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Selected aspects of power distribution for 4wd vehicle driven by diagonal power transmission

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Analysis and modelling

ABSTRACT

Purpose: Paper describes phenomena related to the vehicle acceleration on the curve, with the high speed, considering inertia forces and driving force.

Design/methodology/approach: The phenomena were investigated with three tools: classical analytical mechanics, computer simulation and tests with physical model.

Findings: Simulations and physical tests led to similar results. However, new diagonal transmission system showed advantages only on slippery surface of the test track. Nevertheless such a behavior resulted in conclusion that diagonal transmission system would be useful influencing vehicle safety, specifically at slippery roads.

Practical implications: There is a possibility of diagonal transmission system application for sport vehicles to improve performance. For utilitarian vehicles increasing safety is the most important aspect, especially in cooperation with ESP system. When it comes to special and military vehicles both mentioned above features are desired. Possible power-pack application is an additional advantage.

Originality/value: The goal was the comparison of standard 4x4 transmission system with unconventional patented 4x4 transmission system called diagonal one.

Keywords: Computer Assistance in the Engineering Tasks and Scientific Research; Numerical Techniques, Technological Design; Applied Mechanics

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1. Introduction

Power transmission systems of cars in current shape (2011) are the result of many year's evolution. However, the history of cars lasts longer than 100 years, dynamic development of four wheels driven passenger cars started in the eighties. Such a driving system indicates many advantages in comparison with one driven axle. The most essential factor is safety and in this aspect 4wd exceeds all the other solutions. Contemporary 4wd transmission systems in spite of a big variety of design solutions and operating rules have one common feature. Wheels of each particular axle are connected together by means of one of many differential gears or clutches. Such a design precludes a priori power distribution optimisation. Observations of rally cars lead to conclusions and result in a new approach to the manner of power distribution. The reasoning is as follows: [8,9,10]

- Assuming some simplifications, it can be stated that during monotonous motion of the car all resistance forces result in underloading of the front axle and increasing the load of the rear axle (Rally cars are equipped with special aerodynamic body elements so called spoilers and deflectors to counteract this lifting force). Such a layout of forces results in lifting the front part of the vehicle shown in the Fig. 1.

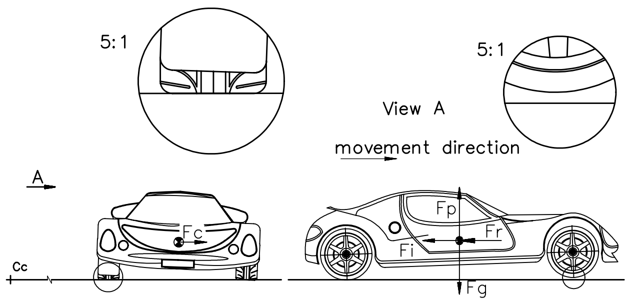

Fig. 1. Front part of the vehicle

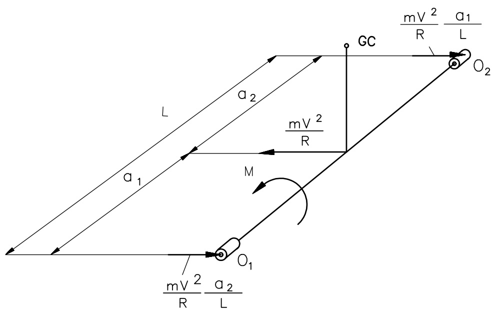

Fig. 2. Centrifugal force distribution along the vehicle

It can be assumed that all the resistance forces act as a single force presented by vector F_r acting in the centre of the mass point. In case of monotonously accelerated motion an inertial resistance force and resistance of rotating masses will appear. Both of them are presented by vector F_i. Like force F_r both forces have the same direction and sense. To simplify the drawing the vehicle moves on the horizontal road. If resistance of the hill appears it will be represented by vector F_h acting in the centre of mass. Force created by driving system, does not matter if comes from front or rear axle (or from both axles), results in underloading of front the axle and overloading of the rear axle. These driving forces are presented by vector F_p. This vector is directed upward as the result of driving torque.

- During the move of the car on the curve, apart from all the forces mentioned above centrifugal force and its reactions appear. This force is represented in the Fig. 1 as vector F_c. Appearance of centrifugal force results in increased pressure of external wheels to the ground and decreased pressure of internal wheels. As internal wheels are understood as wheels located closer to the centre of the curve C_c. The limits of this phenomenon is slide or turn around of the vehicle. All the mentioned forces and springy suspension of the car lead to the situation in which front internal wheel loses contact with the ground. Such a situation is presented in Fig. 1. The turn around of the car happens exceptionally.

As the result of observation and forces analysis some conclusions were expressed:

- During accelerating on the curve the rear external wheel of the vehicle exerts the biggest pressure on the base thus this wheel is able to transfer the biggest driving force.
- During accelerating on the curve the front internal wheel of the vehicle exerts the smallest pressure on the base thus this wheel is able to transfer the smallest driving force.
- Taking into consideration statement a and b it seems to be desirable to transfer part of driving force from front internal wheel to the rear external wheel. This part is the force that can not be transferred to the base by front internal wheel.

Contemporary commercially manufactured transmission systems are not able to do so. In such a situation author of this paper elaborated new conception of driveline in which wheels are connected mechanically diagonally. This solution was patented in Poland in 2007 as "power transmission system for motor vehicle" patent number PL 194829. This new transmission system is called diagonal one.

2. Calculation of vertical load of the wheels

During the move on the curve with speed below the limit speed, the influence of centrifugal force results in change o chassis position. To establish changes of vertical load of the wheels the points of inclination of the chassis for front and rear axle should be determined. These points determine the pivot of chassis side declination. Methods of its determination are described in literature [1]. Draft of the vehicle moving on the curve of R radius is shown in the Fig. 4. This vehicle is equipped with suspension consisting of coil springs with determined turn pivot O_1-O_2 of chassis side declination. Both axles are stiff ones for reasoning simplification. The centre of the mass is located at height h' above the turn pivot of chassis side declination. The influence of centrifugal force leads to mass centre displacement and suspension springs deformation. Being based on professor's Manfred Mitschke argument, it is possible to calculate vertical forces and their reactions [1]. Centrifugal force distribution scheme along the vehicle is shown in Fig. 2.

Forces acting on the front and rear axles are shown in Fig. 3.

Difference between static and dynamic vertical load of the wheels of front and rear axle are:

$$\Delta Z_1 = C_n a\left(\frac{a_1}{L}\frac{h_{o1}}{b_1} + \frac{C_1}{C_1 + C_2 - C \cdot h'}\frac{h'}{b_1} + \frac{C_{o1}}{C_n}\frac{h_{sc1}}{b_1}\right) \tag{1}$$

$$\Delta Z_2 = C_n a\left(\frac{a_2}{L}\frac{h_{o2}}{b_2} + \frac{C_2}{C_1 + C_2 - C \cdot h'}\frac{h'}{b_1} + \frac{C_{o1}}{C_n}\frac{h_{sc2}}{b_2}\right) \tag{2}$$

For calculation of the example shown in Fig. 3 the following values were used:

a_1=a_2 - distance of front and rear axle from mass centre 1/2L [m]
$b_{1,2}$ - wheel track 1.2 [m]
C - vehicle weight 1080 [daN]
C_n - body weight 960 [daN]
C_{o1}, C_{o2} - front, rear axle weight 60 [daN]
h_{sc1}, h_{sc2} - front, rear axle centre of mass height 0.30 [m]
h_{o1}, h_{o2} - front, rear axle center of declination height 0.35 [m]
h' - centre of mass height from axle of declination 0.25 [m]
$Z_{1Z,1W}$ - load of front external, internal wheel [daN]
$Z_{2Z,2W}$ - load of rear external, internal wheel [daN]

after providing:

$$Z_{1Z} = Z_{2Z} = 1080/4 + 255 \cdot a = 270 + 255 \cdot a \text{ [daN]}$$
$$Z_{1W} = Z_{2W} = 1080/4 + 255 \cdot a = 270 - 255 \cdot a \text{ [daN]}$$

For certain relative centripetal acceleration $a = \frac{v^2}{Rg} [-]$

and values:

R - radius of the curve 40 [m]
v - speed of the vehicle 20 [m/s]

$$a = \frac{20^2}{40 \cdot 9{,}81} = 1{,}0193$$
$$Z_{1Z} = Z_{2Z} = 270 + 255 \cdot 1{,}0193 = 270 + 260 = 530 \text{ [daN]}$$
$$Z_{1W} = Z_{2W} = 270 + 255 \cdot 1{,}0193 = 270 - 260 = 10 \text{ [daN]}$$

Calculation results are presented in Fig. 4.

M. Mitschke reasoning is developed by applying driving torque influence. If driving torque on wheels appears (marked DT in Fig. 5) the additional force is lifting the front of the vehicle as the result of reacting torque RT which is shown in Fig. 5. Initially it is only torque influence, finally its torque and inertial force as the result of acceleration. If the reaction torque RT affects Z1W reaction with power bigger than 10 N it leads to loosing contact with ground by front internal wheel.

Fig. 6 shows the real rally car running and accelerating on the curve. Arrangement of the forces is very similar to the one presented in Fig. 5. In both cases influence of driving and inertial forces result in decreasing of vertical force affecting the front internal wheel and increasing the vertical force affecting the rear external wheel.

3. Diagonal transmission

The main feature of diagonal transmission is diagonal connection of the vehicle wheels as it is shown in Fig. 7. Such a design aims at optimization of driving force distribution. It can be applied for standard 4wd cars to improve safety and for special vehicles for better performance. Diagonal transmission has three differential gears, the central one and two lateral ones so called diagonal differential gears.

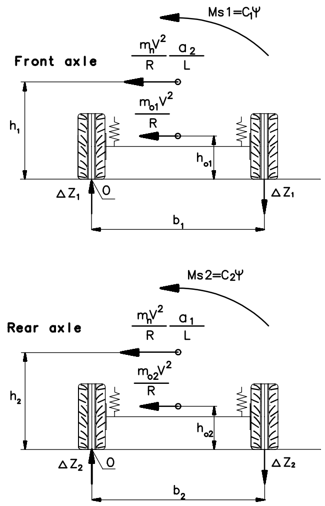

Fig. 3. Forces acting on the front and rear axle

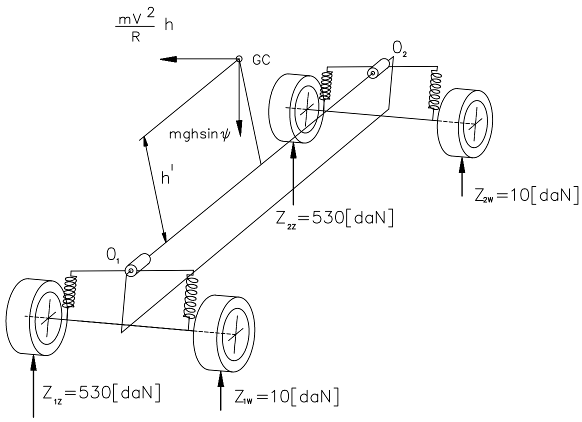

Fig. 4. Vertical reactions as the result of inertial forces

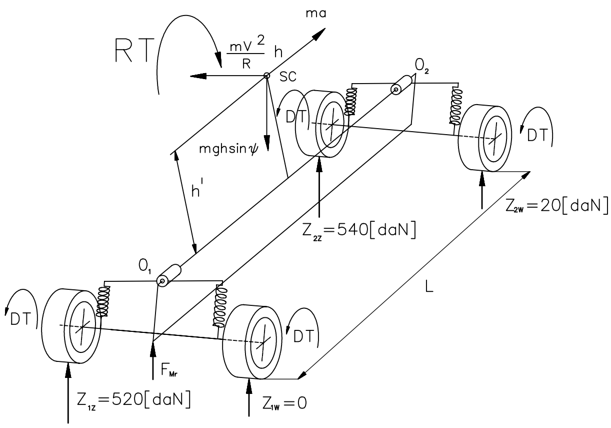

Fig. 5. Vertical reactions as the result of inertial forces and driving torque

Fig. 6. Phenomenon of losing contact with the ground by front internal wheel. Picture from collection Subaru World Rally Team, presented here with approval from Subaru Import Polska Sp. z o.o.

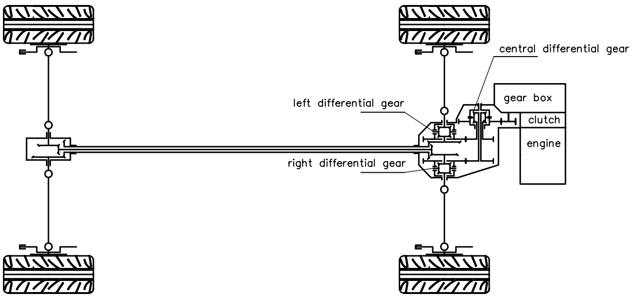

Fig. 7. Diagonal transmission

The way of power distribution optimization during accelerating on the curve is visible in Fig. 8. The front internal wheel is lifted up and contact with ground is lost.

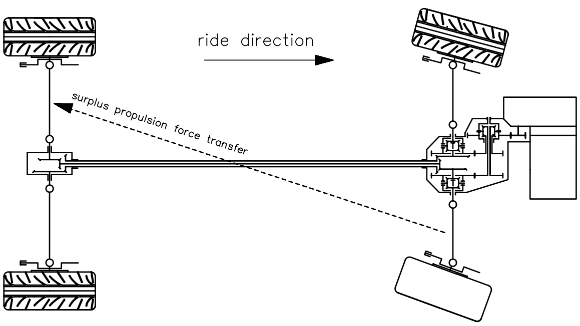

Fig. 8. Power distribution optimisation

In such situation control unit uses internal front wheel brake. The power distribution control unit can use ABS sensor and ABS control unit in a similar way as ASR system does. Thanks to the differential gear the surplus driving force can be transferred to the wheel that has better contact with the ground. This specific feature can be used for cooperation with ESP (ESP- means Electronic Stability Program. The main function is protecting vehicle from losing stability by breaking one of the front wheels to stop undesirable vehicle rotation) creating active ESP. The current generation of ESP systems use generally vehicles inertia for trajectory stabilization. With diagonal transmission system the engine power can be used for increasing safety. It means replacing passive by active system. This solution can be utilized in civilian and special vehicles.

4. Research

Virtual and physical tests were performed using Mechatronic Test Vehicle - MTV. Virtual model was created and tested in Unigraphics NX5 environment. Physical model was based on 2D technical documentation based on 3D model. The 3D model is visible in Fig. 9 in NX5 environment. Fig. 10 shows physical model placed on special frame that protects rubber wheels from deformation when idle. Both models were used for testing the case of vehicle acceleration on curve. [6,7]

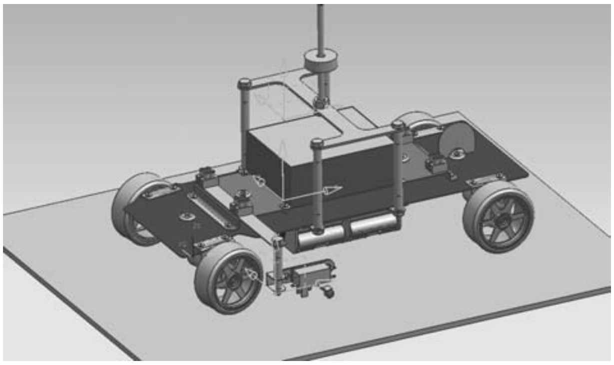

Fig. 9. The 3D model in NX5 environment

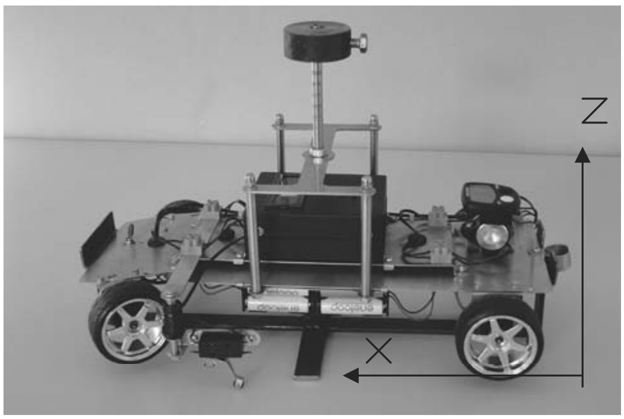

Fig. 10. MTV placed on the frame

4.1. Virtual tests and achieved results

Motion Simulation module enables MVT model motion simulation performed in following steps:

1. Creation of new simulation - New Simulation
2. Vehicle parts definition - Links
3. Correlation between parts definition - Joints
4. Contact with the ground definition - 3D Contact
5. Driving definition - Driver
6. Control function definition - STEP
7. Starting simulation
8. Results presentation

Among listed points function STEP needs a comment. This function has following syntaxes: STEP (x, x_0, h_0, x_1, h_1)

x - independent variable, the most typical is time,

x_0 - real value defining when x parameter should start,

h_0 - initial value of STEP function

x_1 - real value defining when x parameter should finish

h_1 - final value of STEP function

Mathematical description of STEP function is expressed by formula (3) [3]:

$$a = h_1 - h_0$$

$$\Delta = (x - x_0)/(x_1 - x_0)$$

$$STEP = \begin{Bmatrix} h_0 : x \le x \\ h_0 + a \cdot \Delta^2 (3 - 2\Delta) : x_0 < x < x_1 \\ h_1 : x \ge x_1 \end{Bmatrix} \tag{3}$$

Performed virtual tests resulted in drawings of vehicle trajectory, speed diagrams for particular wheels and tables of centre of gravity speed. The trajectory of the vehicle with diagonal transmission is visible in Fig. 11 [2]. The theoretical radiuses without drift are shown in blue.

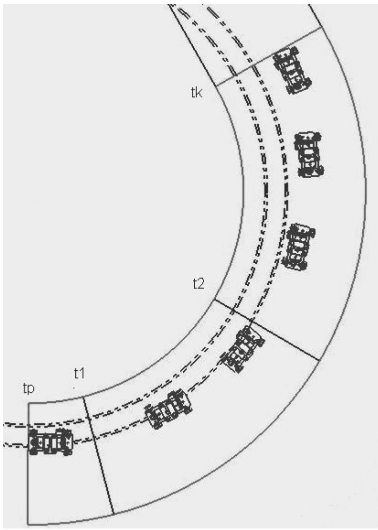

Fig. 11. The trajectory of the vehicle

In comparative test the diagonal transmission system achieved a little bit shorter time. Results are presented in Table 1.

Table 1.
Times of passing through the curve of two different driving systems [2]

Driving system	Time [s]
Diagonal transmission	1,200
Classic 4x4 transmission	1,220

4.2. Physical tests and achieved results

MVT is equipped with control unit for programmable start-stop management. Measuring system enables to register the maximal speed, time of ride, average speed and ride distance. For comparative analysis electrical power transmission system assures switch to classic or to diagonal power distribution. In addition there is a possibility to shift centre of mass along OX and OZ axes (Fig. 10.). The ride trajectory was drawn on the base with felt-tip pen. The drawing equipment is visible in Fig. 12.

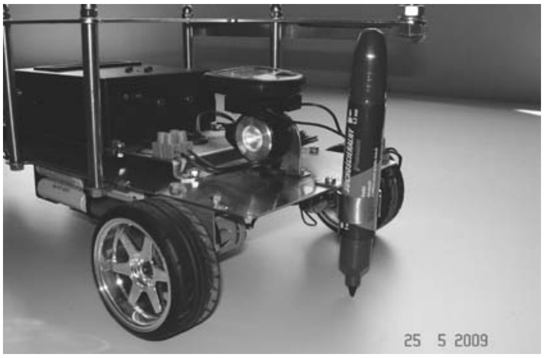

Fig. 12.The drawing equipment

Performed tests resulted in drawn curves for both types of transmission systems. These curves were digitalized (with laser range finder placed on angle gauge dividing head) and transferred to spread sheet and to AutoCAD. Sample curves are presented in Fig. 13.

The average maximal speed taken from four rides is presented in Table 2. Before average calculations the Dixon (*Dixon method according to* ASTM Designation: E 178 - 02) method was used for removing the outstanding values. Better result for diagonal transmission was achieved only on slippery floor. At the rough surface the classic transmission was faster.

Table 2.
Average maximal speeds at slippery surface

Classical transmission system Average maximal speed [km/h]	Diagonal transmission system Average maximal speed [km/h]
7.69	8.62

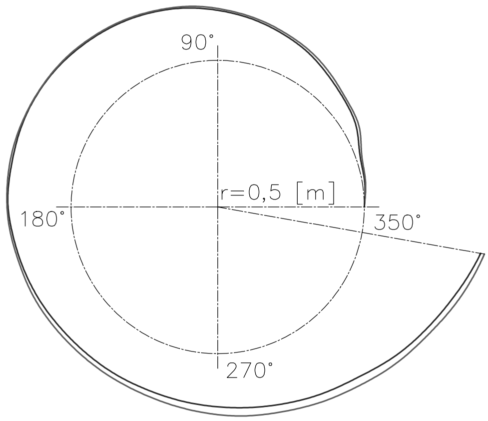

Fig. 13. The curves representing MTV trajectory, blue line represents diagonal transmission and red one the classical drive line

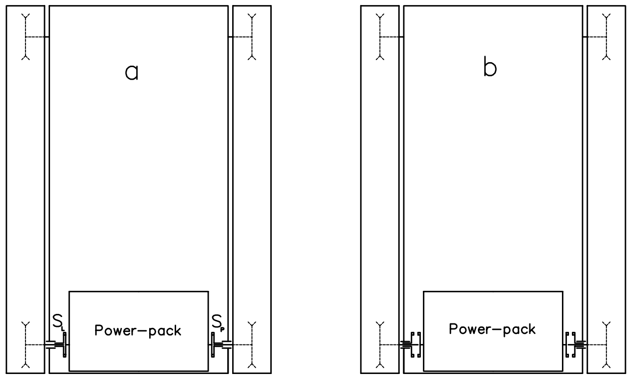

Fig. 14. S_L and S_P are dog clutches for engagement (a) and disengagement of power-pack with sprocket (b)

5. Special vehicle with diagonal transmission as power-pack

An important practical aspect apart from power distribution optimization is using diagonal transmission system in power-pack configuration for special vehicles. Power-pack is a standard solution for tanks. This conception is based on philosophy of very fast replacement of the whole transmission system to bring tank back as soon as possible to the battle field. The power-pack repair should be performed out of battle field in specialized work shop. Such a conception seems to be logical.

The power-pack conception was applied in caterpillar vehicles thanks to their specific design in which all driving system elements are located in one place in front or in the back of the tank. The idea of power-pack design is visible in Fig. 14 [4].

Power-pack consists of engine with cooling system, gear box and steering mechanism, wiring and all the PTO's. All the mentioned systems are integrated together on auxiliary frame. After disconnecting the wiring and disengagement dog clutches and unscrewing bolts connecting the auxiliary frame with tank body the whole power-pack can be taken off by means of crane from tank. When it comes to wheeled special vehicle the power-pack idea is much more difficult in application because of scattered transmission units are placed far from the engine. In case of diagonal transmission all important units are concentrated close to the engine. Such a solution enables to realize power-pack conception much easier. The only not integrated units are two simple bevel gears necessary for driving rear wheels. These two simple gears left far from engine are similar to the two sprockets left far from power-pack to keep caterpillar tighten in tank technology. The special wheeled vehicle equipped with diagonal transmission in power-pack configuration is visible in Fig. 15.

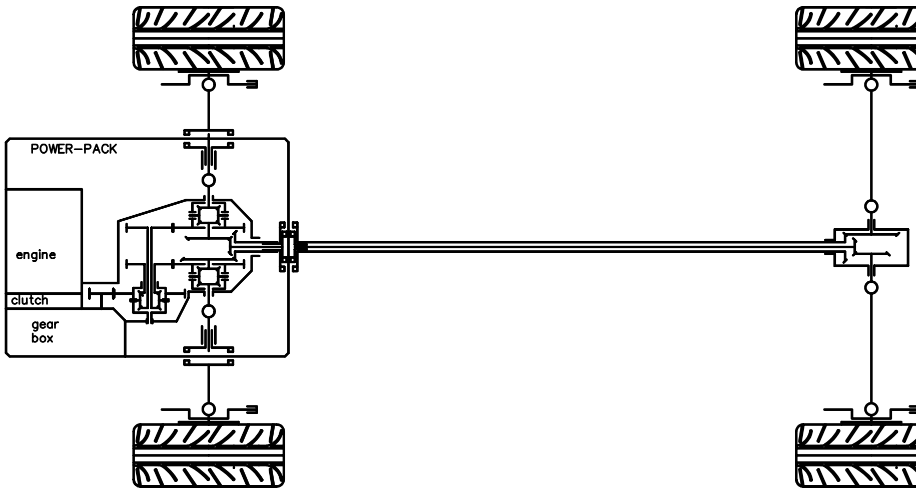

Fig. 15. Conception of diagonal transmission in power-pack configuration

6. Conclusions

Theoretical analysis of vehicle wheels vertical load during accelerating on the curve indicates that external rear wheel bears the biggest vertical force. The front internal wheel is loaded with the smallest vertical load. The classic 4wd transmission system is not able to transfer the surplus driving force from unloaded front wheel to the overloaded rear wheel which is placed on diagonal side of the vehicle. Such a possibility is the main feature of diagonal transmission system. Initial virtual model and physical tests confirm the potential of the new method of power distribution. In some test conditions higher maximal speed for the diagonal transmission was observed (Tables 1 and 2).

There is a possibility of diagonal transmission system application for sport vehicles to improve performance. For utilitarian vehicles increasing safety is the most important aspect, especially in cooperation with ESP system. When it comes to special and military vehicles both mentioned above features are desired. Possible power-pack application is an additional advantage.

Described research is a very first attempt of testing the solution that was created as the result of observation and intuition. Diagonal transmission needs further research in virtual reality as well as physical tests to prepare recommendation for design to continue tests in real vehicle.

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