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Research of the Possibility of Reducing the Weight of Aircraft Onboard Hydraulic System by Application of the Principle of Pressure Boosting at Failure

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Abstract. In this article explores the possibility of reducing the mass of aircraft's hydraulic system by a pressure adaptive system. As initial data, the hydraulic system of a promising front-line bomber considered. The determination of the mass of aircraft's hydraulic system based on the loads perceived by the actuators of the aircraft's control system, various levels of pressure in the hydraulic system, as well as the force margin of the power actuators of aircraft's control system. In course of this exploration was made a comparison between of masses of the units of the system with excessive force drivers and the masses of the units of a pressure adaptive hydraulic system. As the result of this comparison, it was found that the use of a pressure-adaptive system can reduce the mass of the designed hydraulic system in the range from 8 to 22,5%.

1 Introduction

The most important criteria for assessing the quality of any aircraft system are the likelihood of special situations occurring during flight and weight. The other parameters can be considered as constraints and not criteria for quality. By comparing reliability and weight as quality criteria during the design process, numerous arguments can be found in favour of each, as well as the relationship between these parameters [1]. In modern aircraft, the hydraulic system occupies a key position. The hydraulic system includes important subsystems such as the aircraft control subsystem and the landing gear deployment-retraction subsystem. The hydraulic system also performs functions such as opening and closing cargo compartment doors (on large transport aircraft), brake control, nose wheel steering, and more.

Aviation Safety. Currently, the required level of reliability and fail-safety on aircraft is achieved through duplication and redundancy of units and systems. According to statistical data, the weight of the hydraulic system constitutes 1-2% for heavy aircraft and 4-5% for light manoeuvrable aircraft based on the maximum take-off weight of the aircraft. In modern aviation hydraulic systems, the desired level of fail-safety is achieved by selecting the initial structure and parameters of the system during the design process, which remain

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unchanged during flight. To increase the reliability of the system, engineers resort to higher safety factors and the use of redundancy/duplication of elements, which inevitably leads to an increase in the overall weight of the system. However, it is possible to increase fail-safety without increasing weight by adjusting the internal parameters of the system to evolving operating conditions. A particular case of adaptive system adjustment under changing operating conditions is systems with pressure boosting. According to works [2, 3], the application of pressure adaptive systems on heavy aircraft with extensive pipelines allows reducing the overall weight of the hydraulic system by 40%. These works also mention that even greater weight reduction is possible by using titanium pipelines [4]. Additionally, a similar study was conducted based on the heavy fighter aircraft [5]. As a result of this study, a 15% weight saving was achieved, and the feasibility and functionality of these systems were confirmed experimentally.

2 Problem statement

The above-mentioned works do not address the possible reduction of the system's lifespan due to operating on forced modes. Furthermore, the question of the need for additional reinforcement of the system components to reduce the probability of system failure in forced modes remains open. The aim of this study is to assess the potential weight reduction of the hydraulic system through the application of pressure boosting during failures, as well as to determine the acceptable level of reinforcement for the system components, while maintaining their practicality.

3 Research methods

The research was carried out based on the prototype front-line bomber with two engines. Analytical formulas based on the principles of the 3rd strength theory [6, 7] were used to determine the mass of the hydraulic system components and pipelines. The mass of the hydraulic fluid tanks was calculated using the methodology [8]. The assessment of weight reduction potential was based on comparing the calculated mass of the system with pressure boosting capability against a similar system without such capability. The calculation of the necessary geometric parameters of the hydraulic system components was based on their power characteristics. The determination of the power characteristics of the hydraulic system was carried out using the methodology described in the literature [9], as this method provides an acceptable level of accuracy. All calculations and graphs were performed using the MathCad software package.

4 Research of the possibility of reducing the weight of aircraft onboard hydraulic system by application of the principle of pressure boosting at failure

The effective area of hydraulic drives is commonly calculated using the formula [6]:

$$F_n - f_{st} = \frac{R_{\max}}{(1-a)\cdot P_n\cdot(n-n_{otk})} \tag{1}$$

where:
R_{max} - maximum load overcome by the drives, N;
P_n - nominal pressure level in the hydraulic system, Pa;
a - coefficient of relative pressure losses in the hydraulic system;

n - number of hydraulic drives in a given control channel;
n_{otk} - number of permissible failures without consequences;
F_n - piston area of the hydraulic drive, m^2.
f_{st} - rod area of the hydraulic drive, m^2.

From formula (1), it can be observed that the effective area of the piston chamber increases with an increase in the number of permissible failures without consequences. This leads to an increase in the mass of the hydraulic drive due to its inclusion of a chamber that does not contribute to generating necessary force at the end of stroke for moving a specified load at required speed. However, this chamber is necessary to meet flight safety requirements.

The principle behind pressure boosting is to eliminate excess power from drives and, in case failure occurs which could result in loss or failure to accomplish important combat tasks, raise working pressure levels within the system to enhance stiffness among remaining drives and maintain system functionality at previous levels.

When applying pressure boosting principle, formula (1) takes on this form:

$$F_{nf} - f_{st} = \frac{R_{\max}}{(1-a)\cdot P_n \cdot n} \qquad (2)$$

In other words, when designing a system we do not include excessive piston areas required for compensating failure from one drive.

The required pressure level for compensating failure is determined by this formula:

$$P_f = \frac{R_{\max}}{(n-n_{otk})\cdot\left(F_{nf} - f_{st}\right)\cdot(1-a)} \qquad (3)$$

Let's compare nominal pressure level and forced-pressure level:

$$\frac{P_f}{P_n} = \frac{F_n}{F_{nf}} = \frac{n}{\left(n-n_{otk}\right)}, \qquad (4)$$

or:

$$P_f = P_n \cdot \frac{n}{\left(n-n_{otk}\right)} \qquad (5)$$

Reducing effective piston area will lead to changes in energy characteristics within a system. Primarily affecting pump performance and consequently its power output.

Pump power output is determined by this formula:

$$N_n = Q_n \cdot P_n, \qquad (6)$$

where Q_n - theoretical pump performance, l/min.

Theoretical pump performance is determined based on ensuring simultaneous operation of control elements and external hydraulic energy consumers.

The calculation formula for theoretical productivity will have the following form [6]:

$$Q_n = \frac{2}{3}\cdot\sum_{i=1}^{3} Q_i + Q_m, \qquad (7)$$

where Q_i - flow rate in the i-th control channel, l/min; at a mode corresponding to the maximum hydraulic energy consumption, Q_m - flow rate to the most power-consuming consumer, l/min.

Within the scope of studying the influence of pressure boosting capability in the system on the total mass of the hydraulic system, several implementation options of this principle are possible.

Pressure boosting without strengthening components. In this case, no additional safety factors are assumed, and strength calculations will be performed based on system operation conditions at nominal operating mode rather than boosted.

Pressure boosting with component reinforcement. In this case, strength calculations for components will be carried out based on system operation conditions at boosted operating mode rather than nominal. As shown in formula (5), complete compensation for failure in a control system requires doubling the pressure for an aircraft equipped with two systems onboard. However, all hydraulic system components are designed to withstand three times higher than nominal pressure levels. In this case, the overall safety factor will be six units which may prove excessive. Several options for choosing an overall safety factor are considered within this scenario.

Below is a graph showing the dependence of the mass of the hydraulic system on the level of nominal pressure for a system with the ability to boost pressure without strengthening the components.

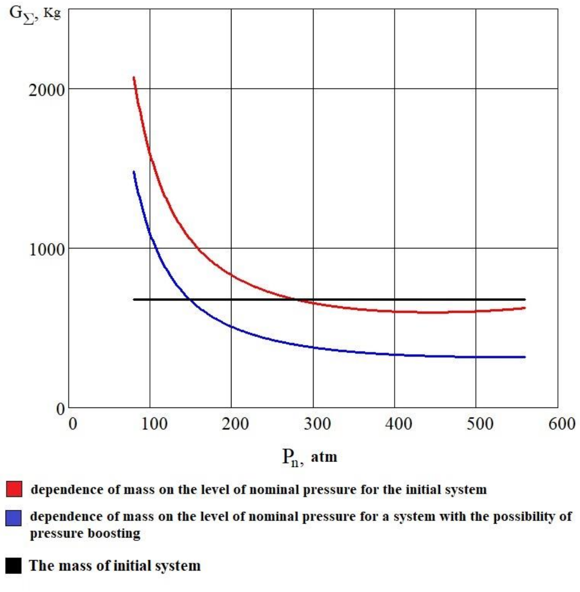

Fig. 1. Mass characteristics of systems with pressure boosting capability, with a main structural material strength σ_b = 1080 MPa, and titanium pipelines.

The considered case provides maximum possible weight savings and is idealized. In this case, weight savings at the same level of nominal pressure are approximately 45%, while the maximum weight savings when increasing the level of nominal pressure are about 62% compared to the initial system mass.

Let's consider a case of a system with pressure boosting capability, with component strengthening and a safety factor equal to six units, where steel with yield strength σ_b = 1080 MPa is used as the main structural material along with titanium pipelines.

Below is a graph illustrating the dependence of hydraulic system mass on the level of nominal pressure.

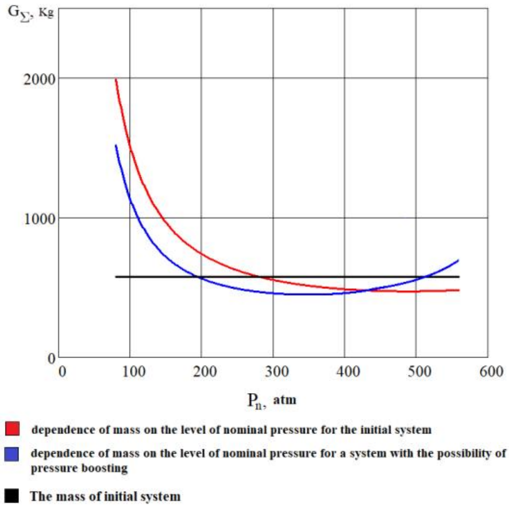

Fig. 2. Mass characteristics of systems with pressure boosting capability, component strengthening σ_b = 1080 MPa, and titanium pipelines.

From this graph it can be seen that there exists a range within which weight savings are possible when using a system capable of boosting pressure. However, these savings are mainly achieved through using titanium pipelines and are insignificant. It is absolutely evident that in this problem formulation it is possible to achieve weight savings by applying pressure boosting during failures; however, these gains do not outweigh difficulties associated with implementing this principle. Weight savings relative to traditional systems with similar parameters lie within an approximate range of 8%. However, having sixfold safety margin in terms of pressure may prove excessively redundant.

According to OST 1 00095-73, all hydraulic system components must withstand three times the nominal pressure level before failure [10]. Increasing the pressure in the system by two times brings us to a critical threshold beyond which system failure can occur with potential catastrophic consequences. However, it is important to understand the following:

Pressure in the system is only increased in cases where a failure in the control system could lead to aircraft loss or failure to accomplish critical combat tasks.

The majority of time, the system operates under nominal conditions, meaning that the ability to increase pressure does not affect its performance or lifespan. Introducing additional reinforcement for compensating work on forced modes enhances both longevity and reliability when operating normally.

Our investigation focuses on the most severe possible scenario: our aircraft has only two hydraulic systems, resulting in maximum pressure increase (required for compensation) of two units. Let's approach this task from another angle and calculate the "critical" value of safety factor regarding pressure at which point our design starts losing mass compared to a traditionally designed analogous system.

Let's approach the problem from the other side and calculate the "critical" value of the pressure safety factor, after which the system begins to lose in mass to a similar system designed in the traditional way. We will search for the critical value from the condition of equality of the mass of the unit designed for forced pressure and the unit designed according to the traditional scheme.

5 Selection of the optimal safety margin for system with the possibility of pressure boosting

Let's introduce the pressure increase factor k_f, which is a multiplier applied to the calculated pressure in strength calculation formulas.

$$P_f = k_f \cdot P_n \tag{8}$$

The overall safety margin in this case is:

$$n_\Sigma = k_f \cdot n, \tag{9}$$

where n represents the safety margin according to OST 1 00095-73 and equals three units. Thus, k_f = 1 corresponds to the calculated case without additional reinforcement of components, while $k_f = \frac{n}{(n - n_{otk})}$ corresponds to the case of reinforcement sufficient for full compensation of system operation under forced modes.

The graphical dependence of coefficient k_f for drives is shown below.

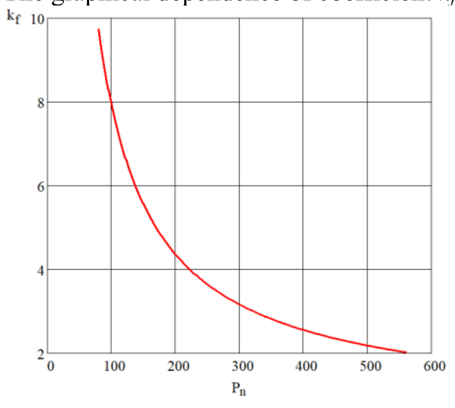

Fig. 3. Graphical dependence of coefficient k_f for drives using the example of AMHT drive.

From this graph, it can be seen that the dependence of coefficient k_f has a nonlinear decreasing nature. Based on this dependency, we can conclude that as the level of nominal pressure increases, there is a reduced likelihood of achieving weight savings through applying pressure boosting principle during failures. If the value of k_f at current nominal pressure level drops below $k_f = \frac{n}{(n - n_{otk})}$, it means that at this nominal pressure level, it is not possible to achieve weight savings through applying pressure boosting principle even with component reinforcement sufficient for full compensation of system operation consequences under forced working conditions.

Let's calculate value k_f for different components and nominal pressures in the system:

Table 1. Values of coefficient k_f for different components and levels of nominal pressure in the system.

Unit	k_f				
	P_n, atm				
	210	280	350	420	560
Axial piston pump	3,72	2,85	2,33	1,98	1,55
AMHT actuator.	4,14	3,30	2,79	2,44	2,00
Rudder actuator	3,56	2,90	2,50	2,22	1,87
Flaperon actuator	3,20	2,65	2,32	2,10	1,79
Actuator of the deflected toe of the wing	3,36	2,77	2,40	2,15	1,83
FHT actuator	2,87	2,43	2,16	1,97	1,72
Main landing gear release-retraction drive	1	1	1	1	1
Front landing gear release-retraction drive	1	1	1	1	1
Pipeline	2,36	2,05	1,85	1,71	1,54

Based on this table, several conclusions can be drawn:

Firstly, weight savings are possible for most components even with component reinforcement sufficient for full compensation during forced working conditions. This can be visually demonstrated in a comparative table:

Table 2. Comparison of aggregate masses between the traditional system and the system with pressure boosting capability at $k_f = 2$.

Unit	Mass, *kg*	
	System Type (P_n = 280 *atm*)	
	Conventional	System with ability of pressure boosting, with $k_f = 2$
Axial piston pump	96	53
AMHT actuator.	73	43
Rudder actuator	27	18
Flaperon actuator	46	33
Actuator of the deflected toe of the wing	32	22
FHT actuator	19	9
Main landing gear release-retraction drive	73	73
Front landing gear release-retraction drive	19	19
Pipeline	115	47

Secondly, for all aggregates except for the chassis drive systems, there is a potential mass reduction when using a coefficient of increased calculated pressure k_f equal to 1.5, even at very high levels of nominal pressure. Thirdly, it is not possible to achieve a mass reduction for the chassis drive systems by applying the pressure boosting principle during failures since these drives do not have inherent redundancy in the first place. To eliminate the influence of pressure boosting on non-redundant aggregates in the system, it is necessary to use relief valves. Below are graphs depicting the dependency of systems with pressure boosting capability where an increase in mass for non-redundant power units has been excluded.

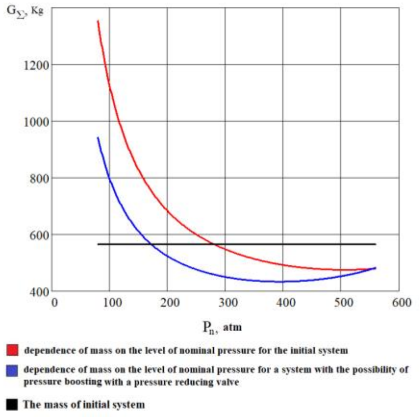

Fig. 4. Comparative dependency graph of systems with relief valve at $k_f = 2$.

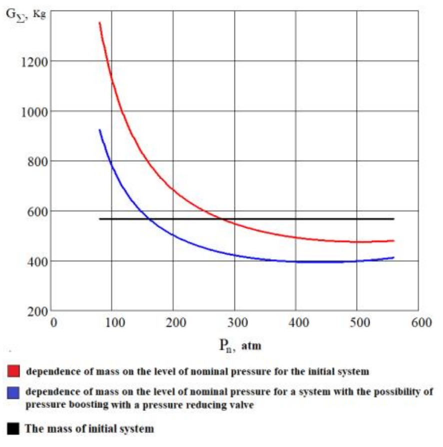

Fig. 5. Comparative dependency graph of systems with relief valve at $k_f = 1.5$.

6 Conclusion

During this research, the following results were obtained:

Firstly, it was shown that applying the pressure boosting principle reduces the installed power of the system by 1.3 to 2 times.

Secondly, it was demonstrated that applying the pressure boosting principle without strengthening the components of the system to compensate for its operation at boosted modes can result in a weight reduction of up to 45% compared to an equivalent conventional system.

A particularly interesting case was a system with the possibility of pressure boosting accompanied by component strengthening. The study of such systems yielded the following results:

Firstly, there is a potential weight saving ranging from 8% to 25.5% compared to a traditional system with similar parameters.

Secondly, there is a need for practical implementation and further detailed investigation of such systems. Current aviation regulations and standards do not consider or regulate the use of these systems; thus, design rules have not been established regarding their strength requirements, service life expectations, and reliability criteria. Additionally, both the weight and maximum achievable benefits from their application strongly depend on the pressure increase coefficient (k_f).

Complete compensation of the consequences of system operation under forced modes may prove to be an excessive solution, while reducing the overall safety factor can be justified by considerations of mass reduction. It should be noted that for a significant portion of time, the system operates in normal mode where all requirements for strength, durability, and reliability are met. Therefore, introducing additional reinforcement will only improve these indicators and short-term operation under forced modes will not lead to premature failure of functional elements within the system.

Taking into account the aforementioned points as well as our obtained results, we can conclude that reliability tests and resource testing of such systems are necessary. Based on these tests, it is important to establish the required safety margin for such systems before making a final assessment regarding their development and application prospects with pressure boosting capabilities.

It should be noted that the maximum working pressure is limited by the ability to ensure reliable sealing and currently ranges from 280 to 350 atm [11]. There are nanosystems with a working pressure of 420 atm, but they have not yet gained widespread use.

Considering this fact and the results obtained, the application of systems with pressure boosting capabilities currently appears as an intermediate solution between traditional systems with a nominal pressure up to 280 atm and traditional systems with a nominal pressure exceeding 420 atm.

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