Modeling of Autovariator Operation as Power Components Adjuster in Adaptive Machine Drives

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Abstract. Full application of the available power and stationary mode preservation for the power station (engine) operation of the transport machine under the conditions of variable external loading, are topical issues. The issues solution is possible by means of mechanical drives with the autovariated rate transfer function and nonholonomic constraint of the main driving mediums. Additional to the main motion, controlled motion of the driving mediums is formed by a variable part of the transformed power flow and is implemented by the integrated control loop, functioning only on the basis of the laws of motion. The mathematical model of the mechanical autovariator operation is developed using Gibbs function, acceleration energy; the study results are presented; on their basis, the design calculations of the autovariator driving mediums and constraints, including its automatic control loop, are possible.

1. Introduction

As it is known [1], one of the significant and achievable objectives for adapting actual mechanical machines systems is the automatic control of the power components of the energy flow from the engine to the machine operating element. This is achieved by means of a targeted variation in the transfer function of the mechanical system rate, which provides the necessary power operation mode or engine stationary operation at an economy mode under conditions of variable external force loading of the operating element.

In machines with an operator (manual) control, the variation of the transfer function is made by an operator, performing a step switching motion of the transmitted power flow through various chains of the kinematic circuit. It is significantly less often when the mechanical system includes a motion variator converter which allows the operator to vary the kinematic dimensions of the driving mediums smoothly, without breaking the force flow, and, as a consequence, the transfer function of the mechanical part of the machine drive.

Recently, especially in the transport engineering, automatic transmissions are serially produced and widely used, their availability improving the machines performance, simplifying the control under multimode operation typical for these machines.

Automatic transmissions, as a rule, include a torque converter, a multi-stage gearbox of an planetary circuit, multi-disk shift units, power and command element of hydraulics and electronics. In general,
this is a complex and expensive technology, requiring considerable power produced by the power plant for its operation.

The variator circuits of mechanical drives are structurally simpler, despite the fact that in order to control the drive transfer function and normal forces level in frictional contacts, hydraulics is used allowing to produce considerable forces in control loops by means of the variator transfer function, i.e. a special power source integrated into the control loop is used in the designs [2–6].

Successful implementations of V-belt variators schemes with sliding pulleys and hydraulic control in the running transmissions of domestic combines, in drives of threshing drums, in reel drives are known. Despite the fact that the fundamental issues of creating V-belt variators have been solved, the problem of providing the operational life for the V-belt remains an issue, since the conditions of its operation are extremely unfavorable, so the synthetic or split-type belt is the resource-determining element of the variator design and the variators with such a belt are realized mainly in low-power machinery only: motor scooters, snowmobiles, mini-cars for inter-quarter trips, etc.

Recently, Honda, Nissan and Audi automakers have developed an equal-resource split-type V-belt, with the main aggregates, containing trapezoidal steel elements connected into a loop by means of a multilayer steel strip. The belt allows the level of normal forces up to 104 N and is able to transmit torque up to 300 Nm, which makes the variator drive promising for a wide range of transport machines.

2. Research objective

The development of mechanical variator and autovariator drives is prevented by the absence of models for the dynamic operation of the reduced masses on either sides of the two-degree-of-freedom nonholonomic constraint of the main driving mediums.

The problem of variator operation modeling is set and solved, the operation being based solely on the use of the laws of motion. The controlling means for transfer function is the targeted driving mediums motion, additional to the main one, by means of an integrated control loop that uses the variable power component of the main transformable power flow, i.e., the proposed design does not have a special source of energy. Such a variator is able to make a significant evolution to achieve the energy perfection of the drive and auto-control by means of the transformed power components.

Similar structures of motion transducers are called autovariators, numerous technical solutions of autovariators, for example [7–8], are obtained on the basis of the mechanical systems designing principle based on the property of the designed system to adapt to the systems real parameters, to operating modes.

The modeling of the autovariator operation is of current concern. It allows to get information on the driving mediums and constraints load at an early stage of the preliminary design, which is necessary for the design calculations of all the autovariator drive components, and the integrated control loop in particular.

3. Theory

The main and current task for transport machines is the full utilization of the available power by maintaining the stationary mode of operation of the power plant (engine) under conditions of variable external loading.

$M_1$ and $M_2$ are the power moments and shafts rates on the engine and the operating elements of the machine (the driving wheel of the transport machine), respectively; $\eta$ being mechanical efficiency of the drive, the variation ratio equals:

$$M_1\omega_1\eta = M_2\omega_2,$$

(1)
\[ \omega_2 = \frac{M_1 \omega_1 \eta}{M_2}. \]  

(2)

Assuming that \( M_1 = \text{const} \) and \( \omega_1 = \text{const} \) and, in the first approximation, \( \eta = \text{const} \), a hyperbolic constraint \( \omega_2 \) is obtained with the variable \( M_2 \), but the transfer function \( \Pi \omega \) of the autovariator

\[ \Pi \omega = U_{1,2} = \frac{\omega_1}{\omega_2} \]

is linear with respect to \( M_2 \); that is, in maintaining the task conditions, a multiple variation \( M_2 \) requires a multiple variation \( \Pi \), and this circumstance is decisive in the synthesis of the parameters of the integrated control loop of the autovariator.

We compose the autovariator dynamic model, taking the differential equation of the coordinates \( \varphi_1 \) and \( \varphi_2 \) derivatives ratio of the main driving mediums in its basis. To be definite, \( r \) and \( \rho \) are assumed the kinematic dimensions for the main driving mediums, and the result is

\[ r \dot{\varphi}_1 = \rho \dot{\varphi}_2. \]  

(3)

Despite the formal division of variables, equation (3) is not integrated due to the time uncertainty of the variable \( \rho \).

A proper consideration of the mutual effect on the motion of the kinematic chains located on different sides of the nonholonomic constraint is given by a model constructed on the Appell equations using the Gibbs function, the accelerations energy:

\[ s = \sum_{i=1}^{n} \frac{J_{r1} \ddot{\varphi}_i^2}{2}, \]  

(4)

Where \( J_{r1} \) is the inertial characteristic of the kinematic chain reduced to \( \varphi_1 \), and the specific partial derivative of the Gibbs function

\[ \frac{ds}{d\dot{\varphi}_1} = J r \dot{\varphi}_1 = M_1 \]  

(5)

Where \( M_1 \) is the force parameter, reduced to the generalized coordinate, from external forces. The acceleration energy for a two-shaft variator, referring to the two reduction links associated with shafts 1 and 2, equals:

\[ s = \sum_{i=1}^{n_1} \frac{J_1 \ddot{\varphi}_1^2}{2} + \sum_{i=1}^{n_2} \frac{J_2 \ddot{\varphi}_2^2}{2}, \]  

(6)

Where \( J_1 \) and \( J_2 \) are reduced to shafts 1 and 2, the inertial characteristics of the kinematic chains, divided by a nonholonomic constraint. After the transformations of [1], the differential equation of motion of the shaft 1 is obtained:

\[ (J_1 + J_2 U_{2,1}^2) \ddot{\varphi}_1 + J_2 U_{2,1} \dot{U}_{2,1} \dot{\varphi}_1 = M_1 + M_2 U_{2,1} \]  

(7)

for shaft 2

\[ (J_2 + J_1 U_{1,2}^2) \ddot{\varphi}_2 + J_1 U_{1,2} \dot{U}_{1,2} \dot{\varphi}_2 = M_1 U_{1,2} + M_2 \]  

(8)
Referring to the features of the technical specification for the autovariator design for transport machine drive, namely, when \( M_1 = \text{const} \) and \( \dot{\Phi}_1 = \omega_1 = \text{const} \) on coordinate \( \Phi_1 \) is preserved, equation (7) is simplified to the following form:

\[
J_2 U_{2,1} \dot{U}_{2,1} \Phi_1 = M_1 + M_2 U_{2,1}
\]

or

\[
\dot{U}_{2,1} = \frac{M_1 / U_{2,1} + M_2}{J_2 \Phi_1}
\]

As the transfer function \( U_{2,1} \) is linear with respect to \( M_2 \), it can be represented as \( U_{2,1} = k M_2 \) and \( M_2 \) remains the only variable in (10), and if the variation law of \( M_2 \) is known in time, then (10) is integrated, thus the problem with of the output autovariator chain motion can be solved in quadratures.

4. Discussion

The ratio of the variation \( U_{2,1} = U_{2,1}(M_2) \) is the initial one for designing of the integrated autovariator control loop. \( \triangleq \) in the numerator (10) means the algebraic addition of multidirected reduced moments \( M_1 \) and \( M_2 \) and the numerator’s vanishing signifies the end of the transient process, the motion becomes steady with the transfer function \( U_{2,1} = \text{const} \). The study of the motion model on (7) and (8) shows that the operation of the main driving mediums is affected not only by the transfer function, but also by the rate of its variation. Moreover, this effect has a damping character for the transmission of transport machines, i.e. the more sharply the transfer function changes, the less controllable the rate of the chain system is.

As in (10), the numerator is the difference in the power characteristic before and after the nonholonomic constraint, i.e., actually, this difference is an excessive moment, at \( M_1 = \text{const} \), being completely determined by the variables \( M_2(t) \) or

\[
M = \frac{\int M_2(t) dt}{U_{2,1}} + M_2(t) = M_{exc}
\]

The excessive moment can be represented by the time integrable function, and then \( \dot{U}_{2,1} \) in (10) has a finite analytic definition.

Several variants \( M_{exc} = M_{exc}(t) \) are considered.

\[
\frac{dU_{2,1}}{dt} = \frac{M_{exc}(t)}{J_2 \Phi_1}
\]

or

\[
dU_{2,1} = \frac{1}{J_2 \Phi_1} M_{exc}(t) dt
\]

The result is

\[
U_{2,1} = \frac{1}{J_2 \Phi_1} \int_{t_0}^{t_1} M_{exc}(t) dt = \frac{M_{exc}(t)}{J_2 \Phi_1}(t_1 - t_0).
\]

At \( t_0 = 0 \), \( U_{2,1} = \frac{M_{exc}(t)}{J_2 \Phi_1} \), the latter means a linear variation \( U_{2,1} \) from time at constant \( M_{exc}(t) \), and rate variation \( U_{2,1} \) significantly depends on inertia \( J_2 \) of the driven chain.
The variant of linear increase or decrease of $M_{exc}(t)$, determined by the value $k$, is quite general.

This variant has an independent value and also it can be used as a linearized representation of $M_{exc}(t)$ on the studied interval for any function $M_{exc}(t)$. If $M_{exc}=kt$, where $k=const$, then:

$$
U_{2,2} = \frac{1}{J_2 \Phi_1} k t dt \quad \text{and} \quad U_{2,1} = \frac{k}{J_2 \Phi_1} \frac{t^2}{2},
$$

i.e. the variation $U_{2,1}$ at a linear variation $M_{exc}(t)$ depends on time in the second power and depends additionally on the variation rate $M_{exc}(t)$, determined by the value $k$.

The most general case is the analytic representation of $M_{exc}(t)$ by a polynom of any power or by a trigonometric function; in any case, under the given conditions, it is possible to obtain the analytic expression $U_{2,1}$ in the function $M_{exc}(t)$.

The results of mathematical modeling in the Excel program are given below. The input motion is carried out with a constant angular rate $\omega_1 = \frac{10}{\varepsilon}$ and the nominal value of the inertia moment $J_1=0.5\,kgm^2$ applied to this shaft. The ratio of the autovaristor's correlation to the inertia of the driven chain $J_2$ is studied for two values $J_2=1\,kgm^2$ and $J_2=4\,kgm^2$.

Several ratio variants of the variation in the excess force moment $M_{exc}(t)$ are examined. $M_{exc}(t)=const=5\,Nm$ is defined. The variation $U_{2,1}(t)$ is presented in Figure 1.

When $M_{exc}(t)$ varies according to a linear law, namely $M_{exc}(t)=kt$ with the accepted values $J_2$, the variation in $U_{2,1}(t)$ occurs, as shown in Figure 2.

The effect of the slope $k$ of the linear characteristic $M_{exc}(t) = kt$ at $k=1$ and $k=4$ at constant $J_2=2\,kgm^2$ is shown in Figure 3.

5. Summary and conclusion

The results analysis shows that an increase in inertia of the driven chain results in a decrease in the variation rate of the transferred ratio $U_{2,1}$, and an increase in the steepness $k$ of the linear characteristic $M_{exc}(t)=kt$ increases the variation rate $U_{2,1}$, which is physically reasonable. The obtained ratio can be used as the initial basis for constructing driving mediums and constraints of a mechanical autovaristor, especially this refers to the synthesis of the integrated control loop with the transferred ratio of this adaptive regulator by power components in the drive of transport machines.

![Figure 1. Variation $U_{2,1}(t)$ at $M_{exc}(t)=const$ depending on $J_2$.](image)
Figure 2. Variation $U_{21}(t)$ at $M_{exc}(t)=kt$ depending on $J_2$.

Figure 3. Variation $U_{21}(t)$ at $M_{exc}(t)=kt$ depending on $k$.

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