ATLANTIS PRESS
Advances in Engineering Research (AER), volume 157
International Conference "Actual Issues of Mechanical Engineering" (AIME 2018)

FUNDAMENTALS OF POSITIVE FEEDBACK WITH ADDITIONAL CONTROL

Maksim Dmitrievich Gavrilenko
Department "Metal-Cutting machines and tools"
Don State Technical University
Rostov-on-Don, Russia
aaakot@yandex.ru

Alexey Romanovich Lebedev
Department "Metal-Cutting machines and tools"
Don State Technical University
Rostov-on-Don, Russia
alex-diplom@mail.ru

Sergei Vasil'evich Lesnyak
Department "Metal-Cutting machines and tools"
Don State Technical University
Rostov-on-Don, Russia
serg.les2010@yandex.ru

Vladimir Ilyich Kravchenko
Department "Scientific and Technical Translation and Professional Communication"
Don State Technical University
Rostov-on-Don, Russia

***Abstract*–The principle of an additional control in an adaptive friction clutch with a positive feedback is suggested. The authors found the dependence of a torque value of the additional feedback actuation device on the current value of friction coefficient when the clutch with the positive direct-action feedback transfers the torque, constant in magnitude, not depending on the current value of coefficient of friction.**

Keywords – adaptive friction clutch, positive feedback, the coefficient of friction, actuation device, torque.

I. THE STATE-OF-THE-ART REVIEW

A feedback is widely used in adaptive friction clutches (AFC) which function as a safety for protecting units and machine parts from torque overloads [1-3]. The AFC with negative feedback is most frequently used in practice [1]. Using the AFC equipped with this type of the feedback sharply decreases the load capacity, since the spacer force acting on a friction group reduces its closure force, thereby reducing a frictional force.

In view of this disadvantage AFCs with negative feedback are only slightly used in current technology [4]. In addition the responce accuracy of the mentioned AFCs is relatively small [5], which does not allow in some cases to successfully solve the tasks of effective overload protection of machine drives.

The positive feedback in up-to-date AFCs is used as a separate type of the feedback [6], or as a component in the combined positive-negative feedback [7].

The undoubted advantage of the positive feedback is an increase in the rated load capacity of the AFC, as a part of which it is applied. The nature of the positive feedback as a part of the AFC is in the fact that an actuation device (AD) in the process of load transfer creates an axial force, by which the friction faces of the friction group are pressed against each other. The magnitude of the mentioned force depends on the overload level of the drive and on the current value of the friction coefficient between friction pairs.

The structural model of AFC with the AD of the positive feedback constructed on rigid structural members provides downforce increase as a value of the friction coefficient and of the overload grow in the drive. Thus, this option of the clutch is used as a safety device providing the increased reliability of the production machine when performing its technological stress on a certain medium.

The precision characteristic of the clutch does not allow classifying it under the clutches providing high stability of the transferred loading under conditions of change in the value of the friction coefficient [8].

The problem of achieving a high response accuracy of the AFC with the positive feedback is partially solved in [6, 7]. The first of the specified works deals with the AFC equipped with the AD constructed on the basis of V-shaped resilient petal elements with so-called "negative" slope angle of petals compared with rolling elements to create the positive feedback.

While in operation the rate of a slope angle of petals decreases in relation to magnifying transmitted load due to the increasing value of a friction coefficient. From this it follows respectively, the latter reduces a force magnitude of pressure of friction faces of friction pairs to each other. It maintains automatic control of the breakdown torque value of AFC on the variation interval of the friction coefficient values.

This way of automatic control is remarkable for a relative simplicity of design realization of the AFC; however, it does not ensure the complete stabilization of the running torque value. This is due to the fact that elastic behaviour of a V-shaped petal does not ensure precisely specified correspondence between the value of hold down pressure for the AD of the feedback and the current value of the friction coefficient.

The second work deals with a positive feedback as a component of the combined positive-negative feedback. Therein the positive feedback is implemented by means of the same technical means as in case of the former.

In this particular case, there is a clear separation of the activity periods of the feedback: a positive one is within the variance of the friction coefficient of $f_{min} \ldots f_{cp}$ (where f_{min}, f_{cp} is respectively minimum and average (rated) values of the friction coefficient); a negative one is within the interval of $f_{cp} \ldots f_{max}$ (where f_{max} is the maximal value of the friction coefficient).

The disadvantage of the above listed modes consists in insufficiently high response accuracy of the AFC equipped with such AD of the feedback.

The fundamentals of the complete stabilization of the break torque value for the AFC with the negative single-loop feedback and with the additional control device are presented in [9]. The substance of the way typified by the conducted investigation is in "taking-away" of some part of the running torque from the feedback actuation device terminal.

To implement this way it has been beforehand established that realization of an "ideal" full-load characteristic of the AFC requires a variability of the spacer force of the AD according to a definite regularity.

The mentioned change, which is realized by means of the AD with a stationary value of CD, is possible when the running torque value coming to the AD terminal changes according to the specified pattern, which differs from the effectual pattern in the existing AFC with the inverse negative feedback.

In [10] we have carried out a synthesis of the AFC with the negative feedback in which a principle of the additional control is partially realized. It has been established that this modified option of the AFC has a higher response accuracy in spite of the fact that the additional control device does not ensure a change in the running torque value that comes to the feedback actuation device terminal according to the specified pattern.

The preliminary analysis showed that the above-mentioned way can be used in the AFC with the positive feedback.

II. Statement of a problem

Establishment of parameters of an additional control device in the AFC with a positive direct-action feedback providing the complete stabilization of the break torque value of the clutch.

III. The solution to the problem

Take the basic option of the AFC with the positive direct-action feedback which key diagram is shown in Fig. 1.

The clutch consists of two half- clutches of 1 and 2 that is kinematically linked to one another with a package of friction disks of 3 and 4. Disks 3 are connected by means of splines to a hub of the pressure disk 5, which is deprived of kinematical connection in the circumferential direction with a hub of half-clutch 1, except for a slight sliding friction, which is not considered in a further research. Disks 4 are connected in the similar way to a drum of half-clutch 2.

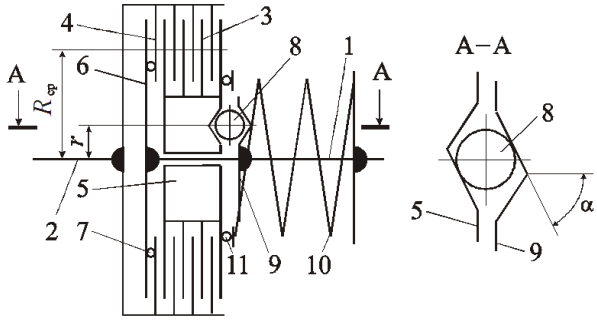

Fig. 1. AFC with the positive direct-action feedback schematic diagram

The thrust bearing (7) is installed between the thrust disc (6) and the extreme left-hand (see Fig. 1) clutch friction disc (4), therefore the packaging scheme of clutch friction group is made in terms of "all the friction pairs being driving" [11].

The AD of the clutch is made in the form of rolling bodies (8) which are placed in rebates with slanted sidewalls of pressure disk 5 and in the form of the bearing sleeve (9) which is rigidly installed on a hub of half-clutch 1 (Fig. 1, Section A-A).

The force closure of friction pairs which is necessary for a starting moment of friction forces and, therefore, downforce x in the course of work, is carried out by the spring 10 established with a preliminary tension and transmitting an effort to a pressure disk 5 by means of a thrust bearing 11.

In the course of work, the pressure disk 5 influences rolling bodies of 8 which transmit a running torque from half-clutch of 1 to half-clutch of 2 therefore between rolling bodies and sidewalls of rebates there is normal force F_n which can be spread out to components F_t – circumferential (tangential) force and F_p – axial downforce.

The value of the force F_p apparently depends on a value of the normal force F_n which is conversely determined by the running torque value taken up by a pressure disk (5). The value of the running torque coming to the pressure disk depends on the current value of the friction coefficient between friction disks 3 and 4, on the value of the current force of pressing friction disks to each other as well as on the parameters of clutch friction group.

The force F_p exerted upon a friction face of clutch friction discs 3 and 4 maintains additional pressing them to each other. Thereunder we will write an expression for computing a value of the current downforce:

$$F_{pi} = \frac{T_{ni}}{r} \mathrm{tg}\alpha , \quad (1)$$

where T_{ni} the running torque exerted upon the pressure disk 5; α is a slant angle of sidewall rabate under a body rolling 8 (see Fig. 1, Section A-A); r is a radius of a circle on which rolling bodies 8 are located (see Fig. 1).

We will write a general expression for computing a value of the breakdown torque of the AFC as:

$$T_{pi} = zR_{cp} f_i (F_{pr} + F_{pi}), \quad (2)$$

where Z is number of friction face pairs of a friction group; R_{cp} is an average radius of friction faces; f_i is a value of the current coefficient of friction; F_{pr} is spring tension force 10; F_{pi} see above.

Expression (2) is written as specified in common principles of the positive feedback action.

The results of investigating a condition of obtaining an "ideal" full-load characteristic of the AFC equipped with a positive feedback are given in work [12]. The "ideal" full-load characteristic of the clutch signifies a total absence of dependence of the breakdown torque value on the friction coefficient.

The condition given above in a mathematical form appears as follows:

$$F_{pi} = F_{pr}\left(\frac{f_{max}}{f_i} - 1\right), \quad (3)$$

where f_{max} is the maximum coefficient of friction. The notations of other parameters in the formula (3) are given above.

The analysis of the formula (3) shows that the peak value of downforce is equal to:

$$F_{pmax} = F_{pr}\frac{f_{max} - f_{min}}{f_{min}}$$

and the minimum value is equal to zero at $f_i = f_{max}$ (f_{min} is – the minimum coefficient of friction).

After the substitution of the right-hand side of Ratio (1) for Expression (2) we will solve the composed equation for an unknown quantity of T_{pi}. We get:

$$T_{pi} = zF_{pr}R_{cp}\frac{f_i}{1 - zCf_i}, \quad (4)$$

where C is the gain constant (GC) of the positive feedback:

$$C = \frac{R_{cp}}{r}\mathrm{tg}\alpha$$

By condition of configuration of the considered AFC, it is accepted that α=const (see Fig. 1, sectional view A-A).

Substituting the right-hand side of the ratio (4) for Expression (1), we will write a computing formula for a downforce value:

$$F_{pi} = zF_{pr}C\frac{f_i}{1 - zCf_i}. \quad (5)$$

For obtaining an "ideal" full-load characteristic, the downforce value is to vary according to the dependence (3).

Let us consider the graphs in Fig. 2 to compare regular changes in magnitude of forces F_{pi}, according to ratios (3) and (5). The curves of 1 and 2 reflecting respectively dependence (3) and (5) are constructed on the following source data: F_{pr} = 800 H, Z =4, f_{max} =0,8. The magnitude of C which depends on a pressure angle of α (see above) was calculated taking into account the following circumstance.

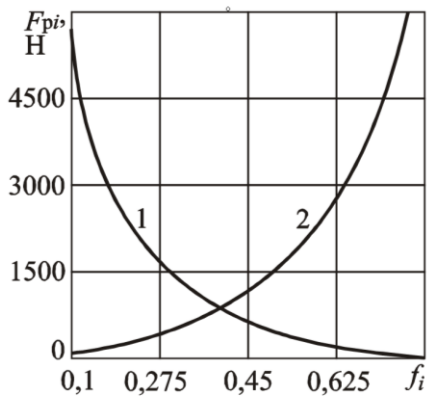

Fig. 2. Influence of the friction coefficient on the clamping force of the real and "ideal" AFC of the third generation

The scheme of the positive feedback in AFC assumes rolling bodies' pressure action on pressure disk (5). Proceeding from the condition which excluding self-braking (jamming) of the friction pairs of 3 and 4, it is to be carried out an inequality in the form [12]:

$$\mathrm{tg}\alpha_{max} < f_{max}$$

where α_{max} is a peak value, by condition of lack of self-braking of elements of friction pairs, the value of a slant angle of sidewall rabate under a rolling body; f_{max} see above.

At R_{cp} = 0,09 m, r =0,03 m and tgα=0,7 we get C= 0,3125.

The curves in Fig. 2 show the behaviour of magnitudes F_{pi} is not identical: according to the demanded pattern of change of the magnitude of the force F_{pi}, it is to decrease from a maximum at the value of the friction coefficient f_{min} to zero at the value f_{max} (Curve 1) whereas the force of F_{pi} increases from a minimum at the value of f_{min} ad infinitum at the value of f_{max} (Curve 2) in the AFC executed according to the scheme according to Fig. 1.

In the latter case the magnitude of force F_{pi} becomes infinite in case of a friction coefficient f_{max} and it physically means self-braking of friction couples of 3 and 4 (see above).

If used the principle of additional control in the AFC with the positive feedback the device of an additional feedback is to "to take a part away" from the running torque entering on the main actuation device of the positive feedback to provide unloading of the latter and the required difference between values of the running torque of an actual AFC (according to the scheme in Fig. 1) and an "ideal" clutch with the positive feedback.

The running torque values transmitted by the "ideal" clutch with the positive feedback is to be independent of a current value of the friction coefficient and is equal to:

$$T_{pr} = zF_{pr}R_{cp}f_{max}. \quad (6)$$

The formula (6) reflects absence of the downforce of the AD of the positive feedback at a maximal value of the friction coefficient.

The running torque value, which transmits an additional AD, is defined as the above difference, taking into account ratios (4) and (6) gives:

$$T_{pi1} = T_{pi} - T_{pr} = zF_{pr}R_{cp}\left(\frac{f_i}{1 - zCf_i} - f_{max}\right). \quad (7)$$

The function graph (7) is given in Fig. 3 (Curve 1). The straight line 2 in the same figure reflects a graphic chart of the load curve of an "ideal" AFC with the positive feedback, and Curve 3 shows the function graph (4).

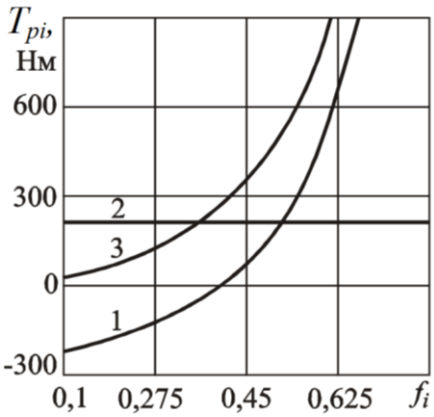

Fig.3 Load characteristics of real and "ideal" AFC of the third generation with positive feedback

The graphic charts are based on the source data given above. The graphic charts (Curves 1 and 3) show that the corresponding running torque values change in a similar fashion, however it is necessary for implementation of the scheme AFC with additional control in the range of values of the friction coefficient f_i =0,1…0,37 not to take a part away from the running torque entering on the main actuation device of the positive feedback, but on the contrary to arrive an additional running torque.

In the range f_i from 0,37 to 0,8, it is necessary "to take a part away" from the running torque entering on the main AD of the positive feedback from half-clutch 1.

As the clutch equipped with only one, the main AD of the positive feedback exerts the maximal downforce at a minimum value of the friction coefficient, it is necessary to use an additional device generating additional downforce for realization of the first of the conditions stated above.

However the maximal downforce is exerted by the complete running torque of the AFC; therefore there is no power reserve for additional downforce generation.

An increase in the downforce generated by the main AD is possible due to the corresponding increase in CD of the positive feedback. However, in this case there will be a decrease of the value of the friction coefficient, at which there comes self-braking of friction pairs. Therefore, in this particular case an additional AD of the feedback is to block supply of a running torque to the main AD of the positive feedback, excluding at the same time self-braking of friction pairs in a particular interval of values of the friction coefficient. Therein an additional AD of the feedback, which is the single regulator for a running torque value in the specified conditions to provide loss in a value of the downforce according to the specified pattern.

IV. Discussion and Results

The investigation results of basic option of the AFC with the positive direct-action feedback showed low response accuracy and in this connection, it is supposed to introduce a feedback to a design of the additional AD without sacrificing a basic unit of the clutch.

The discovered dependence of the running torque value of the additional AD of the feedback on the current value of the friction coefficient allows one to completely stabilize the running torque value of the AFC in theory.

To satisfy our objectives it is necessary a combined scheme of the AD of the additional feedback providing supplying or taking a part away from the running torque entering on the main AD.

The results of this study can be used in the further research directed to synthesis of the basic and construction diagrams of the AFC with the positive direct-action feedback having a high response accuracy by the effect of additional control.

V. Conclusions

1. The AFC with positive direct-action feedback has not initially a high accuracy so that it cannot be used as a reliable safety device.

2. Preliminary studies showed that the response accuracy can be essentially increased by using the principle of additional control in the AFC with the positive feedback.

3. We have found the dependence of the running torque value for additional AD of the feedback on the current value of the friction coefficient whose realization shows that the AFC with the positive direct-action feedback transfers the running torque that is constant in size but not depending on the current value of a friction coefficient.

4. The realization of the AFC with the positive direct-action feedback equipped with additional control requires a combined scheme of the AD of an additional feedback which is to provide arrival of a positive feedback of the additional running torque to the main AD in a particular range of values of the friction coefficient, while in other range of values of the friction coefficient there is taking a part away from the running torque.

5. For the AFC with positive direct-action feedback, an increase of downforce of the main AD requires an increase of pressure angles of the rolling bodies, which leads to a decrease in the variation interval of the coefficient of friction within

which the change in the sign of the torque of the additional AD of a feedback is not required.

References

[1] M.P. Shishkarev, "Optimization of amplification factor of adaptive friction couplings with differentiated friction pair,", Tractors and agricultural machinery, vol. 4, pp. 36-39, 2010.

[2] M.P. Shishkarev, "Increasing reliability of first generation adaptive friction clutches," Assembly in machine building, instrument making, vol. 4, pp. 6-9, 2008.

[3] M.P. Shishkarev, "The functionality of adaptive safety friction clutches," Bulletin of manufacturing engineering, vol. 7, pp. 7-8, 2006.

[4] M.P. Shishkarev, "The backfeed characteristic refinement for adaptive friction couplings," Bulletin of manufacturing engineering, vol. 11, pp. 12-13, 2005.

[5] M.P. Shishkarev, "Distinctive features of response accuracy of adaptive friction couplings," Bulletin of manufacturing engineering, vol. 4, pp. 3-7, 2004.

[6] M.P. Shishkarev, Performance refining of adaptive friction couplings, ISBN: 978-5-89071-158-8. Rostov-on-Don: RGASHM publishing house, 2008.

[7] M.P. Shishkarev, "Theoretic grounds of the combined back feed application in adaptive friction couplings," Bulletin of manufacturing engineering, vol. 7, pp. 16-19, 2005.

[8] M.P. Shishkarev, "Mathematical models of high reliability of adaptive friction clutches," Computer-aided support for workflow, vol. 5, pp. 231-234, May 2003 [XVI Int. Sci. Conf. Mathematical methods in processes and technologies, p. 245, 2003].

[9] M.P. Shishkarev, "Mathematical models of the second-generation adaptive friction solid contact," Computer-aided support for workflow, vol. 5, pp. 72-76, May-June 2005 [XVIII Int. Sci. Conf. Mathematical methods in processes and technologies, p. 261, 2005].

[10] M.P. Shishkarev, A.A. Lushchik, A.Yu. Uglenko, Second generation adaptive friction clutches. Studies, design and calculation, ISBN: 978-5-7890-0818-8. Rostov-on-Don: DGTU publishing house, 2013 (in Russian).

[11] M.P. Shishkarev, "Theoretic grounds of the negative two-circuit back feed in adaptive friction couplings," Bulletin of manufacturing engineering, vol. 8, pp. 19-21, 2005.

[12] M.P. Shishkarev, M.D. Gavrilenko, "Definition of magnitude of coefficient of amplification of adaptive friction couplings with the positive feedback," Proceedings of the Oryol State Technical University, Series "Fundamental and Applied Problems of Engineering and Technology," vol. 6-3 (290), pp. 123-127, November-December 2011.