The Air Suspension Synthesis for the Single-axle Transport and Technological Module

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Abstract. The present paper contains a theoretical justification of composition (assembly of elements or structure) and design of an air suspension of a single-axle transport and technological module. Application of mathematical simulation proves the feasibility of using an independent suspension with air bellows, but without any antivibration devices (shock absorbers). The feasibility concerns the transport and technological module of the proposed design.

1. Preface
The aim of this article is a theoretical justification for the synthesis of a suspension for a transport and technological module (TTM) designed to transport the various packaged cargoes, including 20-feet equivalent units of general purpose (ISO 1496/1).

In order to achieve this, the following problems shall be solved:
- examine possible design variants of suspensions taking into account a transport and technological module’s operating features,
- estimate loads on springing elements and some choose springing elements taking into account the existing market offers,
- make calculations of dynamic characteristics of TTM shock absorbers on the basis of a specified value of the damping ratio, estimate their installation’s advisability taking into account their actual rate of motion.

2. Main Part

2.1. Review of existing solutions
A wheeled vehicle suspension performs the following functions:
- connecting the wheels or rigid axles with a transport vehicle load-carrying system – with a body or a frame,
- transferring the wheel-road interaction forces and moments to the load-carrying system,
- providing the required type of the wheels’ movement relative to a body or a frame, and the required smoothness of movement.

Suspension shall have a reasonable arrangement, and small mass of component parts, particularly the unsprung components, and provide:
- secure road-wheel contact,
- required smoothness of movement,
- required transport facility stability and steerability,
- appropriate kinematics of steering wheels in case of their vertical longitudinal movement.

In the general case, a transport facility suspension consists of springing, guiding and damping elements.

Thus, springs, torsion bars or air bellows; trailing, crossbar or semitrailing arms as guiding elements, hydraulic shock absorbers for vibration damping are used in a freight vehicle suspension. Sometimes one element can serve several functions, for example, a multi-leaf spring serves a function of a springing element, and limits horizontal displacement of suspension elements and damp vibrations.

As required, some additional requirements may be imposed to the suspension, so the following shall be provided:
- adjustment of a transport facility’s body (platform) position and ground clearance leveling,
- variability of elasticity characteristic in order to improve the service properties.

Thus, in Kamateyner project [1] of JSC KAMAZ, KAMAZ-65117 chassis has front and rear air suspensions (Fig. 1), an electronic system of layer management, a special top frame with container twistlocks (as per ISO 1161 Standard), and guide rollers for easy installation and removal of the topsides. In general it makes possible to carry out transportation and quick replacement of multi-purpose demountable bodies and 20-feet equivalent units, and increases transportation profitability.

![Figure 1. Options of the suspension air bellows](image)
a) general view of KAMAZ-65117 chassis rear suspension with shock absorbers and antiroll bars at each axle,  
b) body replacing process due to height variation of the chassis loading surface with an air suspension

To transfer various cargoes, auto and electric forklifts are widely used in storehouses, at terminals and assembly factories. On the basis of operating conditions (mainly indoor sites with smooth hard surface, low rates of motion), the suspension, in most cases, has a simple design: a loader rigid axle is fixed directly to a frame (half-frame), a relatively rigid cast or semipneumatic tires provide a low level of deformation under load, so it increases transport facility stability and decreases rolling power consumption. For the loaders used at open sites with unprepared surface (e.g., building sites), one of the axles, in most cases, is of the balance-beam type, and tires are of the wide cross-section pneumatic type (Fig. 2).

![Figure 2. A running gear of a front loader](image)
1 — a wheel, 2 — a front axle, 3 — a back axle in a balance-beam frame

Thus, in many cases, the loaders’ operating conditions allow using the conventional suspensions simple in design, where the functions both of a springing element, and a damping element are provided by a tire.
However, even when transport facilities have a low rate of motion, it may be necessary of leveling the ground clearance and independent vertical movement of the wheels. Fig. 3 shows a S10 self-propelled transfer platform manufactured by Scheuerle company, designed for transportation of indivisible heavy-weight cargoes. An independent coupled wheel suspension, and cargo platform height adjustment in the range of ±325 mm by means of the independent suspension, allows equally distributing the rolling load and wheel load; keeping the height value of center of gravity of the cargo, when getting over road unevenness, which is important for the transport facility stability; decreasing the loading height during loading operations.

2.2. Specific features of the proposed transport and technological module

The concerned TTM (Fig. 4) consists of a running gear 1 and a lifting device 2, connected between each other by a rolling circle 3. Loading is carried out in the following way: the module drives to a cargo, then goes down using its springing elements to provide the contact of the device stand wheels with a road surface, and then the cargo is captured, lifted and lowered upon the lifting device platform. After the loading, the module shall be lifted on the suspension’s springing elements to provide losing the stand wheels' contact with the road surface, and making the cargo transportation possible. Taking into account the necessity of lifting and lowering, the lifting device’s springing element shall be of air type.

When choosing a suspension gearing diagram, one shall take into account, that the module is driven by means of two electric motors, and torque from electric motors to the wheels is transmitted by means of on-board gearboxes. The simplest design and technological variant of a suspension is appeared to be
a single wishbone with a trailing arm and an air bellows. Herewith, both a reduction unit, and an electric motor shall be fixed at every suspension arm.

TTM suspension gearing diagram is shown on Fig. 5. A springing element is installed in such a way to minimize its lateral deformation. When a wheel travels by an amount equal to \( h \) with respect to a frame, then the springing element axle deviates for the angle \( \alpha \). This deviation at the arm’s dimensions \( l_1 = 440 \text{ mm} \) and \( l_2 = 800 \text{ mm} \) (Fig. 5) in case of the wheel’s travelling for \( h = 100 \text{ mm} \) will be \( \alpha = 2^\circ \).

![Figure 5. Suspension gearing diagram.](image)

2.3. Modeling

General view of a suspension is shown on Fig. 6. The suspension consists of two trailing arms 1, each of them is connected with a frame by means of two metal-rubber mountings 2, it allows eliminating cross motions of the arm. The arm is hollow; an arm case is designed for installation of a clutch to connect an electric motor 4 with an on-board gearbox 3. The arms are connected between each other by means of a cross tie rod 5, which is connected with the arms by means of metal-rubber mountings 6. In this case, the cross tie rod’s function is to provide additional keeping the arms off the cross motions. The design of metal-rubber mountings cross tie rod allows compensating the motions about 10 mm, occurring during operation of the suspension. The air bellows 7 are located at the arm ends. If considered necessary to increase the module lifting capacity by means of installation of brackets 8 of other configuration, a number of springing elements may be increased from two up to four.

![Figure 6. General view of a suspension.](image)

As a result of TTM three-dimensional simulation, it has been determined that unsprung weight is about 3.5 kN, sprung weight in running order is 85 kN, sprung weight under gross weight is 565 kN. Taking into account the lengths of the arms, each springing element load is: 23.4 kN for unladen weight, 156 kN under gross weight.

One of the most important operating characteristics of a wheeled vehicle suspension is the damping ratio. Exactly the damping ratio value affects the smoothness of movement of a transport facility. It is over the range of 0.25 – 0.3 for production cars. The damping ratio value depends on many design parameters of a transport facility suspension, but most of all it is affected by tension and compression resistance forces of hydraulic shock absorbers.
The shock absorber resistance coefficient was considered in the works by authors [7] R.A. Akopjan, G.K. Mirzoev, A.V. Ermolin and others [4-7] as a constant value. It does not allow estimating the operability of the shock absorber’s structural members affecting the parameters of its dynamic characteristics at valve and throttle sections. Also such assumption does not allow considering nonsymmetry and nonlinearity of dynamic characteristics of automobile shock absorbers in rebound and compression stroke modes.

Therefore, in order to calculate dynamic characteristics of the module shock absorbers, let’s use a mathematical model describing the operation of the suspension’s single-mass vibrating design diagram in the module’s general vibrating system [8].

To be able to take into account the asymmetry and nonlinearity of force and speed characteristics of the shock absorbers, the following equation is used:

$$m \frac{d^2y(t)}{dt^2} + \mu \frac{dy(t)}{dt} + cy(t) = \mu \frac{dq(t)}{dt} + cq(t),$$

where $m$ – sprung mass of the module, kg,
$y(t)$ – amount of movement of sprung mass, m,
$\mu$ – shock absorber resistance coefficient, Ns/m,
$c$ – suspension ruggedness, N/m,
$q(t)$ – perturbation caused by a road surface.

The value of $\mu$ in the equation (1) is nonlinear and depends on tension and compression resistance forces, and the shock absorber’s piston travel speed.

An example of variation of the shock absorber resistance coefficient with piston travel speed is presented on Fig. 7.

![Figure 7. Spline interpolation for values of $\mu$ depending on $V$.](image)

Having determined the value of $\mu$ from the equation (1), it is possible to get the shock absorber dynamic characteristics in the generally accepted form: the relation “force at the shock-absorber rod vs. rod travel speed”.

It is possible to solve a non-homogeneous differential equation of the second order (1) by means of numerical methods, e.g. Runge - Kutta method.
The determined values of sprung mass acceleration and unsprung mass acceleration (Fig. 8) are used to plot $R_x(\tau)$ correlation function [9].

![Figure 8. Acceleration values of sprung and unsprung mass.](image)

Acceleration spectral density values of sprung and unsprung mass are determined using $R_x(\tau)$ correlation function

$$S_x(\omega) = \int_{-\infty}^{\infty} R_x(\tau) e^{-i \omega \tau} d\tau = 2 \int_{0}^{\infty} R_x(\tau) \cos(\omega \tau) d\tau,$$

where $\tau = t_1 - t_2$; $t_1$, $t_2$ are time gap boundaries (Fig. 9) [10].

![Figure 9. An example of an oscillogram.](image)

Having divided the vertical acceleration spectral density values of transport facility sprung mass by the vertical acceleration spectral density values of unsprung mass, we obtain the damping ratio.

Let’s solve an inverse problem to determine the dynamic characteristics of the shock absorbers. Taking the damping ratio of 0.3 and the vertical acceleration spectral density values of unsprung mass (Fig. 10), let’s determine the vertical acceleration spectral density values of sprung mass (Fig. 11) under the specified driving conditions (off-the-highway uniform motion at a speed of 10 km/h).
Figure 10. Vertical acceleration spectral density values of the module unsprung mass when moving off-the-highway at a speed of 10 km/h.

Figure 11. Vertical acceleration spectral density values of the module sprung mass when moving off-the-highway at a speed of 10 km/h.

Having determined the vertical acceleration spectral density values of sprung mass, and solved the equation (1), let’s obtain the values of $\mu$ depending on travel speed of the shock absorber piston (Fig. 12).

Figure 12. The shock absorber resistance coefficient values depending on its piston travel speed.

Let’s transform the obtained values presented as a diagram into conventional force and speed characteristics of the shock absorber (Fig. 13).

Figure 13. The required dynamic characteristics of the shock absorber.
The obtained dynamic characteristics show that maximum generated force at the shock-absorber rod does not exceed 25 N, which is comparable to friction forces in the module suspension’s metal-rubber mounting. Therefore, it is not required to install the shock absorber on the module.

3. Conclusion

TTM design solid modeling, as well as numerical simulation of its suspension operation allowed theorizing the design solutions adopted for the air suspension synthesis. The experimental check of the obtained results is supposed using a TTM scale solid model.

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