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ROLE OF POWER RESERVE IN PARALLEL HYDRAULIC MOVEMENTS OF AIRCRAFT CONTROLS

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Abstract

A parallel booster control of transport aircraft controls has been modeled and the positive role of the power reserve for the operation of the structure with permissible operating differences in the pressures of the supply systems has been demonstrated.

Introduction

Parallel hydraulic drives are typical of large transport aircraft, in which one body (for example, a rudder for height or direction) deviates from several hydraulic boosters. Each of these hydraulic boosters is powered by a separate hydraulic system and, of course, there are acceptable differences in the operation of the drive devices. The control unit that drives these devices is the element through which, in real conditions, the common work is redistributed between the individual hydraulic boosters. Some (faster) work with overload, and others (slower) work with supporting efforts. In the vicinity of the end positions, there are significant differences in the strokes of the individual devices. The theory of hydraulic propulsion of aircraft controls recommends that hydraulic boosters be selected with a 50% power reserve and a 20% speed margin, with possible desynchronization within acceptable limits. In practice, such a stock of all modes and structures cannot always be provided, especially for controls such as movable stabilizers for maneuverable supersonic aircraft. In these organs, some limitations in their efficiency are possible (for example, limitation of the range of deviation due to power causes of supersonic speeds) all shunting aircraft. During the flight tests of the MiG-25 aircraft, there is a known case in which it is impossible to bring a differentially deflected horizontal stabilizer out of the extreme position, and the situation ends in a crash. Events for this reason in flight practice are called “biting the control lever”.

Study area

The research idea is to check by modeling the efficiency of the power reserve of the hydraulic boosters, with the allowable for the practice difference in the pressure of the supply hydraulic systems of 10% ... 15%. For transport aircraft, this should ensure that the “stop” phenomenon is eliminated before the rudder reaches the end of the available range due to high hinge torque. “Biting the controls in extreme positions” for the handlebars is impossible because the regulations do not allow them to be overcompensated, as in the case of controllable stabilizers. Possible rudder overcompensation can occur due to incorrect adjustment and operation of servo compensators.

Model of hydraulic booster

The developed computer model of parallel drive of controls from three steering units aims to study the work in case of differences in the pressure of the working fluid in the feed hydraulic systems, leading to synchronization disorders. This type of propulsion is provided by the spoilers, the rudder of the A-320 aircraft and the controls of most heavy transport aircraft.

Fig. 1 shows a model of a steering unit, implemented through the software package “Matlab-Simulink”. Its operation is the basis of the parallel multi-booster drive. Fig. 2 shows the same unfolded model for the “BOOSTER” subsystem. The load is simulated by the product of four multipliers (Fig. 2): spring stiffness C = 2330 N/mm, stroke of the actuator- *y(mm) = f(t)*, coefficient K_p and additional multiplier (*±1*) , formed by a unit for conditional transition on a signal from the distribution valve.

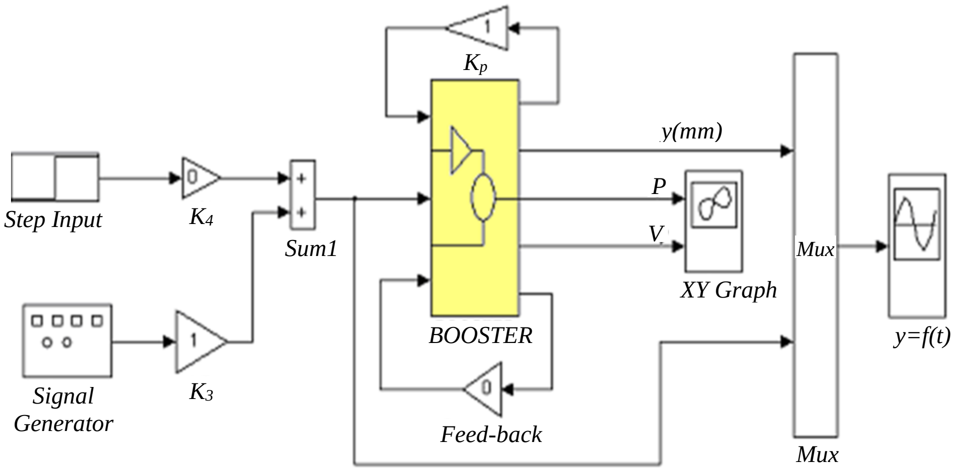

Fig. 1. Model of steering unit, realized through software package “Matlab-Simulink”

The input signals (Step input, Signal Generator) in the model are formed from standard Simulink sources, by appropriate adjustment. The results are visualized graphically as a transient process of displacement *y* = *f(t)* and the dependence of the speed of movement on the load *V* = *f(P)*. By changing the value and the sign in the amplifier K_p, the degree of load and its direction with respect to the movement of the actuator is simulated. At $K_p > 0$ the load is counteracting. If $K_p < 0$ the power steering model mimics an auxiliary load. The operation of the rudders with axial compensation below 28% is characterized by the fact that in case of deviation from the neutral position the counteracting force increases according to an approximately linear law. Returning to a neutral position of such a governing body is a supportive effort, decreasing from a maximum value to zero. Cyclic deflection of the rudder in the whole range is characterized by successive phases of operation with counteracting and supporting load on the power steering. The change of the nature of the load in the end positions in the model is performed automatically by the *Switch1* unit (Fig. 2).

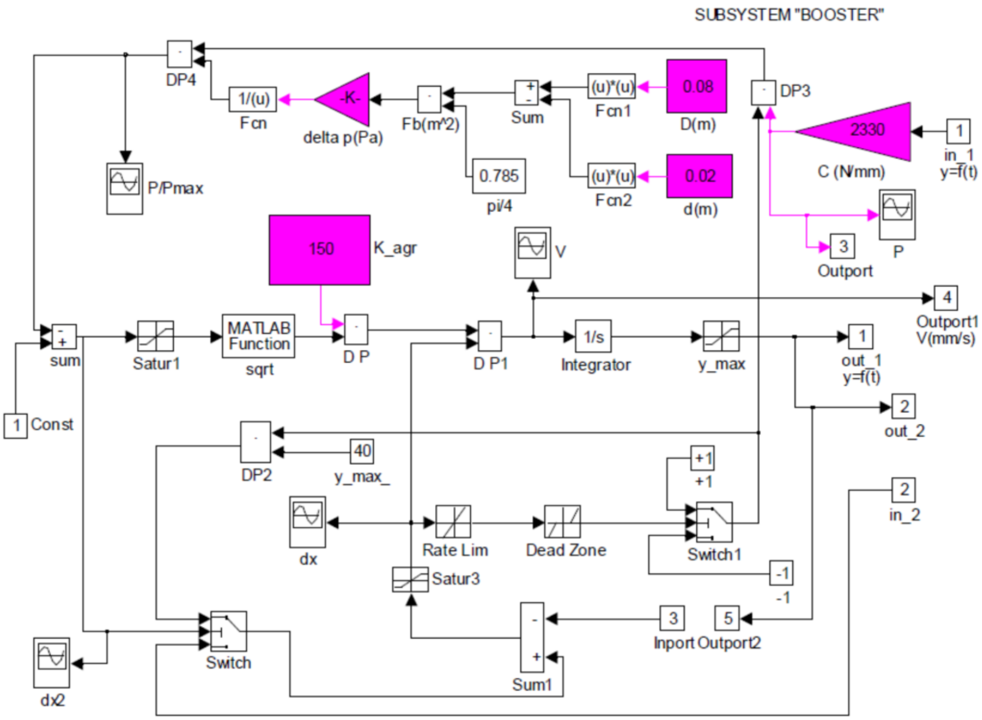

Fig. 2. Model of the BOOSTER subsystem

The possibilities of the model are to study different cases of load on the handlebars and steerable stabilizers. If the amplifier K_p is set to a factor $K_p < 1$, the aircraft control is a controllable stabilizer of subsonic speeds with overcompensation. It is characteristic of such controls that the deviation from the neutral position is associated with supporting efforts, and the return with counter-forces (as opposed to alternating phases for the rudders). The absolute value of the coefficient K_p can mimic the overload of the hydraulic booster $\left(\left|K_p\right| > 1\right)$ or work with a reserve of power capabilities $\left(\left|K_p\right| < 1\right)$. For example, a properly selected power steering in its operating range together with an aircraft rudder has a load factor $\left(\left|K_p\right| < 0{,}67\right)$ in the model. Overloading of the power steering is simulated with a factor $K_p > 1$.

Fig. 1 shows the limit cases for counteracting the load on the handlebars $\left(K_p = 1\right)$, when at the very end of the range of deflection of the handlebars, forces are obtained from the hinge moment equal to the power capabilities of the power steering. The feedback can be switched off by setting the Feed-back amplifier (Fig. 1) to 0 and thus recording the static load characteristics of the hydraulic booster similar to a test bench.

Basic idea of the model

It is known that the speed *V* of movement of the actuator is determined by the following factors:

• Difference between the pressures in the discharge and drain highways;

• The condition of the channels and the associated hydraulic resistance;

• The area of the piston;

• The opening of the channels for the passage of liquid through the distribution device (slide stroke Δx);

• The degree of load, as a ratio of the current to the maximum load P/P_{max} for the power steering.

The first three factors in modeling can be conditionally assumed to be constant (piston size, working fluid pressures, coefficient of resistance of the slide) and determine the so-called maximum speed gain *K*, where:

1) $$K = \frac{1}{F_p}\sqrt{\frac{p_H - p_C}{2K_d}}.$$

The physical nature of the coefficient K is the speed of the actuator (*mm/s*) at one stroke of the slide and is the slope of the throttle characteristic in the absence of load. It depends on the difference between the pressure of the discharge and drain lines, the size of the piston and the hydraulic characteristics of the switchgear. For each specific model of the hydraulic booster, this is a known value. Depending on the purpose of the power steering, the shaping of the channels and the profiling of the switchgear, the coefficient K does not exceed a value of $150\ (1/s)$ and usually occurs in the range $K = 10 \dots 125\ (1/s)$. In the model of Fig. 8, this is the constant $K_{agr} = 150\ (1/s)$, which corresponds to a booster with high speed. The other two factors, when operating the power steering as a tracking system, change the speed of movement of the actuator along its course. If the load is not taken into account, the speed is determined mainly by the stroke of the gate Δx, which is the difference between the input signal and the feedback signal. This difference in the impact with a stepped input signal at the input of the power steering changes, as at the beginning and end of the process is zero (the cross-section of the gate when the feedback is variable).

The main idea for modeling the movement of the actuator under variable load is realized by representing the speed V as a product of three multipliers:

- the constant $\mathrm{K} = K_{agr}$;
- the movement of the slide Δx;
- $\sqrt{(1 - \mathrm{P}/\mathrm{P}_{\mathrm{max}})}$.

2) $$V = \Delta x K \sqrt{(1 - P/P_{max})}.$$

After integration (block integrator in Fig. 2) at the output of the model is registered the movement of the actuator "*y*" in *mm*. It is limited for the specific model in the range ± 40 *mm*.

The maximum load that the hydraulic booster is able to take depends on the size of the piston and the difference between the pressures of the working fluid on both sides and does not depend on the stroke of the slide. Data on the geometric dimensions (constants *D = 0,08 m* and *d = 0,02 m*) and pressure (*Δp = 19700000 Pa*) are entered in the model, which determine $P_{max} = 92,8\ kN$.

The model can be readjusted for different sizes, fluid pressures or loads. The reconfiguration points of the model in Fig. 2 are colored.

Modeling results

The operation of the model of a power steering is presented with the results of the following graphs. Figure 3 shows *the static load characteristic of the hydraulic booster V = f(P)* at $x = x_{max} = const$. For this purpose, the model is tuned by ***switching off the feedback*** and setting the input signal from the "*Signal Generator*" block with rectangular polarity changing signals — frequency *0.5 Hz* (*3,14 1/s*) and amplitude *1,5 mm*. Operating time of the model *t = 2,5 s*. The coefficient K_p is set for counteracting load to the maximum power capabilities of the hydraulic booster $(K_p = 1)$. With such a setting in the end positions, the input signal keeps the channels of the switchgear open up to the maximum stroke ofthe slide (*±1,5mm*) — which is a condition for obtaining the static load characteristic in the four quadrants (alternating phases of counteracting and supporting force in both directions).

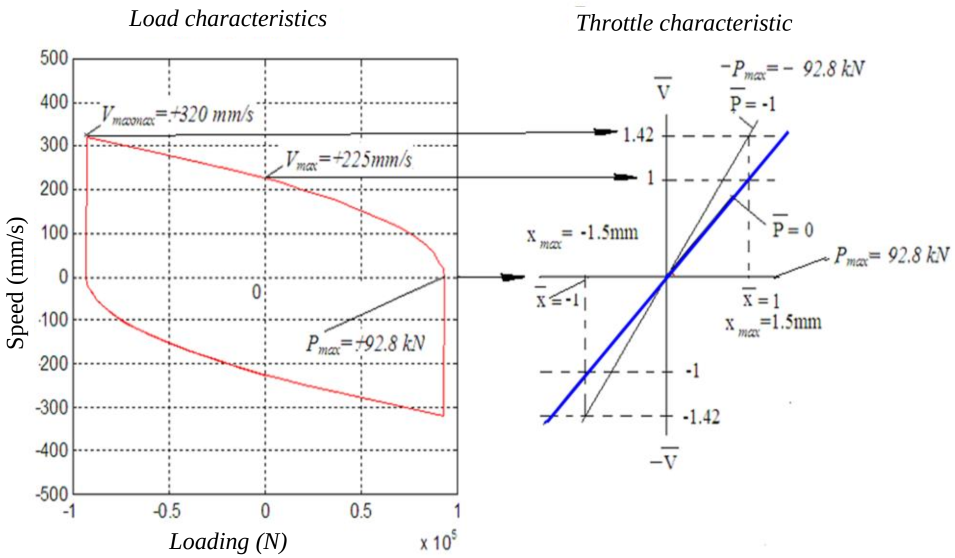

Fig. 3. Static load characteristics: abscissa- load P(N); ordinate- speed V(mm/s);
$P_{max}= \pm 92{,}8$ kN; $V_{max}= \pm 225$ mm/s; $V_{max\,max} \approx \pm 320$ mm/s;
Load characteristics V = f(x) at P = const.

Fig. 4 shows the stroke of the distribution valve at the set input signal and the switched off feedback. By characteristic points of the load characteristic ($P_{max}= \pm 92.8$ kN; $V_{max}= \pm 225$ mm/s; $V_{max\ max}= \pm 320$ mm/s) *the throttle characteristic of the hydraulic booster* can also be built.

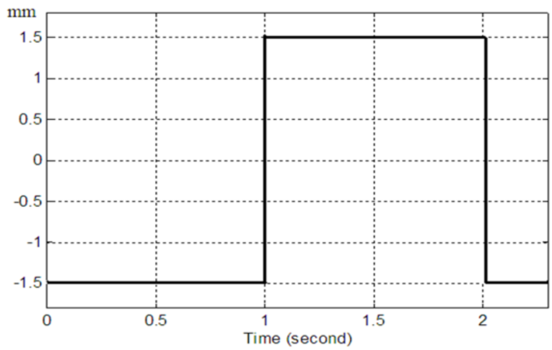

Fig. 4. Stroke of the camshaft x (mm) with feedback off to model
the load characteristic V = f(P) at $x = \pm x_{max}$

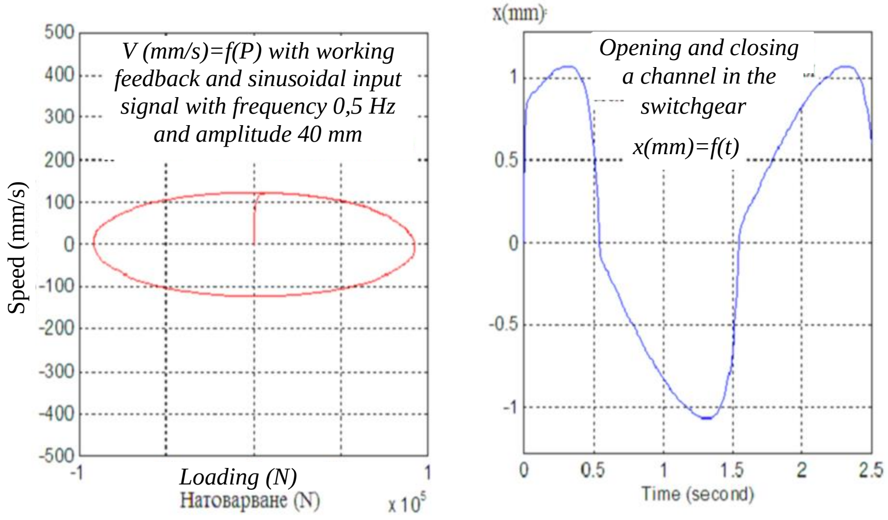

Fig. 5. Operation of the model with working feedback and input sinusoidal signal

The role of the feedback can be seen in Fig. 5 when the booster input device moves according to a sinusoidal law with a frequency close to the natural movements of the pilot (one complete deviation of the control and return to neutral for one second).

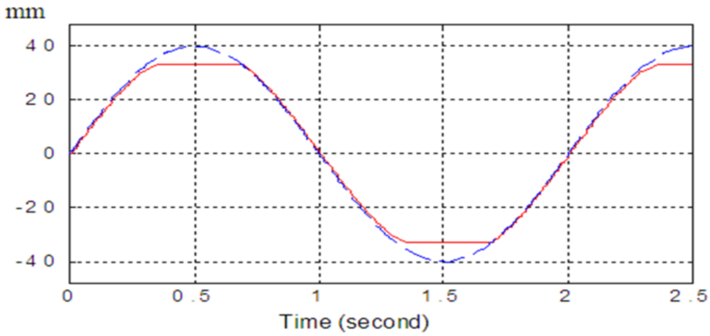

Fig. 6. Transient with sinusoidal input signal with amplitude 40 mm and frequency 0.5 Hz. The actuator, when overloaded with 20% $(K_p = 1{,}2)$ *of counteracting force, is visibly delayed by the input signal (dashed line), stops at a "stop" near the end positions* $(\pm 33\ mm)$ *when it reaches* $|P_{max}| = 92800\ N$ *and not can use the entire control range.*

The phenomenon of "stop" is demonstrated by the model of Fig. 6 when operating the power steering as a tracking system (with feedback included). In real practice, this phenomenon can occur even with properly designed aerodynamic rudders, because it is related to the ultimate power capabilities of the booster, and they change with decreasing operating pressure (for example, failure in the control system of the booster pump). The exclusion of such phenomena in practice is done in the design with the selection of a sufficiently strong hydraulic booster (with a load margin).

Simulation of asynchronous parallel drive

For the purposes of modeling, the model of Fig. 1 is modified in the form of Fig. 7.

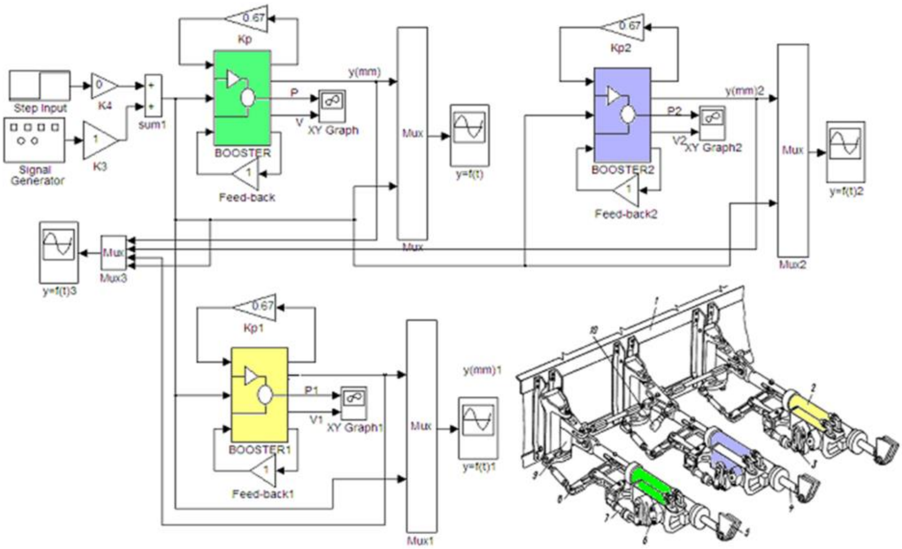

Fig. 7. Model of multi-chamber parallel drive with the same hydraulic boosters and pressure differences of the hydraulic systems

Differences in operating pressures affect the force P_{max} and speed characteristics K_{agr}.

The different hydraulic systems in the model are set to work with different operating pressures, which hypothetically may be real in the functions that these systems perform. The results shown in Fig. 8 show that if hypothetically the hydraulic boosters are selected in the design process to work without load margin (in the model setting K_p, K_{p1}, $K_{p2} = 1$), the actuators tend to move at different speeds.

The desynchronization is most noticeable at the end of the turn. This further burdens the drive control, which absorbs these differences and redistributes the load so that slower power boosters receive support from faster ones and vice versa. This synchronization mechanism cannot be modeled with the proposed model, but it can prove that with proper selection of hydraulic boosters (with a margin of 50% by piston rod force) the pressure difference of different hydraulic systems up to 10% does not contribute significant desynchronization in the movements of the actuators (the existing asynchrony is permissible in operating conditions). This is shown in the right graph of Fig. 8 and is achieved by adjusting the load coefficients in modeling K_p, K_{p1}, K_{p2} with the same value 0,67, which is a ratio between *1* and *1,5* and is recommended by the theory of hydraulic propulsion of aircraft controls [4].

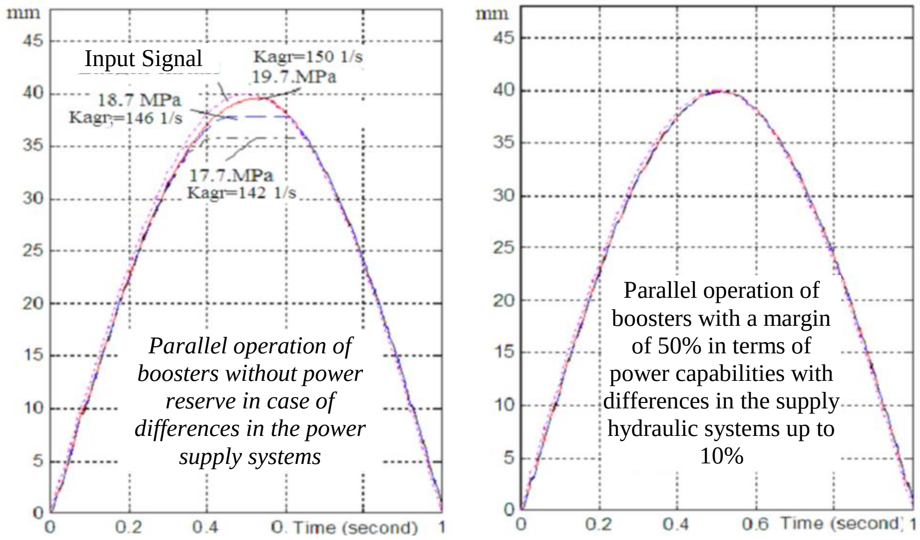

Fig. 8. Stroke of actuators in parallel operation, when the power supply hydraulic systems have different pressures within the operational admissibility (differences up to 10% ... 15%). The input signal is sinusoidal with a frequency of 0.5 Hz and amplitude of 40 mm

The location of the operating area in relation to the speed and power limit characteristics, according to these recommendations is shown in Fig. 9 — left graph. The two zones are obtained with the model of Fig. 7 with different setting of two of the modeled boosters and subsequent graphic processing.

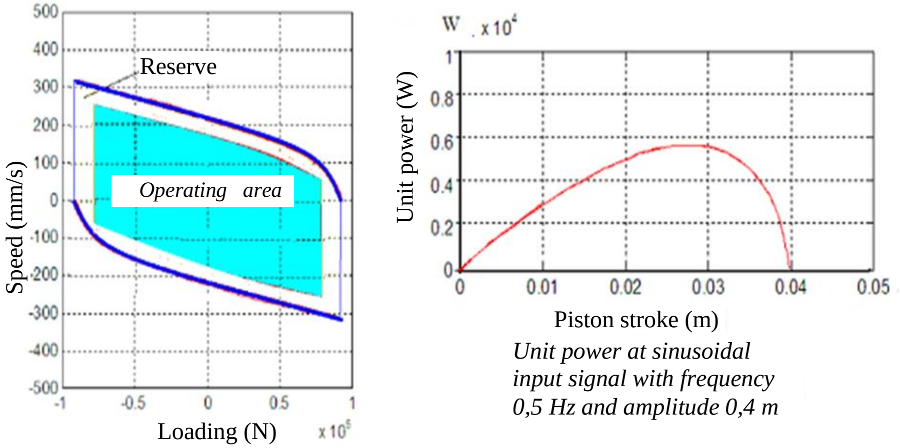

Fig. 9. Approximate location of the operating area and the actual power capabilities of the booster on the load static characteristic; power of the steering unit (power steering - transmission - steering wheel) in case of steering deviation

When operating the booster in real conditions, the maximum power is obtained at about 75% of its maximum stroke. The model can also, with some tuning and development of its graphical capabilities with the Sinks library from Matlab-Simulink, confirm this fact known from the theory. The dependence of the power along the stroke of the piston rod is shown in Fig. 9-right graph.

Conclusion

The work of the model confirms the main idea of the recommendation in the selection of hydraulic actuators: to work with a margin of at least 20% and strength — 50%.

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РОЛЯ НА РЕЗЕРВА ОТ МОЩНОСТ ПРИ ПАРАЛЕЛНИ ХИДРОЗАДВИЖВАНИЯ НА САМОЛЕТНИ ОРГАНИ ЗА УПРАВЛЕНИЕ

Д. Йорданов, Н. Загорски, Ст. Чолаков

Резюме

Статията представя процес на моделиране на паралелно бустерно управление на органи за управление на транспортен самолет, като е демонстрирана положителната роля на запаса от мощност за работата на конструкцията при допустими експлоатационни разлики в наляганията на захранващите системи.

Паралелните хидравлични задвижвания са характерни за големите транспортни самолети, при които един орган (например, кормило за височина или направление) се отклонява от няколко хидроусилватели (бустери). Всеки от тези хидроусилватели се захранва от отделна хидросистема и, естествено, съществуват допустими разлики в работата на задвижващите устройства. Органът за управление, който тези устройства задвижват, е този елемент, чрез който при реални условия се преразпределя общата работа между отделните хидроусилватели. Едните (по-бързите) работят с донатоварване, а други (по-бавните) работят с подпомагащи усилия.