle body bends during acceleration and braking. The vehicle frame is also subjected to higher or lower loads depending on the cargo transported. In the designing of motor vehicle bodies, an opinion prevails, that high torsional stiffness will also ensure high resistance to bending loads.

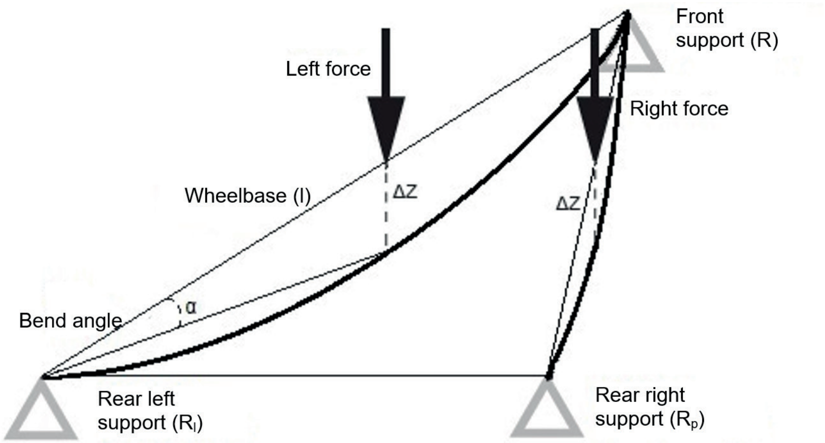

Fig. 11. Schematic diagram of loading a bent vehicle's load-bearing structure with a concentrated force (resolved into two symmetrical components)

$$\alpha = arctan\left(\frac{\Delta Z}{\frac{1}{2} \cdot l}\right)[deg] \tag{10}$$

$$M_g = \frac{1}{2} \cdot l \cdot R_l\ [Nm] \tag{11}$$

$$K_g = \frac{M_g}{\alpha}\left[\frac{Nm}{deg}\right] \tag{12}$$

α – vehicle body bend angle

Mg – vehicle body bending moment

Kg – bending stiffness

Testing of the vehicle frame

A discretized model of the vehicle frame, prepared in the HyperWorks program, was subjected to, inter alia, simulation tests carried out to determine such parameters as the values of twist angle or deflection (bending) of vehicle's load-bearing structure, as functions of the vertical forces, acting on the vehicle and to calculate the torsional and bending stiffness of the structure. The vehicle's load-bearing structure has the form of a tubular frame, although of the open type but having been strongly stiffened in its lower portion. Therefore, the bending stiffness of this structure may be expected to be much higher than the torsional stiffness.

2.1. Determining the torsional stiffness by simulation tests

To determine the torsional stiffness of the vehicle frame, a twisting moment of a predefined value was applied to the frame in a way as shown in Fig. 12 and the twist angle was measured. Pursuant to (9), the torsional stiffness was calculated as the twisting moment to twist angle ratio.

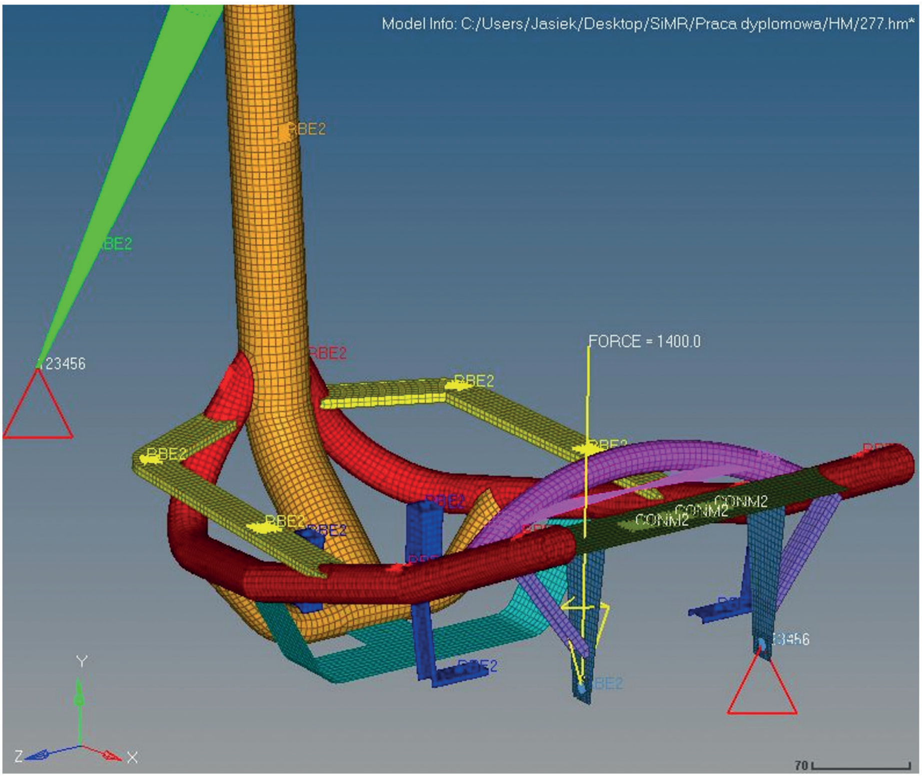

Fig. 12. Method of loading the vehicle frame to measure its torsional stiffness [source: authors' materials]

For the twisting moment of 224 Nm applied to the frame in the simulation tests, the twist angle was 0.47 deg. Hence, the torsional stiffness of the vehicle frame was found to be

$$K_s = \frac{224\ Nm}{0{,}47\ deg} = 476\frac{Nm}{deg}$$

2.2. Determining the bending stiffness by simulation tests

The bending stiffness was determined with using a model where the vehicle frame, with its steering head (frame neck marked in green in Fig. 13) being fixed, was loaded with forces defined by the vehicle structure loads caused by vehicle motion. The bending stiffness was calculated as the bending moment to bend angle ratio.

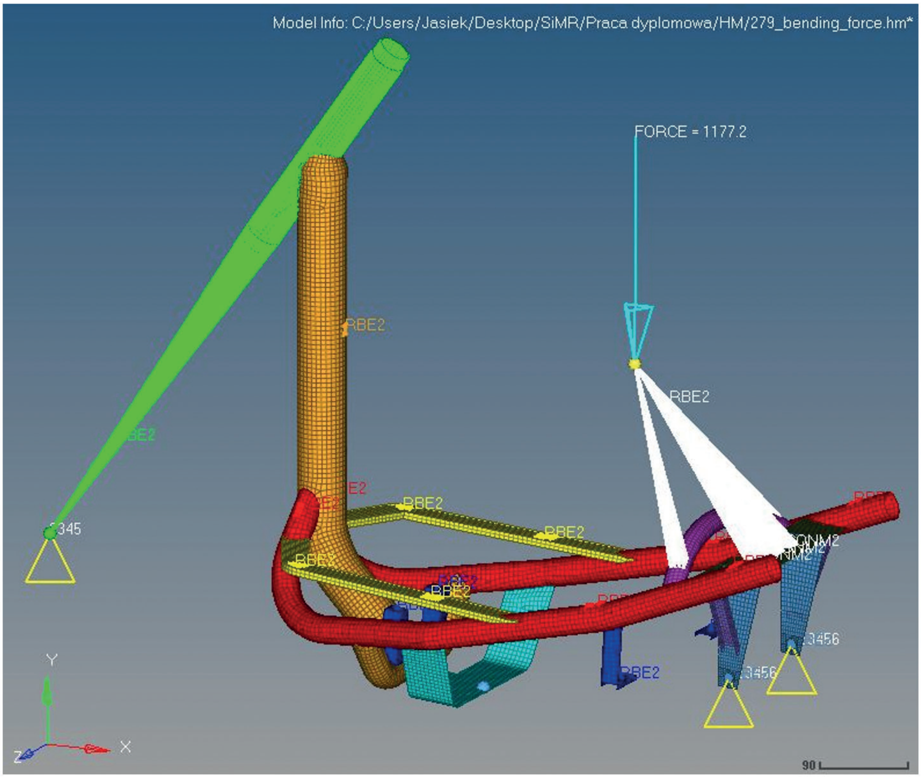

Fig. 13. Loading the vehicle frame when determining its bending stiffness [source: authors' materials]

The loads applied to the simulation model and the reactions determined on these grounds, made it possible to calculate the bending moment, and the frame bend angle and, in consequence, to calculate the bending stiffness, which was found to be:

$$K_g = \frac{588{,}6}{0{,}031} = 18987\ \frac{Nm}{deg}$$

2.3. Determining the torsional stiffness by experiments

The torsional stiffness of the vehicle frame was also determined experimentally, as a basic test to verify the vehicle structure quality and the simulation methods adopted. For technical purposes, special adapters were made, dedicated to the vehicle's load-bearing structure having been prepared, for the structure to be positively fixed and loaded with an appropriate twisting moment.

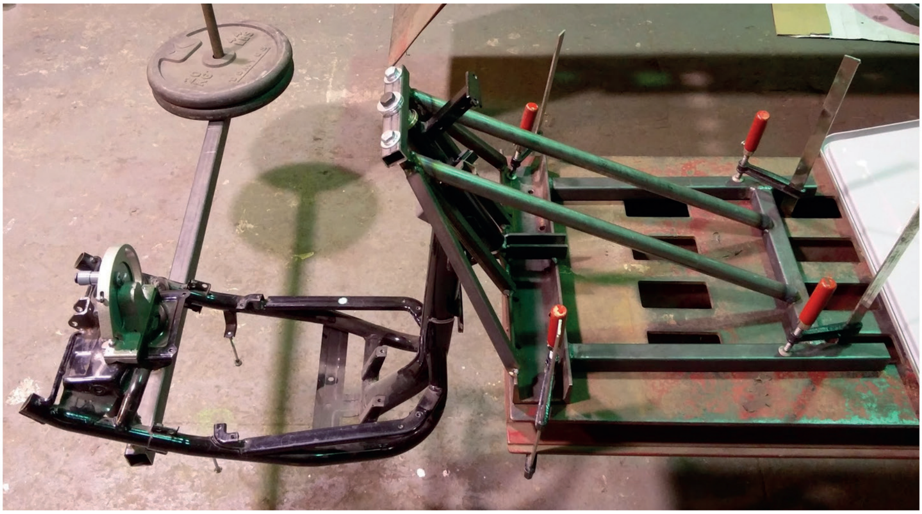

Fig. 14. Test rig for determining the torsional stiffness of the frame of a three-wheeled vehicle

To load the vehicle frame with a twisting moment, a force defined by the mass of weights was applied to an arm at a distance of 1 m from the frame centreline. The frame twist angle was read from a protractor situated, in the plane of twisting the frame, with a moment applied by the loading weights.

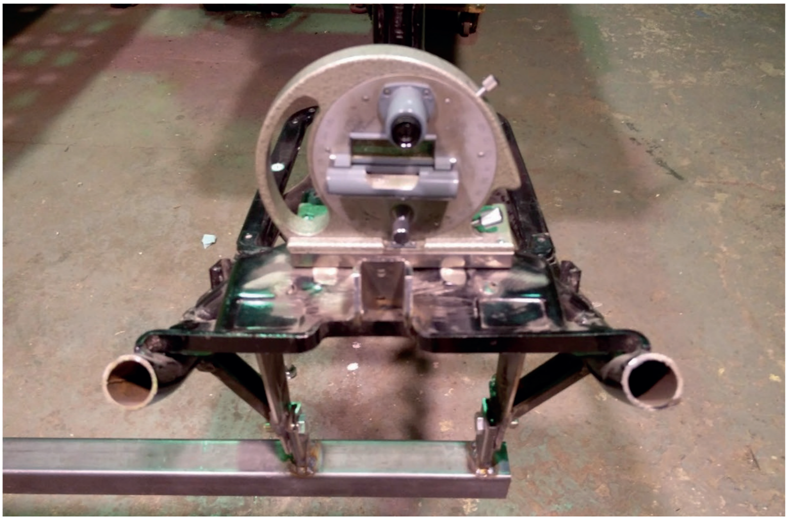

Fig. 15. Mounting of the protractor used to measure the vehicle frame twist angle

With using 20 kg, and 10 kg weights, five measurements were carried out with the loads applied varying from 20 kg to 80 kg, in succession and producing twisting moments from 196.2 Nm to 784.8 Nm. The torsional stiffness at individual loads was determined by dividing the twisting moment values by the corresponding values of the frame twist angle, in result of which a torsional stiffness characteristic curve could be plotted. The measurement results have been presented in Table 1 and Fig. 16.

Table 1. Results of rig measurements of the three-wheeler's frame torsional stiffness

Item	Twisting moment [Nm]	Twist angle [deg]	Twisting moment to twist angle ratio [Nm/deg]
1.	196.2	0.48	408.75
2.	392.4	1.40	208.28
3.	588.6	2.35	250.47
4.	686.7	3.10	221.51
5.	784.8	3.42	229.47

The lowest level of loading the frame, with a twisting moment during the experiments, corresponded to the value of the twisting moment applied to the frame model in the simulation tests. The twisting moment to twist angle ratio in the experiments and simulation tests was 408.75 Nm/deg and 474 Nm/deg, respectively. This conformity between simulation and experimental test results is satisfactory; however, the torsional stiffness value thus determined corresponds to the first measurement point only, as it can be seen in Fig. 16. The actual value of the torsional stiffness of the vehicle frame was determined by approximating the torsional stiffness values obtained at various loads by a linear function. The approximating line has been marked in red in Fig. 16; based on this, the actual torsional stiffness has been estimated at 193.82 Nm/deg.

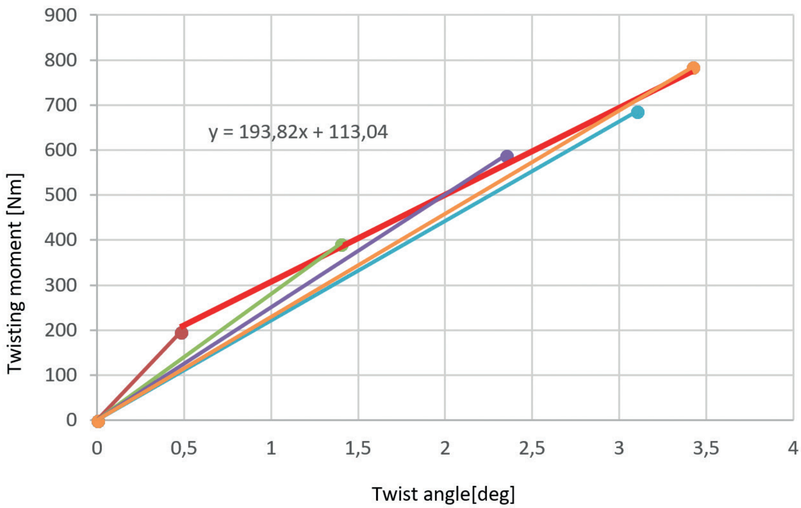

Fig. 16. Graph characterizing the torsional stiffness of the three-wheeler's frame

The full text of the article is available, in Polish, online on the website http://archiwummotoryzacji.pl.

Tekst artykułu w polskiej wersji językowej dostępny jest na stronie http://archiwummotoryzacji.pl.

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