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I. I. ARTOBOLEVSKII

THE PRINCIPAL TASKS IN THE SPHERE OF AUTOMATION OF THE CONTROL AND DRIVE OF MACHINES

The rapid and progressive development of Soviet mechanical engineering has been possible due to automation of the technological processes of production machines and units. By extensive use of electric drive, hydraulic drive, automation and remote control facilities, Soviet scientists and engineers, in collaboration with production innovators, have during the past few years created many new designs of electrically driven high-production machines, including machines with automatic electric, hydraulic, and electronic operation, new types of automatic machine tool lines, and the first automatic plant in the world. A general theory of electric drive and self-regulation, which generalizes and guides practical activity in this field of engineering, has been developed in the USSR.

Despite the success achieved, however, there are still serious shortcomings in the matter of development of drive and automatic operation. For example, questions of coordination of the scientific research work being performed by various establishments of the Academy of Sciences of the USSR, the branch scientific research institutes, universities, and industrial enterprises in the sphere of application of automatic operation facilities in mechanical engineering have not been satisfactorily stated.

This condition leads to duplication of effort in the work of scientific institutions and causes non-productive expenditure of government funds. In certain scientific research institutes and in individual branches of industry, technical achievements in the field of drive and automatic operation, regardless of their great practical significance, remain little-known and are not utilized in other branches of industry. The scientific research organizations of the Academy of Sciences not infrequently scatter their attention over petty, unimportant subjects the elaboration of which could be accomplished with greater success and in less time by the branch scientific research institutes or directly by the industrial enterprises themselves. At the same time, one occasionally encounters in the subject matter of the branch institutes problems which only the scientific establishments of the Academy of Sciences of the USSR are competent to solve.

Thus far there has been no clear-cut perspective plan of the development of questions of drive and automatic operation in mechanical engineering, a plan pointing out the path which the progressive development of automation of technological processes must henceforth follow.

The party and the government devote an enormous amount of attention to strengthening the ties between science and production. Thus far, however, a number of scientific research organizations have not been bound closely enough to practical work. As a result, many scientific research works in the field of drive and automatic operation on which large sums of money are spent prove unsuitable and useless at the first attempt to put them to practical use. Such works are usually consigned to the files, while a large number of valuable proposals, theoretic works, and scientific discoveries are not adopted in production as a result of underestimation of their significance by practical workers.

A serious shortcoming in the matter of development of drive and automatic operation is the lack of due attention on the part of the electric industry to problems of equipping mechanical engineering with what it requires in the form of electric machines, special types of machines, electric equipment, control devices, and complex devices for automatic and remote control. The Ministry of the Electric Industry has refused even to participate in a conference, and the lectures on the production of electric equipment for mechanical engineering by the plants of this Ministry, projected by the organizational committee, have not been given.

A special section of the conference on automation of the drive and control of machines has been prepared in order to mark out the way towards elimination of the shortcomings listed above and to formulate the principal tasks and directions of further improvement and development of automatic drive and automatic control systems, as well as to throw light on existing achievements in this field of engineering. The program of this section includes lectures embracing the following basic questions of automation of the drive and control of production machines:

1) the problem of regulation of electric drive by the introduction of ion-electron control;

2) the sequence of switching on current in the systems of electric tracking drives, as applied to duplicating metal-cutting machines;

3) problems of drive automation and its prospects;

4) the basic tasks in the field of development of automatic hydraulic drives,

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including high-pressure hydraulic drives;

5) problems related to cam drives for automatic machines and lines.

P. V. MARKIN

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ION-ELECTRON DRIVE OF METAL CUTTING MACHINES

Electronics is playing an increasingly greater part in the development of automation of production processes. It is employed for purposes of direction of production processes, control, regulation, and various measurements. Electronic devices, since they are highly sensitive, insure that specified operations will be performed with great accuracy.

Electronic devices in systems of automatic drive and direction of machines are in the majority of cases the connecting link between the directing member, i.e., an emitter, and the performing member (relay, switch, electric motor, electromagnet, electromagnetic coupling). The use of electronic connections eliminates the necessity of employing complicated mechanical or other transmissions, reduces the amount of switch-relay equipment, and insures continuity of direction. The electronic devices themselves are simple in production, since they are made up chiefly of finished standard serially or mass-produced articles.

The characteristics of electronics mentioned determine the extensive possibilities for its utilization for automation of the technological processes of machining metals in relation to any parameters: course, speed, time, stress, form, and dimensions of machined parts. Thus, for example, electronic devices are successfully employed for automatic direction of the feed drive in duplicating machines, automatic control of the dimensions of an article in the process of its machining on circular grinding and honing machines, stress control in deep drilling machines and in the electric shaft system in heavy screw-cutting lathes, where the thread-cutting operations are performed without a rigid kinematic connection between the spindle and screw-cutting rest, whereby accuracy of thread cutting and simplification of the kinematics of the machine are insured.

Electronic devices have been employed on a large scale in automatic machine lines. The automatic plant build in the USSR for the production of automobile pistons is equipped with a number of electronic devices for various purposes.

An adjustable ion drive, which, combined with a highly-sensitive system of electronic direction, can perform various operations with great accuracy, including automatic stabilization of fixed values of adjustable parameters and their automatic change in accordance with a fixed rule, occupies a prominent position among the var-

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ious electronic devices. Such a system of drive makes it possible to accomplish any rate of speed of the driving gear, including automatic rapid changing of speeds while a machine is running, smooth change of speed or maintenance of its stability in the process of machining a part.

As we know, operation with the natural mechanical characteristics of an ion drive is unsuitable for the mechanism of a machine in consequence of their insufficient rigidity. A greater rigidity of the mechanical characteristics is achieved by self-stabilization, which is accomplished through regenerative couplings by speed or by voltage and current. The possibility of changing speed within wide limits (by tens and hundreds of times) is insured in the same way.

The system of continuous electronic direction of a drive makes it possible to obtain the most favorable character of the course of transitional processes on the score of their rapidity, smoothness, absence of abrupt dynamic shocks, etc. The considerable reduction of the number of gear transmissions, and occasionally even their elimination entirely, achéeved through the use of automatic electric regulation in place of stepwise changing of speeds, promotes increase in the vibration stability of a machine, and consequently also improvement of the quality of machining of parts.

In the solution of the problem of creating the most rational system of automatic electric drive, the most tempting idea is that of creating an adjustable alternating current electric drive with an ordinary asynchronous electric motor with a short-circuited rotor. As we know, this type of motor is the simplest, most inexpensive, and most reliable machine in operation. However, repeated attempts to create such a drive have not ended in positive results either in the USSR or abroad.

The drive systems proposed by various authors have probed very complicated or not sufficiently perfected in their performance features. For this reason, the problem of creating an adjustable alternating current ion drive remains very pressing and requires the most serious attention.

At the present time, the basic form of adjustable drive is still the electric drive with direct current motor. The methods of regulating the speed of such an electric motor in an ion drive system may be very diverse; for example, by change of the armature voltage, change of excitation flux, or a combination of both in one drive. Cases are known of employment of electric ion drive with rheostat regulation. For example, in one foreign upright milling machine, the armature and exciting winding of the electric motor of the spindle drive are fed through one uncontrolleSTATon rectifier, and the motor speed is regulated by a rheostat in the armature circuit. The primitiveness and shortcomings of such a drive system are obvious.

In certain foreign machines, regulation of the speed of revolution of the motor is accomplished by change in the magnitude of anode voltage of an uncontrolled rectifier which feeds the motor armature. This method of regulation has the advantage that the power factor of the electrid drive does not drop as sharply with reduction of speed as it does in the case of grid regulation of a rectified voltage. However, it also has serious shortcomings, which are determined by the necessity of performing control of the drive in the power circuit. In this case it is necessary to use cumbersome equipment; serious difficulties arise in achieving mechanical features of the specified rigidity; and there is no possibility of more flexible control of the transitional processes.

The best operating qualities of an adjustable electric drive are achieved through the use of controlled ion devices which feed the armature of the electric motor. Both controlled and uncontrolled discharge-tube or selenium rectifiers are employed in this drive system for feeding the exciting winding, depending on specific requirements.

The principal disadvantage of an adjustable electric ion drive with grid control of rectified voltage feeding the motor armature is the sharp drop in the power factor of the drive when the motor speed is reduced. This is explained chiefly by the fact that drecrease in the rectified voltage fed to the armature of a motor is achieved by increasing the ignition angle of the ion devices of the rectifier, this resulting in a corresponding increase in the angle of displacement of the main impulse of the rectified current with respect to the anode voltage. If, however, we take into consideration the fact that even in an electric drive with a mechanical transformer on the generator - motor (G - M) system, the power factor and the efficiency also drop sharply with reduction of speed and that its efficiency is less than with ion drive, then we see that, as studies of the Experimental Scientific Research Institute for Lathes demonstrate, the ion and mechanical drive systems are approximately equivalent in their power features.

It is convenient in this instance to make use of a generalized power index -- the ratio of the useful power of a drive P_2 to the full apparent power P_k.

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Figure 1. The functions $\frac{P_2}{P_k} = f(M)$ for ion and mechanical electric drive: 1 -- ion drive; 2 -- drive on the generator - motor system.

Figure 1 shows the function $\frac{P_2}{P_k} = f(M)$, obtained by experimental compuation, for both drive systems, with the same running 15-kilowatt electric motor. As may be seen from the illustration, in operation at higher speeds this generalized power index is better with the ion drive, and at lower speeds with the mechanical drive. The curves of the relation of $\frac{P_2}{P_k}$ to the speed and load of the electric motor show that the power indices of the drive in this instance should be estimated with averaged efficiency (weighed mean values) and power factor.

Another disadvantage of the ion drive with grid control is the presence of current pulsations in the armature circuit which cause increased heating and greater power losses in its windings and supplementary fields. According to data from studies of the Experimental Scientific Research Institute for Lathes, in a drive with a two-phase rectifier the ratio of the effective value of the rectified current in the main circuit of an electric ion drive having a power of 0.25 to 60 kilowatts, 2200 -- 2800 rpm, to the average value of the current during operation with a nominal load is 1.4 -- 1.5. The magnitude of this ratio decreases with reduction of the nominal speed of revolution of the electric motor to 1.12 -- 1.2 (when $n = 600$ -- 800 rpm). In a drive with a three-phase rectifier, this magnitude varies from 1.3 to 1.1, depending on the running speed of the electric motors, and when the number of rectifier phases $m = 4$ or more, it does not exceed 1.1. With reduction of load, the ratio of the effective value of the current to its average value increases, reaching 1.6 to 1.7 with an idling motor with a two-phase rectifier. Owing to this, the relative value of power losses in the armature circuit increases, while these losses decrease in absolute magnitude.

The presence of current pulsations in the armature circuit when there is a small number of rectifier phases, this being common in low-power drives, makes it necessary either to use a cathode choking coil to smooth out the pulsations or to select an electric motor with an appropriate power reserve. In high-power drives the influence of current pulsations on heating and losses in the machine are comparatively small, insofar as multiphase rectifiers and comparatively slow-speed motors are employed for these drives.

One of the important advantages of ion drive as compared with a drive on the G - M system is the relatively simplicity of its electric equipment. The ion drive has only one static electric transformer consisting of ion devices and transformer, while in mechanical drive the transformer consists of two revolving machines -- an asynchronous electric motor and a generator which twice transform one type of energy into another.

Besides the operating superiorities of the static ion transformer -- absence of noise and vibrations, simplicity of maintenance, higher efficiency -- it also differs favorably from the electromechanical transformer by the initial expenditures. In serial production of ion devices the expenditure of labor for the production of ion transformers and the amount of materials expended in this process (copper, steel) are far less than for a mechanical transformer.

The limited experience in utilization of ion drives and the absence of established types of ion devices for them do not permit of a proper evaluation of the reliability of their operation and of the cost of employing them. However, even from this aspect, to judge by available disconnected data concerning periods of operation of individual ion devices, both with a fluid cathode (mercury rectifiers, ignitrons) and with an incandescent one (thyratrons, gas rectifiers), the economic suitability of electric ion drives can scarcely be doubted, on the condition of proper quality of production and further perfection of ion devices.

The field of rational employment of adjustable ion drives in machine tool construction includes a wide group of machines differing in purpose, dimensions, and technical characteristics. Particularly effective use may be made of ion drive in automation of metal machining processes in specialized machine tools. For example,

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in turning lathes, vertical lathes, and grinding machines, it is possible by smooth regulation of the speed of the spindle to maintain a constant economic cutting speed upon change of machining diameter; in milling machines it is possible to achieve automatic regulation of feed depending on the load of the tool, which changes with change in the depth or width of milling or in the event of nonuniformity of the material; in centerless honing machines it is possible to insure automatic maintenance of the speed of revolving parts, etc.

No less important is the use of adjustable ion electronic drive in universal lathes, the automation of which presents a particularly urgent problem. Electric ion drive with electron control can insure various operating conditions, exceptional flexibility, and simplicity of control on these machines.

The employment of automatic drives for the main driving mechanisms of machine tools, as is well-known, does not meet the requirement of rational utilization of the operating electric motor, regulation of the speed of which is performed under constant momentum; regulation at constant power, which is required in the drives of such mechanisms in order to obtain an economical cutting speed upon change of diameter and nature of machining of an article, makes it necessary to increase the over-all dimensions of the electric drive motor. In consequence of this, the practically acceptable regulation range in the drives of main driving mechanisms, in the event ion drive or any other drive with automatic regulation is employed, is limited, being approximately three to five for heavy machine tools and slightly more for medium and light machines.

In a number of cases where it is necessary to achieve drive operating conditions at reduced power and at low speeds, for example, in increased accuracy threading on screw-cutting lathes or on machines designed for tool operations, the utilization of automatic regulation proves to be advantageous.

However, the advisability of employing ion or direct-current mechanical drive for main driving mechanisms is frequently determined not so much by automatic regulation as by other factors, chiefly the requirements of transitional processes.

The requirement of rational utilization of an adjustable motor is met the most fully in the drives of machine tool feed mechanisms; the magnitude of the torque required, which is determined by the forces of friction, remains approximately constant at all speeds. In this instance, it is the most effective to utilize automatic ion drive with any large adjustment range. STAT

Electric ion drives of two main groups are being employed at present in industry: high-power drives, on the order of hundreds and thousands of kilowatts, used in the metallurgic, coal, paper-making, and other industries, and low-power drives, on the order of several kilowatts, used chiefly in machine tool construction. Mercury rectifiers with controlling grids are used as controlled ion devices in high-power drives, and thyratron rectifiers in low-power drives.

In the electric drives with thyratron rectifiers for various powers from 0.25 to 3 kilowatts of the ELIR type, developed in the Experimental Scientific Research Institute for Lathes, rigid mechanical characteristics of the ELIR drive are achieved only by means of regenerative voltage and current couplings of the armature of the electric motor; this is fully adequate for the overwhelming majority of machine tools. In this case the necessity is eliminated of employing a tachogenerator, which is particularly undesirable in small universal lathes.

The ELIR electric drives have been employed in grinding machines, turning lathes, milling machines, jig-boring and other machines.

The setting of specified speeds or feeds in these machines is accomplished either manually, by means of a potentiometer equipped with a graduated scale, or automatically, by switching to previously determined fixed speeds corresponding to the specified operating conditions of the machine. The speed adjustment range of the drive motor in these machines ranges from 8 (in main driving mechanisms) to 120 (in feed and rapid and slow transfer mechanisms). Besides the convenience of control and other operating advantages, the ELIR drive also possesses important qualities of design. This drive, unlike the electromechanical drive which requires special space and base for its transformer, consists of separate, compact units convenient for installation in the machine or in the control cabinet.

Figure 2 shows an example of installation of ELIR drive units in a universal circular grinding machine. The motor which rotates the article is located directly in the body of the stock of the article (upper left), on the front of which is located a dial and rotating handwheel built into the body of the regulating rheostat. In the niche in the right pedestal is a panel with the thyratrons and elements of the electronic control circuit. In a similar niche in the left pedestal is the power transformer. Figure 3 illustrates the installation of an ELIR panel (upper left) and power transformer (bottom left) in the common electric control cabinet of a centerless lapping machine. There are also examples of installation in a com STAT electric control cabinet of two and three ELIR drive panels and transformers which serve a corresponding number of electric drive motors of individual machine mechanisms.

The limited power of the ELIR-type drives being manufactured (up to 3 kilowatts) is explained by the lack of thyratrons of appropriate power. For high-power drives it is necessary to employ fluid cathode ion devices (controlled mercury rectifiers, ignitrons) which have a greater emissive capacity in contrast to the incandescent thyratron cathodes.

Adjustable electric ion drives of medium power (from 10 to 100 kilowatts), which are of great interest to machine tool construction, are for the present not being practically employed in the machinebuilding industry. There are only a few examples known of the use of such drives on individual foreign metal-cutting machines, which are experimental models.

Figure 2. Installation of an ELIR drive unit in a circular grinding machine.

As a result of the lack of medium-power adjustable ion drives, ion devices of appropriate power and purpose have thus far not been developed. The existence of a number of undoubted advantages of the adjustable ion drive make it possible, however, to consider it expedient to utilize it for medium-power machines also.

One of the attempts to create an ion drive for such machines is the development and production in the Experimental Scientific Research Institute for Lathes of an experimental electronically-controlled ion drive with a power of up to 30 kilowatts. Ignitrons are used in the drive of the machine as adjustable ion devices. Mercury rectifiers with a controlling grid are not being produced at present for STAT power range. Only very recently has the electric industry begun the utilization of such devices.

Figure 3. Installation of an ELIR drive panel and power transformer in the common cabinet of a centerless lapping machine.

A three-phase ignitron rectifier with an independent ignition circuit consisting of condensers and thyratrons is employed in the electric ion drive produced by the Experimental Scientific Research Institute for Lathes. Ignition of the ignitrons is accomplished by current impulses which originate upon discharge of the condensers at the moment of delivery of opening potential to thyratron grids, which are connected in series by their anode circuits with the ignitron igniters. The amount of rectified voltage and the speed of revolution of the motor armature is regulated by an electronic device which controls the moment of ignition of the thyratrons and, consequently, of the ignitrons during each positive half-period of anode voltage. The regenerative speed coupling employed in the circuit which is effected by means of a tachogenerator connected with the motor shaft insures that rigid mechanical characteristics will be obtained within a wide range of motor revolution speeds.

Control of starting and stopping of the motor is accomplished in the electronic control and ignitron ignition circuits when commutating devices are absent from the main circuit.

Owing to the negligible amount of power needed by the grid circuits of the thyratrons, great flexibility of control and compactness of equipment are provided.

Figure 4 gives an overall view of an ignitron rectifier installed in a separate cabinet, with all the elements of an ignition and electronic control circuit.

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Figure 4. Overall view of the control cabinet of a drive with a power of 30 kilowatts:

1 -- ignitrons; 2 -- ignitron ignition block; 3 -- electronic control block.

The results of tests of an experimental model of a drive with an ignitron rectifier on a stand and on a machine have demonstrated that its operation is fully satisfactory.

With changes in the amount of torque on the shaft of the driving motor from $0.2\ M_{nom}$ to $1.25\ M_{nom}$, the deviation of the number of revolutions per minute of the motor with respect to the average established value n_{sr} does not exceed $\pm 2\%$ when $n_{sr} = n_{nom}$ -- $0.1\ n_{nom}$, $\pm 4\%$ when $n_{sr} = 0.1\ n_{nom}$ -- $0.04\ n_{nom}$, and $\pm 8\%$ when $n_{sr} = 0.04\ n_{nom}$ -- $0.02\ n_{nom}$. The system insures starting and reversing of the motor with automatic limitation of torque and armature current to a previously established limit, as well as stable operation under all specified revolution speeds and loads.

Experience in the operation of ELIR ion drives on machine tools has confirmed their excellent qualities. However, they have not as yet been employed extensively in machine tool construction due to insufficient reliability in operation. STATgether with a large number of instances of fully satisfactory continuous operation of the ELIR drive on a number of machines, there are many cases of interruption of its operation and of its becoming unserviceable. One of the causes of such a situation is defective design, quality of manufacture, and installation of the apparatus. However, by their nature these defects may be easily eliminated. In addition to this, in many cases unsatisfactory performance of a drive is explained by the lack of fully competent personnel in the places of operation and by improper operation.

The most important cause of insufficient reliability of the ELIR drive in operation is poor quality thyratrons and hot-cathode rectifiers. The results of tests and observations in operation indicate that in addition to those of the devices which are efficient and well-made, there are very many which are either unserviceable or have a very short service life. The nomenclature of the devices being produced is also completely unsatisfactory. It is very limited and does not meet the technical requirements of electric ion drive.

Decisive measures are needed for improvement of the quality and expansion of the nomenclature of ion devices for drives, without which it is impossible to count on extensive introduction and further development of this progressive drive system. Together with this, it is necessary to amplify work towards perfection of the diagrams and designs of adjustable ion drives, chiefly in the direction of simplification and of increasing their reliability in operation.

A particularly important problem in this process is the creation of a simple and reliable adjustable alternating current ion drive with a short-circuited asynchronous motor; its solution will require unremitting effort on the part of the research organizations.

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A. M. KHARITONOV

MULTISPEED ELECTRIC MOTORS IN MACHINE DRIVES

Three-phase asynchronous motors with short-circuited rotor represent the basic type of electric machines employed for the electric drive of machine tools. Besides such qualities as simplicity of design, reliability in operation, etc., they also have a serious defect -- poor properties of regulation.

As we know, the speed of revolution of an asynchronous motor may be regulated by changing the sliding or speed of revolution of the magnetic field.

The first is possible only in the event of existance of loading moment on the shaft of the motor and is usually achieved by switching resistance into the rotor circuit.

The latter is accomplished by changing current frequency or the number of poles of the motor winding.

Regulation of the speed of revolution of motors by changing frequency requires the presence of a special grid the current frequency in which may be changed by stages or smoothly. Such a system of regulation is encountered in a number of special-purpose electric drives (roller beds, continuous spinning machines, high-speed drives, and others).

The number of poles of the motor winding may be changed only by stages, and, consequently, the speed of the motor may be only thus regulated. Owing to its simplicity, this is the most widespread method in universal electric drives.

The number and arrangement of the poles of asynchronous motors is determined by the electric connection of the stator winding. When there are separate, mutually independent windings with varying numbers of poles on it, it is possible to obtain different speeds of revolution of the motor. In asynchronous motors with one stator winding, change in the number of poles is achieved by reversal of the direction of the current in the winding or by changing the arrangement of the winding phases on the circumference of the stator bore. Multispeed motors usually have 2, 3, or 4 different speeds.

Table 1 gives the practically possible versions of multispeed motors.

Single-winding multispeed motors have smaller overall dimensions as compared to the two-winding motors, which correspond to them in power and speed of revolution. With the same dimensions, the power of the former may exceed by approximately 1.6 to

2 times the power of the two-winding motors with higher power indices and lesser frequency of starting current. The connections for switching the windings for different speeds are simpler in motors with two windings than in motors with one.

Table 1.

Method of regulating motor speed	Number of speed stages	Speed regulation limit
One winding with reversal of poles at ratio of 1 : 2	2	1 : 2
Two independent windings for different number of poles	2	Depends on number of poles of each of the windings
Two independent windings, one of which has reversal of poles at ratio of 1 : 2	3	Same
Two independent windings, each with reversal of poles at ratio of 1 : 2	4	"
One winding with reversal of poles for 2, 3, or 4 speeds	-	Depends on number of poles for which winding is reversed. In majority of cases not more than 1 : 4.

With feeding of current of normal frequency, the following synchronous speeds of revolution of the motor are possible: 3000, 1500, 1000, 750, 600, 500... rpm, which correspond to 2, 4, 6, 8, 10, 12... poles of its winding. For multispeed asynchronous motors having a power of up to 100 kilowatts, the speeds are usually limited to 3000--500 rpm with 2, 3, or 4 stages.

Many of the qualities of the ordinary short-circuited asynchronous motor are preserved in motors in which the number of poles are switched, this insuring it extensive application in machine tool construction.

EMPLOYMENT OF MULTISPEED MOTORS IN THE ELECTRIC DRIVES OF VARIOUS MACHINE TOOLS

Multispeed asynchronous motors are employed in various branches of industry and transport: in the coal, metallurgical, cement, oil, shipbuilding, and food industries; in hoisting, pumping, and crane installations; on conveyors, elevators, etc.

The extensive use of multispeed motors plays an important part in the further development of machine and machine tool construction. When single-speed motors are replaced by multispeed motors on machine tools, it frequently becomes possible to:

1) simplify the designs of machine tool transmissions, including even elimi-

nation of gear and feed boxes;

2) increase the output and performance qualities of machine tools;

3) improve the quality of machining on a machine tool owing to reduction of vibrations and inaccuracies of the mechanisms resulting from the presense of a large number of gear couplings;

4) increase the efficiency of the machine tool as a result of reduction of elements of the kinematic chain;

5) change speed during operation, without stopping the machine;

6) simplify automatic control of the processes of starting, stopping, reversing, and braking;

7) simplify automatic control of machining conditions in relation to technological factors.

Electric drives in which multispeed motors are employed instead of single-speed motors possess other advantages as well, e.g., the possibility of starting a machine at a minimum speed of revolution of the motor and switching the windings during operation to higher speeds; this is of particular importance in the acceleration of drives possessing considerable inertia. Starting a motor at a lower speed of revolution also has the advantage that the absolute value of the starting current in this case will as a rule be the least as compared with the starting currents at lower speeds. The losses of electricity in the case of switching from a lower to a higher speed will be much smaller in importance than in the case of direct starting at increased speed. When a winding is switched from a smaller to a larger number of poles, i.e., when the speed of the motor is reduced, its recuperative braking, which does not involve power losses, takes place automatically, as in the case of opposition braking.

Multispeed motors may be used extensively for universal and special machines -- lathes, turret lathes, drills, grinders, planers and shapers, tool-grinding machines, etc.

In addition to their use in main movement and feed drives, multispeed motors may also be utilized to accomplish accelerated or decelerated shifting of various machine mechanisms (rapid delivery and removal of a tool, softening the blow at the moment of indexing, etc.) and performance of technological operations connected with change in the speed of machining or with the necessity of correction of a tool (e.g., straightening the grinding wheel), without stopping the machine.

Replacement of the ordinary single-speed motor by multispeed motors in many instances substantially improves the technological and performance qualities of the machine and reduces the amount of labor expended in its manufacture.

It is expedient to employ multispeed motors in the feed drives of woodworking machines, since in the majority of cases three or four deliveries are adequate; this may be obtained with a multispeed motor without any auxiliary mechanical devices.

In deep regulation of the speeds of universal metal-cutting machines, particularly with a small denominator in the progression of the geometrical series of the number of revolutions per minute, reduction gears or transmissions with a large number of stages are required. When speed is regulated by only one mechanical means, the transmissions become considerably more complicated in design and at times require a complicated control system, this increasing the expenditure of labor and rendering the production of transmissions expensive. For this reason, combined electromechanical speed regulation systems may be employed more extensively in machine tools.

Partial or complete replacement of mechanical by electric speed regulation of a machine tool becomes possible when a multispeed asynchronous motor is employed in the drive of the machine. In machine tools in which it is possible to limit the speed stages to 2, 3, or 4, on the condition that the speeds of revolution of the machine and motor spindle be equal, the most effective use is that of built-in multispeed motors. The stator of the motor is built into the headstock of the machine, and the spindle is connected through a coupling with the rotor shaft or the latter is mounted directly on the spindle. Such a design of the machine proves exceptionally simple, and its kinematic chain is the shortest. If the speeds of revolution of the spindle and the multispeed motor do not coincide, then the latter is connected with the spindle by means of a belt or gear drive. Machine tools with built-in multispeed motors should be particulaмly widespread among small operation machines designed for turning, milling, drilling, and grinding operations.

The installation of a simple gear train greatly extends the speed regulation range of machine tools with built-in multispeed motors, lengthening the kinematic chain of the machine only at low speeds. The arrangement with the gear train may be employed for machine tools in which the cutting speeds differ considerably from one another, for example, in turning and thread cutting, in drilling and reaming. It is

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also fully possible to employ a multispeed motor in combination with a two-stage gear train or with a transmission.

Multispeed motors for machine tool drives may be effectively utilized for the drives of machine tools in combination with a mechanical speed modifier, sliding coupling, and other types of stageless regulators. In these cases, not only are the regulation limits extended, but also the energy losses are reduced which occur in a combination of asynchronous electric motor and sliding coupling.

The development of designs in machine tool construction is following the path of increased machinang speeds. In the drives of machine tools and machines with speeds of revolution above 10 to 15 thousand rpm, the greatest effect is provided by high-speed (high-frequency) short-circuited motors fed an increased frequency current from a special generator or frequency changer. Regulation of the speed of revolution of these motors increases considerably the universality of the drive, since it makes it possible to change the speed in relation to the requirements of the technological process, quality of the material, dimensions of the tool, etc. In addition to this, in the case of high-frequency motors with speeds on the order of 40 to 50 thousand rpm and above, stepwise increase of the speed of revolution makes it possible to solve the most simply the problem of starting the motor.

Smooth regulation of the speed of revolution of a high-frequency motor is achieved by changing the frequency of the converter (generator), and stepwise regulation may be accomplished by three methods:

1) stepwise change of the speed of the drive of the converter, and, consequently, its frequency;

2) change in the number of poles of the motor winding;

3) a combination of the first and second methods.

The simplest and most reliable method of stepwise change of the speed of the drive of a frequency changer is the employment of a multispeed short-circuited motor rigidly connected with the changer. If the drive motor has n_1 speeds of revolution, then the number of different current frequencies which may be obtained from an asynchronous frequency changer when it is rotated by a multispeed motor in and counter to the speed of revolution of the magnetic field of the changer will be $2n_1$. When the windings of the high-frequency motor are switched to n_2 poles, we obtain n_2 speeds of revolution of the motor. Consequently, the total number of speed stages of the high-frequency motor in this case will be $2n_1n_2$. When, for example, a two-speed motor in the drive of a frequency changer and a three-speed motor are employed, 2x2x3 = 12 different speeds of revolution may be obtained. With a three-speed frequency changer drive motor and a three-speed high-frequency motor, 2x3x3 = 18 different speeds of revolution may be obtained, etc.

CONTROL OF MULTISPEED MOTORS

Control of multispeed motors is accomplished by means of manual and contactor control systems or by a combination of the two.

Manual control devices are employed in the case of infrequent starting for multispeed motors of comparatively low power (3 to 5 kilowatts). The utilization of these devices is limited by the current which is broken by their switches. If the poles of the motor winding are reversed, when the motor is switched off from the circuit, then a safe continuous current switch may be employed for the more powerful motors.

For the control of multispeed motors of medium power (5-10 kilowatts), use is usually made of a system wherein the motor is switched on and off by means of a contactor and the poles changed by means of a changer with the motor switched off. When the changer knob is turned to the new position, the normally closed changer block switch, which is connected in series with the circuit of the contactor coil, opens, in consequence of which the latter is cut off from the system.

Such a system for connection of the VII changer together with a magnetic starter is illustrated in Figure 1.

Contactor control of multispeed motors insures a high frequency of switchings on and off (occasionally as many as several hundred an hour). It is more reliable in operation in comparison with manual control equipment. Thus the average number of switch-ons of the changer due to mechanical wear of the contacts amounts to approximately two hundred thousand, and in contactors, one million.

Automatic control of multispeed motors is accomplished with magnetic starters or contactors. For low-power motors use is frequently made of electromagnetic alternating current relays (EP, RPM, etc). Occasionally, automatic control of multispeed motors is effected by means of special control devices. For example, in the turret lathes designed and produced by the Experimental Scientific Research Institute for Lathes, the control device which performs changing of the poles of the

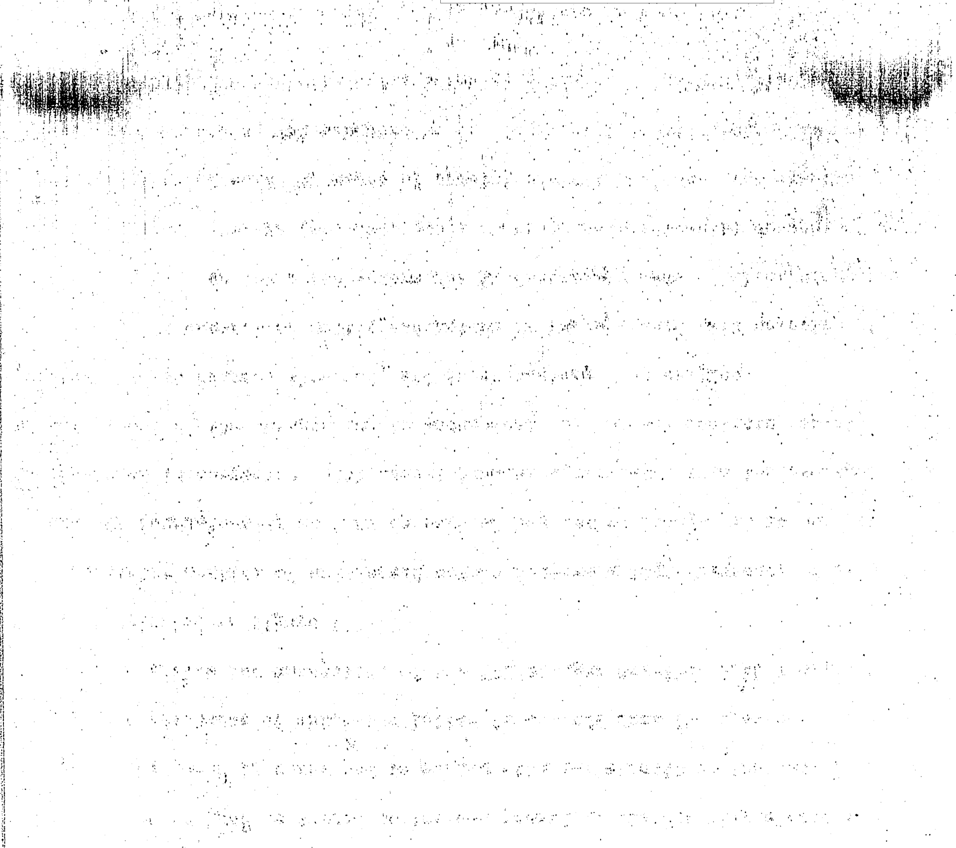

Fig. 1. Diagram of connection of the VII universal pole changer together with a magnetic starter. 1 - Start; 2 - Stop; 3.- Connection of switches; 4 - Closed; 5 - block contact; 6 - handle; 7 - Closing of switches; 8-Switches.

three-speed motor (750/1500/3000 rpm) in the main movement drive is mechanically connected to the turret, turning of which simultaneously causes change by the control device of the motor winding to the speed corresponding to that of the part machining process (turning, drilling, threading, etc.). The moving part of the dontrol device is a cylinder with drums arranged uniformly around its circumference. Each of them has several fixed positions. The number of drums corresponds to the number of positions of the turret. Terminals are led to the fixed switches of the control device from the winding of the drive motor. The latter, depending on the tool mounted in the turret, develops the required speed, owing to preliminary setting of the drums, which pass in succession under the fixed switches of the control device and close them.

SERIES T MULTISPEED SINGLE-WINDING MOTORS

The factories of the Ministry of the Electric Industry at the present are producing multispeed motors, on the basis of a single series, A, with change of the

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winding to 12/6, 8/4, 6/4, 4/2, 6/4/2, 8/6/4, and 12/8/6/4 poles. The two-speed motors, in which the ratio of the number of poles is not 1:2, as well as the three and four-speed motors, have two stator windings, and, because of their overall dimensions, are in many instances unacceptable.

A series T of multispeed motors with one stator winding, including six standard dimensions of motors having a power of 1.7, 2.8, 4.5, 7, 10, and 14 kilowatts at 1500 rpm, made with three different external diameters of the stator plate, 182, 245, and 327 millimeters, was developed in the years 1948 and 1949 at the Experimental Scientific Research Institute for Lathes for machine tool construction.

The packs of metal plates for each diameter are produced in two different lengths. The dimensions of the cog zones of the stator and rotor, the number of grooves of the latter, and a number of other features of the design of the motors have been selected with a view toward achieving optimum use of them as multispeed motors.

The series T motors in constructional execution are closed with external ventilation. In motors up to and including 7 kilowatts in power, the bed is formed by casting around the stator pack with an aluminum alloy. The shields, fan, and other parts are also made of an aluminum alloy.

Four types of motors are being serially produced at the present time: the T-41, T-42, T-51, and T-52, which in their overall and installation dimensions fully correspond to the similar dimensions of the motors of the single series of A types: AOL-41, AOL-42, AOL-51, and AOL-52.

The basis for determining the power of the motors was the extent to which the stator temperature might be safely exceeded or the conditions for obtaining favorable starting characteristics.

All series T multispeed motors are made with one stator winding, owing to which they have greater power than the two-winding motors of types A and AO, although they are of the same dimensions as the latter. Depending on the interrelations between the speeds, the two-speed series T motors have from 6 to 14 terminals to the pole changer, the three-speed motors from 9 to 18, and the four-speed motors, up to 22.

The power range of the series T motors is shown in Table 2.

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Table 2

Number of poles	Power range (in kilowatts), the number of pole pairs being				
	12	8	6	4	2
4/2	--	--	--	1.7--6.5	2.0--7.0
6/2	--	--	1.0--5.5	--	1.3--6.5
6/4	--	--	1.0--5.0	1.1--5.0	--
8/2	--	0.7--2.7	--	--	2.0--6.0
8/4	--	0.7--3.0	--	1.2--4.5	--
6/4/2	--	--	1.3--4.5	1.5--4.5	1.8--5.0
8/4/2	--	0.7--2.7	--	1.7--5.5	2.0--6.0
8/6/4	--	0.7--2.7	0.8--3.2	1.2--4.0	--
8/6/4/2	--	0.7--2.7	0.8--3.2	1.2--4.0	1.5--4.5
12/8/6/4	1.4--1.7	2.5--3.0	3.0--3.5	4.0--4.5	--

Series T multispeed motors are employed by 35 machine tool construction plants in more than 60 models of metal-cutting and woodworking machines.

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N. I. LEVITSKII

CAM DRIVE IN AUTOMATIC MACHINE TOOLS
AND LINES

THE CONDITIONS WHICH DETERMINE SELECTION
OF THE LAW OF MOVEMENT OF THE DRIVEN MEMBER
IN IN THE CAM GEARS OF AUTOMATIC MACHINES

Cam gears are among the most widespread; this is explained by the number of their positive qualities as compared with other machines.

The merits of cam gears are (1) the possibility of effecting almost any law of movement of the driven member by means of appropriate profiling of the cam; (2) achievement of a high performance due to rational selection of the law of movement of the driven member; (3) changing the law of movement of the driven member by replacement of the cam; (4) small overall size of the gear; (5) simplicity of performance of coordinated operation of several gears in automatic machines. For this reason, cam gears have been extensively employed in the feed mechanisms of automatic metalworking machine tools, in the displacement mechanisms of the operating members of various automatic machines and in many other instances when it is necessary to obtain reciprocating rotary or rectilinear motion of the driven member in accordance with a specified law.

The disadvantages of cam gears as compared with crank-lever gears and several others are the considerable amounts of specific pressures on the contact surface of the members, and, in consequence, increased wear of the friction surfaces and shortening of the life of the gear. Moreover, in the event of high speed of movement of the driven member, there exists the possibility of occurrence of impacts, particularly if provision has been made only for power locking. Usually these disadvantages may be reduced to a minimum by proper selection of the law of movement of the driven member and of the basic parameters which determine the arrangement and structural shaping of the gear.

In this article, consideration has been given only to selection of the law of movement of the driven member, which selection must be accomplished in such a way that the following conditions are fulfilled:

1) correspondence of the law of movement of the driven member to the requirements of the technological process; 2) achievement of high machine efficiency;

3) minimum expenditure of energy for setting the gear in motion; 4) assurance STAT of durability and long life of the gear; 5) simplicity of production of the profile of the cam.

<u>The first condition</u> is the basic one, since the gear must above all suit its functional purpose, i.e., displacement of the operating member connected to the driven member of the gear must insure the performance of a specific technological operation and proper quality of the latter. This condition may be satisfied by selection of the law of movement of the driven member, this law being specified in certain instances by the technological process during a certain portion of the movement. For example, in feed mechanisms there is frequently need of uniform displacement to a certain length during a specified interval of time, which is determined by the height of the speed of movement of the driven member or by the requirements of coordinated operation of the mechanisms. In many instances, however, the requirements of the technological process determine only certain individual parameters of the law of movement of the driven member. These parameters include: the amount of total displacement of the driven member (in one direction), the maximum speed and maximum acceleration, as well as the amount of speed and acceleration of the driven member which correspond to specific positions of the latter. Assignment of maximum speeds and accelerations of the driven member is necessary because increase in the speed or acceleration of the operating member of a machine above the maximum may cause distubance of the technological process or lowering of its quality. In addition, one may add to the parameters to be assigned the full time of movement of the driven member and the time of individual stages of its movement (for example, time of accelerated or uniformly retarded motion, time of shutdowns, etc.). This time is occasionally determined only by the sequence of completion of individual stages of the technological process and by the requirements of coordinated operation of the mechanism.

<u>The second condition</u> - achievement of high machine efficiency - is also related to selection of the law of movement of the driven member, since of all the different laws of movement which satisfy the requirements of the technological process, it is possible to select the one with which the time of completion of the individual stages of motion, including periods of idle running, is the minimum. Reduction of the time of completion of individual stages of motion is related to increase in the impact loads, in friction losses, in torque on the camshaft, and other factors connected

with the dynamics of the motion of the gear members. For this reason, proper selection of the law of movement of the driven member from the conditions for achieving high machine efficiency presupposes that consideration will be given without fail to the influence of reduction of the time of movement on change in various dynamic factors.

In addition to this, it must be borne in mind that increase in machine efficiency may be achieved by combination of single operations, it being possible by proper selection of the form of the graph of the law of movement of the operating members of the individual mechanisms to effect a closer combination of the operations in time.

The third condition - achievement of minimum expenditure of energy for setting the gear in motion - just as is the preceding condition, is related to evaluation of the economy of the machine and should be considered together with the other basic requirements made of a properly designed mechanism. If friction in kinematic couples is taken into consideration, then this condition will be seen to be related not only to selection of the law of movement, but also to establishment of the basid dimensions of the diagram of the mechanism (minimum radius of the cam, length of the guide, etc). However, even through rational selection of the law of movement of the driven member alone it is possible, as will be subsequently demonstrated, to reduce considerably the amount of torque on the camshaft and, consequently, to reduce the expenditure of energy needed to set the gear in motion.

The fourth condition - assurance of durability and long life of the gear - is achieved not only by selection of materials for production of the members of the gear and determination of the dimensions of the members by calculations for strength, but also by maximum reduction of active loads on the members of the mechanism and by rational designing of the individual units. The loads on the members of the mechanism depend on the law of movement of the driven member, and for this reason effort must be made when selecting this law to reduce the forces acting on the members of the mechanism.

The fifth condition - simplicity of production of the cam profile - is of great significance in comparison of various versions of the mechanism which satisfy the four conditions listed above. It is obvious that the version should be selected which corresponds to the simplest and most economical production of the members of the cam gear. Not infrequently in this situation, the conditions which insure simplicity of production of the cam force a deviation from the optimum relationships which are established by the preceding conditions.

Consequently, when selecting the law of movement of the driven member it is necessary to take into consideration the various and often contradictory conditions. For this reason, before giving consideration to concrete examples of selection of the law of movement of the driven member, it is necessary to study the influence of the individual characteristics of the mechanism on selection of the law of movement.

These characteristics include: 1) the maximum velocity of the driven member, 2) the maximum acceleration of the driven member, 3) the load impact factor, 4) the spring constant, 5) the maximum torque on the camshaft, and 6) the maximum amount of pressure of the cam on the driven member.

The characteristics enumerated do not, of course, exhaust fully all the requirements which might be encountered in the designing of the entire variety of cam gears of automatic machines. However, in the overwhelming majority of cases, comparison of the various laws of movement in accordance with these characteristics will furnish an adequate basis for a selection of the law of movement in which the five conditions formulated above will be satisfied to the greatest extent.

SOME LAWS OF MOVEMENT OF THE DRIVEN MEMBER

In order to facilitate selection of the law of movement of the driven member, we shall take up certain widespread laws of movement. In subsequent sections there will be given both the comparative characteristics of the laws of movement under consideration and the conclusions which furnish a basis for selection of other laws of movement.

We shall arrange subsequently, when investigating the various laws of movement of the driven member, to study only a portion of its displacement in one direction, from one moment of stopping to the one following it, since all the conclusions which may be made in such a study may easily be extended to cover the event of movement of a driven member with any number of stops and double strokes during a full revolution of the cam. Moreover, we shall arrange to give all graphs, formulas, and functions only for a driven member which moves rectilinearly. It is obvious that for a rotating driven member one must correspondingly replace the linear displacements, velocity, and acceleration by the angular ones.

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We shall designate by t_i the time of the interval from one moment of stopping of a driven member to the one following it, and by S_i, the amount of the course traversed during this interval. Then the course S, the velocity v, and the acceleration a in any position of the driven member may be expressed by the amounts S_i and t_i:

$$S = \zeta S_i; \tag{1}$$

$$v = \delta \frac{S_i}{t_i}; \tag{2}$$

$$a = \xi \frac{S_i}{t_i^2}. \tag{3}$$

In these formulas, the dimensionless coefficients ζ, δ, and ξ depend on the law of movement selected and are variable magnitudes which it is convenient to express as functions of the dimensionless magnitude k, which varies from 0 to 1 and represents the relationship of time t to the time of the entire interval t_i or of the turning angle of the cam α to the turning angle during the entire interval α_i (in the case of a uniformly rotating cam), i.e.,

$$k = \frac{t}{t_i} = \frac{\alpha}{\alpha_i}. \tag{4}$$

Table 1 gives a summary of the magnitudes of ζ, δ, and ξ as functions of the magnitude k for several laws of movement. The acceleration graphs $a = f(k)$ are also given in this table for purposes of illustration.

All the laws of movement of a driven member in cam gears may be subdivided into three groups:

1) laws of movement insuring the achievement of the optimum magnitude of any single characteristic of the mechanism;

2) laws of movement which are determined by the form of the acceleration graph or the course graph;

3) laws of movement which are determined by the form of the curves which serve as the cam profile.

We shall take up first the laws of movement of the first group.

The optimum magnitude of any characteristic of a mechanism ordinarily corresponds to the condition of its constancy, since in this case it is the minimum. For this reason, the laws of constant velocity, constant acceleration, constant angle of pressure, and constant torque have come to be well-known among the laws of movement of the first group.

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With the law of constant velocity (Table 1, number 1), there takes place uniform movement of the driven member with minimum velocity (with specified S_i and t_i), but at the beginning and end of movement the acceleration equals infinity, i.e., rigid impacts occur.

The law of constant acceleration (Table 1, block 2) corresponds to variable motion, there being breaks in the acceleration graph, i.e., soft impacts occur. The course graph represents a combination of squared parabolas; for this reason, this law is sometimes termed parabolic. The velocity maximum occurs in the position where the acceleration changes its sign, i.e., when $k = i$. The velocity graph (tachogram) is symmetrical when $i = 0.5$.

The law of constant angle of pressure is expressed by the condition

$$\theta = \theta_0$$

where is the angle of pressure, i.e., the angle between the line of action of the force Q, which acts on the driven member from the direction of the cam (the forces of friction not being taken into account), and the velocity of the point of application of this force v.

In gears with forward-motion cams, the angle of pressure θ is proportional to the velocity of the driven member; consequently, in the present instance the laws of constant velocity and constant angle of pressure coincide. In gears with a rotating cam and a forward-motion driven member, the geometrical axis of which passes through the center of rotation of the cam, we have

$$\operatorname{tg} \theta_0 = \frac{v}{\omega (R_0 + S)}, \tag{5}$$

where ω is the angular velocity of the cam;

R_0 is the initial (minimum) radius vector of the cam.

It follows from (5) that when $S = 0$ and $S = S_0$, velocity $v \neq 0$, i.e., rigid impacts take place at the beginning and end of movement. Equations of the course, velocity, and acceleration as functions of time are usually not made in this instance, the profile of the cam, which represents a logarithmic spiral, being determined directly.

The law of constant torque is expressed by the condition

$$M = \text{const}, \tag{6}$$

the value of the torque without friction being taken into account usually being understood by the magnitude M.

Similar course, velocity, and acceleration graphs of the driven member, as well as a similar cam profile, are obtained with the different hypotheses as to the nature of the changes in the forces acting on the driven member. Rigid impacts are observed in all these instances if use is not made of transition zones outlined in accordance with other laws.

It will be seen from the examples cited of the laws of movement of the first group that endeavor to satisfy completely only one characteristic of the mechanism (minimum of greatest velocity, acceleration, or torque, etc) results in the other characteristics' proving to be unfavorable. Consequently, the laws of movement of the second group, which are determined by the form of the acceleration graph, have become widespread. By changing the form of the acceleration graph, we may obtain various values of the characteristics of the mechanism and find the optimum solution.

Let us investigate the laws of movement in which the acceleration graphs are made up of segments of a straight line. As the simplest example we may use the law of constant acceleration, discussed above, in which mild impacts (acceleration jumps) occur in the center, as well as at the beginning and end of the interval, if there are halts (stops of finite duration).

An acceleration jump in the center of movement may be avoided by applying the law of movement with uniformly decreasing acceleration (Table 1, numbers 3 and 4), which is sometimes termed the cubic parabola law of movement in keeping with the form of the course graph.

Instances of both a symmetrical tachogram ($i = 0.5$) and an asymmetrical one ($i \neq 0.5$) are given separately in Table 1.

The condition of complete absence of acceleration jumps is satisfied by the law of movement with acceleration changing along a triangle, in which case the course graph consists of several segments of a cubic parabola (Table 1, number 5).

More common are the laws of movement with acceleration changing along a trapezium (Table 1, number 6), from which we may obtain all the preceding laws with acceleration graphs made up of segments of a straight line. Number 1 of Table 1 contains the equations for the coefficients ζ, δ, and ξ with the peculiar condition that the velocity graph is symmetrical and the sections of constant acceleration each comprise 0.25 of the interval in question.

In addition to segments of a straight line, we may use progressive polynomials

Table 1

Certain laws of movement of a driven member

No.	Designation	Acceleration graph	Limits from	Limits to
1	Constant velocity		0	1
2	Constant acceleration		0	i
3	Uniformly decreasing acceleration (symmetrical tachogram)		0	1
4	Uniformly decreasing acceleration (asymmetrical tachogram)		0 i	i 1
5	Change in acceleration along a triangle		0 0.25 0.75	0.25 0.75 1
6	Change in acceleration along a trapezium		0 0.125 0.375 0.625 0.875	0.125 0.375 0.625 0.875 1
7	Change in acceleration along a cosine curve (symmetrical tachogram)		0	1
8	Change in acceleration along a sine curve (asymmetrical tachogram)		0 i	i 1
9	Change in acceleration along a sine curve		0	1
10	Change in acceleration along a sloping sine curve		0	1
11	Constant velocity with transitional sections along a sine curve		0 i 1-i	i 1-i 1

Table 1 (continued)

No.	Course coefficient	Velocity coefficient	Acceleration coefficient
1	k	1	∞ 0 $-\infty$
2	$1/i\ k^2$ $1 - \frac{(2-k)^2}{1-i}$	$2/i\ k$ $\frac{2}{1-i}(1-k)$	$2/i$ $-\frac{2}{1-i}$
3	$3k^2(1 - 2/3k)$	$6k(1-k)$	$6(1-2k)$
4	$\frac{3k^2}{2i}(1 - \frac{1}{3i}k)$ $\frac{1}{2(1-i)^2}[2i^2 - i - 3(2i-1)k + 3ik^2 - k^3]$	$\frac{3k}{2i}(2 - \frac{1}{i}k)$ $\frac{3}{2(1-i)^2}(1-2i+2ik-k^2)$	$\frac{3}{i}(1 - \frac{1}{i}k)$ $\frac{3(i-k)}{(1-i)^2}$
5	$16/3\ k^3$ $1/6 - 2k + 8k^2 - 16/3\ k^3$ $1 - 16/3(1-k)^3$	$16k^2$ $16k(1-k) - 2$ $16(1-k)^2$	$32k$ $16(1-2k)$ $-32(1-k)$
6	$64/9\ k^3$ $1/3(1/24 - k + 8k^2)$ $1/3(7/6 - 10k + 32k^2 - 64/3k^3)$ $1/3(97/24 + 15k - 8k^2)$ $1 - 64/9(1-k)^3$	$64/3\ k^2$ $1/3(-1 + 16k)$ $2/3(-5 + 32k - 32k^2)$ $1/3(15 - 16k)$ $64/3(1-k)^2$	$128/3\ k$ $16/3$ $64/3(1-2k)$ $-16/3$ $-128/3(1-k)$
7	$\frac{1}{2}(1 - \cos\pi k)$	$\pi/2 \sin\pi k$	$\pi^2/2 \cos\pi k$
8	$i(1 - \cos\pi/2i\ k)$ $i + (1-i)\cos\frac{(1-k)}{2(1-i)}$	$\pi/2\ \sin\pi/2i\ k$ $\pi/2\ \sin\frac{\pi(1-k)}{2(1-i)}$	$\pi^2/4i \cos\pi/2ik$ $-\frac{\pi^2}{4(1-i)}\cos\frac{\pi(1-k)}{2(1-i)}$
9	$1/2\pi(2\pi k - \sin 2\pi k)$	$1 - \cos 2\pi k$	$2\pi \sin 2\pi k$
10	$z - 1/2\pi \sin 2\pi z$ $k = z - b/2\pi\ \sin 2\pi z$	$\frac{1 - \cos 2\pi z}{1 - b\cos 2\pi z}$	$2\pi(1-b)\frac{\sin 2\pi z}{(1 - b\cos 2\pi z)^3}$
11	$\frac{1}{2(1-i)}(k - \frac{i}{\pi}\sin\frac{\pi}{k}k)$ $\frac{1}{1-i}(k - \frac{i}{2})$ $1 - \frac{1-k}{2(1-i)} + \frac{i}{2\pi(1-i)}$ $\sin\frac{\pi}{i}(1-k)$	$\frac{1}{2(1-i)}(1 - \cos\frac{\pi}{i}k)$ $\frac{1}{1-i}$ $\frac{1}{2(1-i)}(1 - \cos\frac{\pi}{i}(1-k))$	$\frac{\pi}{2i(1-i)}\sin\frac{\pi}{i}k$ 0 $\frac{-\pi}{2i(1-i)}\sin\frac{\pi}{i}(1-k)$

of higher orders in order to construct acceleration graphs. However, laws STAT ovement with acceleration graphs representing trigonometric polynomials are much more widespread. Graphs in the form of a cosine curve and a sine curve would be the simplest instances. The law of movement with acceleration varying along a cosine curve is sometimes termed the harmonic law of movement (Table 1, number 7), and the law of movement with acceleration varying along a sine curve, the cycloidal law of movement (Table 1, number 8). In contrast to the law of movement with acceleration along a cosine curve, in conformity with which there will be acceleration jumps, if there are halt sectors between the intervals of movement $(0, t_1)$, this law does not have mild impacts (acceleration jumps). On the other hand, the law of movement with acceleration along a cosine curve, as will be demonstrated further on, yields more favorable values of the characteristics of the mechanism in consequence of displacement of the center of gravity of the graph of positive accelerations towards the beginning of movement.

A successful combination of the properties of the sine curve and the cosine curve may be obtained by application of the law of movement with a course graph in the form of a sloping sine curve (Table 1, number 10). The form of the sloping sine curve is characterized by a certain constant parameter b, which varies from -1 to 1. In Table 1, number 10, acceleration graphs are shown for two values of the parameter b: $b = 0.15$ and $b = -0.15$. It is evident from these graphs that by altering the parameter b it is possible to displace the center of gravity of the graph of positive accelerations, preserving the absence of acceleration jumps. With a positive b, the center of gravity of the positive accelerations is displaced towards the origin of the coordinates, and with a negative b, from the origin of the coordinates. When $b = 0$, we have an ordinary sine curve. The coefficients ζ, δ, and ξ are expressed by z, an auxiliary variable parameter which varies from 0 to 1. This parameter is connected to the variable k by the following relationship:

$$k = z - \frac{b}{2\pi}\sin 2\pi z.$$

In addition to the laws of movement with acceleration along a cosine curve, sine curve, and sloping sine curve, laws of movement have also been proposed for which the acceleration graphs represent trigonometric polynomials in the form of the sum of the harmonics of the various orders.

The laws of movement shown in numbers 2-10 of Table 1 have a two-period tacho-

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gram, i.e., the movement cycle consists of periods of acceleration and deceleration. If it is necessary to effect movement with constant velocity (three-period tachogram) between these periods, any of the laws of movement shown in numbers 2-10 of Table 1 may be utilized for periods of acceleration and deceleration. An instance of a three-period tachogram with transitional sectors along a sine curve (Table 1, number 11) is given as an example.

The laws of movement of the third group are determined by the form of the curves which serve as the cam profile. The appearance of these laws of movement is occasioned by the endeavor to insure the simplest technological execution of the cams. For example, one may receive as multipurpose equipment cam profiles outlined along the arcs of circles. Equations are not generally made for S, v, and a in this instance, since the cam profile is determined directly from the condition of its composition from circle arcs, and the extremum values of velocity v and acceleration a are found by means of the ordinary methods of kinematic analysis. We shall remark only that the acceleration graph contains breaking points which are unavoidable with a profile composed of circle arcs, since at the points of their interconnection the tangents coincide, but the radii of curvature are different.

Besides profiles composed of circle arcs, profiles have also been proposed which are outlined along an extended involute, and these also may easily be received as multipurpose equipment.

MAXIMUM VELOCITY OF THE DRIVEN MEMBER

We shall begin investigation of the characteristics affecting selection of the law of movement of the driven member with clarification of the relationship of the maximum velocity of the driven member v_{max} to the law of movement with specified S_i and t_i.

Designating the maximum value of the coefficient δ in formula (2) by δ_{max}, we have

$$v_{max} = \delta_{max}\frac{S_i}{t_i}. \qquad (7)$$

The connection between the magnitude δ_{max} and the law of movement may be clarified by utilization of the following property of the course, velocity, and acceleration graphs.

Increase in the course at a certain interval equals the product of the area of the acceleration graph by the distance from the center of gravity of this area to the end of the interval, if the velocity is a monotonic function equalling zero at the beginning of the interval.

Designating the time corresponding to the distance from the beginning of the interval to the center of gravity of the graph of the positive accelerations by t_c, on the basis of this theorem we have for laws of movement with a symmetrical tachogram

$$\frac{S_i}{2} = (0.5t_i - t_c)\int_0^{\frac{t_i}{2}} a\,dt \qquad (8)$$

Taking into consideration the fact that $\int_0^{\frac{t_i}{2}} a\,dt = v_{max}$, we obtain

$$v_{max} = \frac{S_i}{t_i - 2t_c}. \qquad (9)$$

We shall designate by λ the distance from the beginning of the interval to the center of gravity of the area of the graph of positive accelerations, which is expressed in fractions of the run sector $(0, it_c)$,

$$\lambda = \frac{t_c}{it_i}. \qquad (10)$$

Then, comparing expressions (7) and (9), for laws of movement with a symmetrical tachogram we obtain

$$\delta_{max} = \frac{1}{1 - 2i\lambda}. \qquad (11)$$

When $i = 0.5$ (symmetrical tachogram),

$$\delta_{max} = \frac{1}{1 - \lambda}. \qquad (12)$$

For laws of movement with an asymmetrical two-period tachogram, on the basis of the properties of the course, velocity, and acceleration graphs given above, we have

$$S_1 = (1 - \lambda)it_iv_{max}, \qquad (13)$$

where S_1 is the course traversed during the run period. On the basis of the same theorem, the course traversed during the run-out period is:

$$S_i - S_1 = (1 - \lambda)(1 - i)t_iv_{max}. \qquad (14)$$

Hence,

$$\frac{S_1}{S_i - S_1} = \frac{i}{1 - i}, \qquad (15)$$

or

$$S_1 = iS_i \qquad (16)$$

i.e., the point of interconnection of the run and run-out curves on the course graph lies on the straight line connecting the initial and terminal points of this graph.

From formulas (13) and (16) we obtain

$$v_{max} = \frac{1}{1-\lambda}\,\frac{S_i}{t_i},$$

i.e., the coefficient δ_{max} is determined by formula (12) and does not depend on the relation between the run and run-out periods.

The values of the coefficient δ_{max}, computed by formulas (11) and (12), are given in table 2, in which are assembled all the characteristics under discussion for the majority of the laws of movement given in Table 1. The following conclusions may be drawn on the basis of formulas (11) and (12) and examination of Table 2.

1. All the laws of movement with a two-period tachogram (i.e., without sectors of constant velocity) in which the graphs of positive and negative accelerations are symmetrical ($\lambda = 0.5$), have the coefficient of maximum velocity $\delta_{max} = 2$.

2. In order to reduce the coefficient δ_{max}, it is necessary to displace the center of gravity of the graph of positive accelerations towards the beginning of the interval. For example, the law of uniformly decreasing accelerations ($\delta_{max} = 1.5$), the law of change of acceleration along a cosine curve ($\delta_{max} = 1.57$), and the law of change of acceleration along an inclined sine curve when $b > 0$.

3. The lack of symmetry of the two-period tachogram does not affect the magnitude of δ_{max}.

4. The minimum value of the coefficient δ_{max} is obtained with the law of constant velocity ($\delta_{max} = 1$), but since with this law rigid impacts will occur at the beginning and end of movement, an example has been given in Table 2 of a three-period tachogram with transitional sectors along a sine curve with $i = 0.1$ ($\delta_{max} = 1.1$). In order to reduce δ_{max}, it is necessary to displace the center of gravity of the graph of positive accelerations towards the beginning of the interval, i.e., reduce λ and i, reduction in the magnitude of i being limited by the condition of increase in the accelerations.

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Table 2

Characteristics of certain laws of movement

No.	Designation of law of movement	Acceleration graph	Maximum velocity δ_{max}	Maximum acceleration ξ_{max}
1	Constant velocity with transitional sectors along a sine curve $i = 0.1$		1.11	17.4
2	Constant acceleration (instance of symmetry) $i = 0.1$		2	4
3	Constant acceleration (instance of asymmetry) $i = 0.7$		2	-6.67 2.86
4	Constant acceleration (instance of asymmetry) $i = 0.3$		2	6.67 -2.86
5	Uniformly decreasing acceleration		1.5	6
6	Change of acceleration along a triangle		2	8
7	Change of acceleration along a trapezium		2	5.33
8	Change in acceleration along a cosine curve		1.57	4.93
9	Change in acceleration along a sine curve		2	6.28
10	Change in acceleration along an inclined sine curve with $h = 0.15$		1.74	5.9

Table 2 (continued) STAT

Number	Load impact factor K_{dyn}	Spring constant	Cam pressure			Maximum torque			Machining method
			Impact load q_3	Increasing resistance q_1	Decreasing resistance q_2	Impact load $\lvert\delta\xi\rvert_{max}$	Increasing resistance $\lvert\delta\xi\rvert_{max}$	Decreasing resistance $\lvert\delta(1-\xi)\rvert_{max}$	
1	1	17.6	18.6	1.2	1.2	9.6	1.05	1.05	attachment
2	3	6	6.88	1.1	1.1	8	1.1	1.1	[illegible]ter form
3	3	9.5	11.5	1.2	0.9	13.3	1.4	0.9	"
4	3	9.5	11.5	0.9	1.2	13.3	0.9	1.4	"
5	2 with halts, 1 without halts	6	4.4	1.05	1.05	3.45	1.0	1.0	"
6	1	8.8	9.8	1.2	1.2	8.5	1.3	1.3	"
7	1	7.2	8.2	1.1	1.1	8.9	1.2	1.2	"
8	2 with halts, 1 without halts	5.1	3.9	1.1	1.1	3.8	1.0	1.0	attachment
9	1	7.0	8.1	1.1	1.1	7.8	1.3	1.3	"
10	1	6.2	5.9	1.1	1.1	4.5	1.1	1.1	"

MAXIMUM ACCELERATION OF THE DRIVEN MEMBER AND THE LOAD IMPACT FACTOR

The maximum acceleration of the driven member a_{max} is characterized by the coefficient ξ_{max}, which represents the maximum value of the coefficient ξ :

$$a_{max} = \xi_{max} \frac{S_i}{t_i^2} \tag{17}$$

We shall designate the relationship of average acceleration to the maximum filling coefficient by ν :

$$\nu = \frac{a_{av}}{a_{max}}. \tag{18}$$

For laws of movement having symmetrical tachograms, we have

$$a_{av} = \frac{v_{max}}{it_i}$$

or

$$a_{av} = \frac{\delta_{max}}{i} \frac{S_i}{t_i^2}. \qquad (19)$$

Substituting in formula (18) the value of a_{max} from term (17), a_{av} from term (19), and δ_{max} from term (11), we have

$$\xi_{max} = \frac{\delta_{max}}{i\nu} \qquad (20)$$

or

$$\xi_{max} = \frac{1}{i\nu(1 - 2i\lambda)}. \qquad (21)$$

Similarly, for asymmetrical two-period tachograms we obtain for the run period

$$\xi_{max_1} = \frac{\delta_{max}}{i\nu}, \qquad (22)$$

and for the run-out period,

$$\xi_{max_2} = \frac{\delta_{max}}{(1 - i)\nu}. \qquad (23)$$

Consequently, the relationships between the maximum accelerations during the run and run-out periods are inversely proportional to the corresponding relative durations of the periods

$$\frac{a_{max_1}}{a_{max_2}} = -\frac{i}{1 - i}. \qquad (24)$$

It follows from formulas (20), (22), and (23) that in order to reduce ξ_{max}, it is necessary to increase the filling coefficient ν and reduce the coefficient δ_{max}, i.e., displace towards the beginning of the interval the center of gravity of the graph of positive accelerations.

The minimum value of ξ_{max} is obtained with the law of movement of constant acceleration with a symmetrical tachogram ($\xi_{max} = 4$). It must, however, be borne in mind that we must not compare the laws of movement by the amount of maximum acceleration without taking into consideration the presence or absence of acceleration jumps (mild impacts). Indeed, the magnitude of the maximum acceleration is limited by the magnitude of the impact load on the members of the mechanism. If we take into account the resilience of the members of the mechanism, we see that momentary change in the magnitude and direction of inertia causes elastic fluctuations and increase in the impact loads. In these instances, when impact loads (loads from the forces of inertia) predominate, this increase may be estimated through the impact load factor. Designating by m the mass of the driven member and by P_{dyn} the force acting on the driven member (taking into account the resilience of the members), we have:

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$$K_{dyn} = \left| \frac{P_{dyn}}{ma_{max}} \right|_{max}$$

It has been theoretically and experimentally established [6] that the magnitude of the load impact factor K_{dyn} depends on the parameters which determine the magnitude of the elastic fluctuations of the members of the cam mechanism (mass of the members, their resilience, nature of friction, etc.), and on the behavior of the first derivative from acceleration. In a first approximation, we may consider that after change in acceleration by magnitude, $K_{dyn} \approx 2$, and after change in acceleration by magnitude and direction, $K_{dyn} \approx 3$. When acceleration jumps are absent, in the majority of cases $K_{dyn} \approx 1$.

Consequently, in the case of the law of constant acceleration due to change in acceleration both in magnitude and direction, the load impact factor proves to be equal to three. For this reason, the actual magnitude of maximum impact on the members of the mechanism with this law proves to be much greater than, for example, with the law of movement with acceleration along a sine curve ($\xi_{max} \approx 6.28$), even though comparison of the maximum accelerations by magnitudes alone yields the opposite results. When the magnitude of maximum acceleration is limited by the requirements of the technological process, it is likewise in many instances not permissible to permit changes in acceleration by direction; in both cases, therefore, the law of constant acceleration will not be the most advantageous one.

If acceleration changes only in magnitude at the ends of the interval, this occurring, for example, in the case of the law with acceleration along a cosine curve and in that of the law of uniformly decreasing acceleration in the case of movement halts (stops of finite duration), then the load impact factor equals two. With the same laws of movement, but without movement halts, the load impact factor equals one, since acceleration in this case does not change its magnitude at the ends of an interval.

If comparison is made of the magnitude of maximum acceleration of the laws of movement, with the condition that acceleration jumps are absent, the minimum magnitude of ξ_{max} can be closely approximated by means of the law of movement with acceleration changing along a trapezium (Table 2). Of the simplest laws of movement in the form of trigonometric polynomials, the law of movement with acceleration along a cosine curve yields a small magnitude of ξ_{max}, if there are no halts. When halts are present, a good result may be obtained by means of the law of movement with acceleration along an inclined sine curve.

THE SPRING CONSTANT

Power locking of the higher pair in a cam gear is usually accomplished by a spring, dimensions for which must be selected which will insure the absence of break-away of the driven member from the cam.

Let us consider separately the straight and reverse stroke of a driven member, and assume that the load on the driven member consists only of the force of the spring and the force of inertia.

In Figure 1 are shown graphs of the change in acceleration a, the force of inertia P_i, and the force of the spring P_p in relation to the path of the driven member S during a straight stroke.

Figure 1. Determination of the spring constant during a straight stroke. 1 - P_i; 2 - P_i; 3 - S_p; 4 - P_p; 5 - S_p.

Figure 2. Determination of the spring constant during a reverse stroke. 1 - P_i; 2 - P_i; 3 - P_p; 4 - S_p.

The same graphs for reverse stroke are given in Figure 2. Let us assume the graphs of change of acceleration a for the law of movement with a symmetrical tachogram to be identical for the straight and reverse strokes. In addition, let us assume that for the laws of movement with an asymmetrical tachogram, the run interval during a straight stroke equals the run-out interval during a reverse stroke.

The force of the spring

$$P_p = K_o(S + S_p), \tag{25}$$

where K_o is the spring constant;

S_p is the deformation of the spring corresponding to preliminary tension.

The condition

$$|P_p| > |P_i| \tag{26}$$

in the interval of negative accelerations during a straight stroke and in the interval of positive accelerations during a reverse stroke must be fulfilled in order for there not to be break-away of the driven member from the cam. This condition may be satisfied by proper selection of the constant K_o and of the magnitude of preliminary tension. The problem of selection of the magnitudes indicated is solved the most simply by means of graphs. It is necessary for this purpose to combine the graph $P_i = f_1(S)$ and the graph $P_p = f_2(S)$, arranging the latter in the way illustrated in Figures 1 and 2 by the dotted lines. Condition (26) is satisfied if the graph $P_p = f_2(S)$ is situated above the graph $P_i = f_1(S)$. At the extreme (if the spring has been taken without a reserve), the graphs mentioned will touch one another, the point of contact usually being near the point where acceleration has its extremum value.

By comparing the construction shown in Figures 1 and 2 we may see that the spring constant is identical if the run interval of the straight stroke equals the run-out interval during a reverse stroke. The point of contact of the graphs $P_i = f_1(S)$ and $P_p = f_2(S)$ must be found in each instance in order that the spring constant may be determined. The position of this point is determined by the dimensionless coefficient $\zeta_m = \frac{S_m}{S_i}$, where S_m is the path of the driven member corresponding to the point of contact indicated.

We shall designate by ξ_m the dimensionless coefficient which characterizes the magnitude of negative acceleration at this point. Condition (26) then assumes the following form:

$$K_o(\zeta_m S_i + S_p) \geqslant \xi_m \frac{S_i}{t_i^2} m,$$

where m is the mass of the driven member.

$$K_o = \frac{\xi_m}{\zeta_m + \frac{S_p}{S_i}} \cdot \frac{m}{t_i^2}. \tag{27}$$

Assuming subsequently for comparison of the laws of movement that $S_p = 0$, we obtain for estimation of them the dimensionless coefficient

$$\varkappa = \frac{\xi_m}{\zeta_m}. \tag{28}$$

The values of the coefficient $\varkappa$, computed with formula (28), are given in Table 2. The best results are yielded by the symmetrical laws of movement in which the maximum of negative acceleration is displaced towards the end of the interval (during a straight stroke). For this reason, the laws of movement with acceleration along a cosine curve, with acceleration along an inclined sine curve (with $b > 0$), and with uniformly decreasing acceleration give a much smaller magnitude of the force of the spring in comparison with the law of constant acceleration.

MAXIMUM TORQUE ON THE CAMSHAFT

Let us investigate the relationships which determine the magnitude of maximum torque on the camshaft, with the condition that the forces of friction may be ignored. We shall consider only mechanisms with a rotating cam and driven member which move progressively; the conclusions drawn from such an investigation will also be valid for other types of cam mechanisms. In order to take into account the influence of the law of movement on the amount of torque, it is convenient to consider the following four instances of loading of the driven member.

First instance: Static load, i.e., the load of external forces of resistance, predominates, the force of resistance P being constant in magnitude.

This instance is of significance for those slow-speed and heavily loaded mechanisms in which the forces of inertia are small in comparison to the external forces of resistance.

From the condition of equality of power for the driving and driven members, we have

$$M = Pv\frac{1}{\omega}, \tag{29}$$

where ω is the angular velocity of the cam.

It follows from formula (29) that when the magnitudes of ω and the force of resistance are constant, the law of change in torque on the camshaft coincides with the law of change in the velocity of the driven member. For this reason, the most favorable laws of movement with respect to reduction of the amount of torque will be those which give the smallest magnitude of the coefficient δ_{max}.

Second instance: The force of resistance increases in accordance with the linear law

$$P = \frac{P_{max}}{S_i} S.$$

The torque on the camshaft, in accordance with formula (29), has the value

$$M = vS\frac{P_{max}}{S_i \omega} \tag{30}$$

or

$$M = \delta\xi \frac{P_{max} S_i}{\omega t_i}, \tag{31}$$

i.e., when P_{max}, S_i, ω, and t_i are specified, the torque is proportional to the product of the path of the driven member by its velocity. Hence, with increasing force of

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resistance, the forces of inertia being ignored, it is possible to compare the various laws of movement by the amount of the maximum of the product of the dimensionless coefficients δ and ξ.

The amount of the maximum of the product of $\delta\xi$ is computed separately for each law in accordance with the ordinary rules for finding the extremum values of a function. It is advantageous in order to reduce this amount to displace the maximum of velocity towards the beginning of the interval.

Third instance: The force of resistance decreases in accordance with the linear law

$$P = P_{max}\left(1 - \frac{S}{S_i}\right).$$

In accordance with formula (29), the torque on the camshaft has the value

$$M = v(S_i - S)\,\frac{P_{max}}{S_o\,\omega} \qquad (32)$$

or

$$M = \delta(1 - \xi)\,\frac{P_{max}S_i}{\omega t_i}, \qquad (33)$$

and the laws of movement may be compared by the amount of the maximum of the product of $\delta(1 - \xi)$. It is advantageous in order to reduce this amount to displace the maximum of velocity towards the end of the interval.

Fourth instance: Impact load (loading from the forces of inertia) predominates.

This instance is of significance for those high-speed and lightly loaded mechanisms in which the forces of inertia are great in comparison to the external forces of resistance. In this case, $P = ma$ and from the condition of equality of power for the driving and driven members we have

$$M = va\,\frac{m}{\omega}$$

or

$$M = \delta\xi\,\frac{mS_o^2}{\omega t_i^3}. \qquad (35)$$

Consequently, the maximum amount of torque is proportional to the maximum amount of the product of the velocity and acceleration of the driven member. For this reason, when impact load predominates it is possible to compare the various laws of movement by the amount of the maximum of the product of both the dimensionless coefficients δ and ξ.

We may draw the following conclusions on the basis of examination of formulas

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(29-35) and Table 2, in which are given the values of the coefficients figuring in these formulas for several laws of movement.

1. If the force acting on the driven member may be considered constant and the force of inertia slight, the minimum amount of torque is yielded by those laws of movement which have the smallest magnitude of δ_{max}, i.e., the conclusions relating to reduction of the magnitude of maximum velocity are justifiable for this instance.

2. If the force acting on the driven member is directly proportional to the path of the driven member (increasing force of resistance), then it is advantageous to have asymmetrical tachograms with $i < 0.5$. Of the laws of movement with symmetrical tachograms, the most favorable prove to be those of the type of the law of movement with acceleration along a sine curve in consequence of reduction of the coefficient δ_{max}.

3. If the force acting on the driven member decreases with increase in the path, it is advantageous to have asymmetrical tachograms with $i < 0.5$, and the preceding conclusion is valid for laws of movement with symmetrical tachograms.

4. If the force of resistance acting on the driven member is small compared to the force of inertia (impact load), good results are yielded by the laws of movement in which the center of gravity of the area of the graph of positive accelerations is displaced towards the origin of the coordinates (uniformly decreasing acceleration, acceleration along a cosine curve and along an inclined sine curve with $b > 0$. For example, the law of movement with acceleration changing along a sine curve gives half the amount of maximum torque yielded by the law of constant acceleration.

The conclusion that the magnitude of δ_{max} should be reduced remains valid for other instances of loading of the driven member not discussed here. However, if friction in kinematic couples is taken into account, i.e., if consideration is given to the efficiency of the mechanism, the conclusions may change somewhat. As G. A. Shaumian has demonstrated, the momentary efficiency of a cam mechanism has its maximum magnitude with a certain value of the angle of pressure, which we shall designate by θ_k. The maximum value of average efficiency is usually obtained on the condition that

$$\theta_{max} = \theta_k,$$

where θ_{max} is the maximum value of the angle of pressure in a given mechanism. It follows from this that the results of comparison of the laws of movement of the driven member by the magnitude of maximum efficiency may vary in relation to the length of the profile of the cam L (for progressively moving cams) or the maximum radius vector R_o (for rotating disc cams).

If the magnitude of L (or R_o) selected is so large that with the laws of movement being compared $\theta_{max} < \theta_k$, in order to increase the efficiency it is necessary to increase the angle of pressure and, consequently, select laws of movement in which the magnitude of maximum velocity is great, since the angle of pressure increases with increase in velocity. If, with the magnitude of L (or R_o) selected, $\theta_{max} > \theta_k$ (an instance which is more often encountered), then in order to increase the efficiency it is necessary to reduce the angle of pressure and correspondingly select laws of movement having a smaller magnitude of δ_{max}.

Maximum amount of pressure of the cam on the driven member

Without taking into account the forces of friction, we find the amount of pressure of the cam on the driven member Q from the condition

$$Q = \frac{P}{\cos\theta}. \tag{36}$$

In the case of a progressively moving driven member, the angle of pressure for cylindrical and progressively moving cams is determined by the formula

$$\operatorname{tg}\theta = \frac{v}{v_k}, \tag{37}$$

where $v_k = \frac{L}{t_i}$ -- the velocity of the cam;

L is the length of the profile of the cam.

Substituting the value of the angle of pressure θ from relation (37) in formula (36), and taking formula (2) into consideration, we obtain

$$Q = P\sqrt{1 + \delta^2 c^2}, \tag{38}$$

where $c = \frac{S_i}{L}$.

Formula (37) is also applicable for rotating disc cams when displacement of the axis of the driven member with respect to the axis of rotation of the cam is absent. But in this instance

$$v_k = (R_o + S)\omega \tag{39}$$

and the coefficient c in formula (38) receives the value

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$$c = \frac{S_i}{(R_0 + S)\omega t_i}. \qquad (40)$$

In order to clarify the influence of the law of movement on the magnitude of maximum pressure of the cam on the driven member, let us examine the same four instances of loading of the driven member which were cited above.

First instance: The force directed along the axis of the driven member may be considered approximately constant: $P = \text{const.}$

In this instance, it follows from formula (36) that the minimum magnitude of pressure of Q_{max} is obtained when $\theta = \text{const}$, i.e., with the law of movement with a constant angle of pressure. Of the laws of movement shown in Table 1, the smallest magnitude of Q_{max} is given by those having a correspondingly smaller magnitude of δ_{max}.

Second instance: increasing force of resistance: $P = P_{max}\frac{S}{S_i}$.

In this instance, by substituting in formula (38) the value of the force P and taking formula (1) into consideration, we obtain

$$Q = P_{max}\xi\sqrt{1 + \delta^2 c^2}, \qquad (41)$$

i.e., the various laws of movement may be compared by the magnitude

$$q_1 = \left|\xi\sqrt{1 + \delta^2 c^2}\right|_{max}. \qquad (42)$$

In formula (41), the coefficient ξ grows continuously when S varies in the interval $(0, S_i)$. The coefficient δ in the same interval varies from zero to the maximum value of δ_{max} when $v = v_{max}$ and then again to zero. For this reason, the value of the coefficient q_1 will be the smaller, the closer to the beginning of the interval is the position corresponding to v_{max}, i.e., the smaller is the coefficient i.

Third instance: decreasing force of resistance: $P = P_{max}(1 - \frac{S}{S_i})$.

Here the opposite conclusion is obtained: In order for the maximum amount of pressure Q_{max} to be reduced, the coefficient i must be greater than 0.5, since

$$Q_{max} = q_2 P_{max},$$

where

$$q_2 = \left|(1 - \xi)\sqrt{1 + \delta^2 c^2}\right|_{max}. \qquad (43)$$

Fourth instance: The force of resistance P and the force of the spring P_i are slight or are absent, i.e., it may be considered that the force directed along the axis of the driven member consists only of the force of inertia $P_i = ma$.

In this instance

$$Q = \xi\sqrt{1 + \delta^2 c^2}\,\frac{mS_i}{t_i^2}, \qquad (44)$$

i.e., the various laws of movement may be compared by the magnitude

$$q_3 = \left|\xi\sqrt{1 + \delta^2 c^2}\right|_{max}. \qquad (45)$$

Table 2 gives the values of the magnitudes q_1, q_2, and q_3 for the various laws of movement with a constant magnitude $c = 0.7$. It follows from formulas (41 - 45) and the data contained in Table 2 that the conclusions pertaining to the influence of the amount of maximum pressure of the cam on the driven member to a large extent coincide with the conclusions drawn in investigation of the amount of maximum torque.

METHOD OF MACHINING THE CAM PROFILE

All the preceding conslusions relating to the obtaining of laws of movement with optimum magnitudes of the individual characteristics will conform to reality only in the event that the method of machining the cam profile insures the obtaining of accuracy of reproduction of a specified profile with which deviations from the law of movement adopted are less than the corresponding difference between the laws of movement being compared.

All the existing methods of machining series of cams may be subdivided into three groups:

1. Machining on multipurpose lathes, by means of which it is simple enough to obtain profiles of disc cams outlined along the arcs of circles, Archimedean screw, and extended involute, and profiles of cylindrical cams along a spiral. In the machining of these profiles, the blank and the tool have the same relative movement which the cam and driven member have in the mechanism (kinematic method of machining). The machining of other profiles, which is accomplished by means of successive shifts of the tool and blank by a specified amount in accordance with the shop drawing of the profile, requires the expenditure of much labor and frequently does not insure the proper accuracy.

2. Machining in attachments by means of which it is possible to obtain cam profiles corresponding to the laws of movement with acceleration varying along a cosine curve and an inclined sine curve, as well as to combinations of the latter with the law of constant velocity.

3. Machining on duplicating machines, with which it is possible to obtain any

cam profiles permitting the production of master forms.

The most promising form of machining is the kinematic method, since it insures a much greater accuracy of production than does the copying method. In order for one to convince himself of this, it is enough for him to recall the fact that proper accuracy of production of profiles of teeth in gear wheels was obtained only after transition from the copying to the runnin in (kinematic) method. For this reason, in comparison of the various laws of movement having approximately equal characteristics, preference should be given to those laws which permit machining in attachments or on multipurpose machines by the kinematic method (see the last column of Table 2).

EXAMPLES OF SELECTION OF THE LAW OF MOVEMENT OF THE DRIVEN MEMBER

For correct selection of the law of movement with which are fulfilled the conditions insuring full conformity to the requirements of the technological process, achievement of high efficiency, minimum expenditure of energy, and proper durability and life of the mechanism, in many instances it is necessary to make a comparison of several versions of the law of movement, since it frequently proves impossible to satisfy fully all the requirements set. However, several recommendations may nevertheless be given concerning application of the laws of movement for certain standard cases.

<u>It is necessary to obtain constant velocity of the driven member</u>

If rigid impacts (slow-speed and lightly loaded mechanisms) may be permitted, for profiling of cams direct use should be made of the law of constant velocity, the magnitude of the impact load at the moment of impact, the resilience of the members being taken into account, being verified by the formula

$$P_{dyn} = v \sqrt{mC}, \qquad (46)$$

where v is the velocity of the driven member at the point of contact with the cam profile $[\frac{m}{sec}]$;

m is the mass of the driven system $\frac{kg/sec^2}{m}$;

C is the rigidity of the driven system, i.e., the ratio of the force acting on the driven system to the full magnitude of the deformation corresponding to this force $[\frac{kg}{m}]$.

The profile of a progressively moving cam obtained in this process is in the form of a segment of a straight line; the profile of a rotating disc cam without displacement in the form of an Archimedean screw in the case of a progressively moving driven member or in the form of an epicycloid (hypocycloid) in the case of a rotating driven member; and the profile of a rotating disc cam with displacement in the form of an extended involute.

All the profiles mentioned admit of machining by the kinematic method.

If with a given velocity of the driven member v and its mass m, rigid impacts prove to be inadmissible, but soft impacts (acceleration jumps) do not yet cause considerable loads (due to slight forces of inertia), for formation of the cam profile use should be made of the same curves, but with the addition of transitional sectors outlined along the arcs of circles which insure smooth change of velocity from zero to the constant magnitude and then back to zero. Finally, if it is necessary to avoid even soft impacts, this necessity being established by previous determination of the maximum magnitude of the force of inertia, with the impact load factor taken into account, and by comparison of this magnitude with that permitted by the conditions of durability of the driven member, then use should be made of the law of movement with a three-period tachogram. For example, it is possible to employ the law of constant velocity with transitional sectors in accordance with the law of movement with change in acceleration along a sine curve (Table 1, number 11), since this law of movement permits machining of the cam profile by the kinematic method.

<u>It is necessary to obtain minimum velocity of the driven member with predetermined magnitudes of displacement of the driven member S_i and interval time t_i</u>

This instance coincides with the preceding one.

<u>It is necessary to obtain a minimum magnitude of acceleration of the driven member with predetermined magnitudes of S_i and t_i</u>

The minimum magnitude of acceleration is obtained with the law of constant velocity. However, as was pointed out above, the load impact factor must without fail be taken into consideration, this resulting in the condition of absence of acceleration jumps. If there are no halts at the beginning and end of the interval, the law of movement with acceleration along a cosine curve may be recommended, since in this case accelerations jumps are absent; maximum acceleration is small in magnitude ($\xi_{max} = 4.93$); the maximum magnitude of the product of velocity and acceleration which characterizes the amount of torque is also sufficiently small; and the cam profile may be machined by the kinematic method. [redacted]lts at the beginning and end of the interval, one may recommend the law of movement varying along an inclined sine curve with the parameter $b > 0$ (for example, with $b = 0.15$). Somewhat better results may be achieved with the law of movement with acceleration varying along a trapezium (common instance of an asymmetrical acceleration graph). However, in contrast to the law of the inclined sine curve, this law results in a form of cam profile for which there are no sufficiently simple attachments permitting machining by the kinematic method.

It is necessary to obtain a minimum interval time t_i with a specified magnitude of S_i, the forces of inertia being small in comparison to the forces of resistance, which are considered constant

(slow-speed mechanisms with a constant load)

Reduction of the interval time t_i with a specified magnitude of S_i is limited by increase in the amount of maximum torque. The minimum conditions of these magnitudes generally do not coincide, but in the instance under consideration we usually limit ourselves to comparison of the laws of movement only by the maximum magnitude of cam pressure on the driven member. Then, when the basic dimensions of the mechanism are equal, we assign a smaller magnitude of the time interval to the laws which have a correspondingly smaller magnitude of the angle of pressure θ. The minimum magnitude of the angle of pressure is obtained with the law of constant angle of pressure, which in the case of progressively moving cylindrical cams coincides with the law of constant velocity. In the case of rotating disc cams without displacement, the cam profile with constant angle of pressure is in the form of a logarithmic spiral, which cannot be machined simply enough by the kinematic method and, in addition, produces rigid impacts at the beginning and end of movement. Even more complicated curves are obtained in mechanisms with displacement of the driven member and in the case of a rotating driven member, the rigid impacts being retained. Hence, in the case of rotating disc cams, application of the law of constant angle of pressure (logarithmic spiral) without transitional sectors cannot be recommended. Since soft impacts are permissible in consequence of the weak forces of inertia, use should be made of profiles consisting of arcs of circles with transitional sectors which insure smooth change in velocity. If the machining is done by the kinematic method in attachments, it is also possible to employ the law of movement with acceleration changing along a cosine curve with $i > 0.5$, i.e., with an asymmetric tachogram and maximum of velocity displaced towards the end of the interval (Table 1, number 8). Finally, if the profile is to be machined by the copying method, it is possible to employ a combination of the law of constant angle of pressure (logarithmic spiral) with transitional sectors according to the law of constant acceleration.

It is necessary to obtain a minimum interval time t_1 with a specified magnitude of S_1, the forces of inertia being great in comparison to the forces of resistance (high-speed lightly loaded mechanisms)

In this case, the law of movement with acceleration along a cosine curve, halts being absent at the beginning and end of movement, and the law of movement along an inclined sine curve ($b > 0$) with movement halts are to be recommended.

The recommendations indicated for the five instances investigated pertain mainly to the cam gears of automatic machines; for drives it is necessary to investigate in addition such characteristics as impact in gap transition, amount of "time -- section", etc. Moreover, they are justifiable only in fulfilment of the conditions discussed in the assignment (ratio of the forces of inertia to the forces of resistance, etc). In the event of deviation from these conditions, for example, in case of increasing or decreasing force of resistance, it is necessary to make a more detailed comparison of the characteristics of the various laws of movement, use being made of the conclusions formulated during investigation of each individual characteristic.

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I. Z. ZAICHENKO

THE BASIC PROBLEMS IN THE SHHERE OF VOLUMETRIC AUTOMATIC HYDRAULIC TRANSMISSIONS

Volumetric hydraulic transmissions for progressive and rotary movement have become one of the basic means of automation of work processes.

However, the majority of the designs of hydraulic transmissions are completed experimentally, and much time is consumed in experimentation.

Further development of volumetric automatic hydraulic transmissions in mechanical engineering requires a sharp rise of the level of scientific research work to meet increased industrial needs. Scientists and designers must solve a number of very complicated problems embracing the complex of scientific-theoretic, constructional, and production questions of general importance which pertain to rotary motion hydraulic drives; control and adjustment, distribution, and reversible devices; and hydraulic tracking systems.

Increase of the pressure in the hydraulic system should be noted as one of the principal directions of further development of hydraulic transmissions.

At the present time, operating pressures not exceeding 50 to 70 kg/cm^2 are employed in the majority of the hydraulic systems of machines, except those in aircraft and press construction. Possible doubling of the pressure will make it possible to reduce the diameters of the operating cylinders by $\sqrt{2}$ times, and accordingly the weight of the hydraulic transmission also.

Increasing the pressure poses the problem of perfecting seals for progressive and rotary motion. Experience has shown that the difficulties connected with the creation of reliable and wear-resistant seals can be overcome. Increasing the pressure also makes necessary the conduct of scientific research work aimed at improvement of the filtration of pressure fluids and at selection of materials for the parts of hydraulic transmissions and pipelines.

The nomenclature of the functional hydraulic assemblies being produced at specialized plants should be extended, primarily through creation of modifications of basic hydraulic assemblies, on the principle of standardization and unification. Much attention should be devoted to perfection of the technology of production of hydraulic mechanisms. Work aimed at creation of model hydraulic transmissions should end in their standardization.

Among the scientific and technical questions of general importance, the problem of rigidity and vibrations in hydraulic drives in particular should be singled out; it has not yet been studied with sufficient thoroughness. Nevertheless, hydraulic systems are strongly subject to vibrations, this having its effect on the accuracy of operation of the machines and the quality of the articles machined.

Some factors affecting reduction of the intensity of vibration can be explained by the example of a hydraulic system (Figure 1) including an adjustable pump 1 and operating cylinder 2. If we ignore the mass of the pressure fluid, we may consider the vibrations of the piston as vibrations of the system with one degree of freedom, assuming that the piston, which is loaded with a force R, is at a certain instant taken as the beginning of the reading momentarily relieved until loaded with a load R_o, which corresponds to the velocity v_o. The piston of the operating cylinder is acted upon by a force of resistance (load to be overcome), which, generally speaking, is a function of the velocity of the oscillatory motion of the piston arising as a result of the compressibility of the pressure fluid (Figure 2). This function may be represented by Taylor's series, in which the terms with powers exceeding unity are discarded:

$$\varphi_1(x') = \varphi_o(v_o) + (x' - v_o)\,\varphi_1'(v_o) + \dots$$

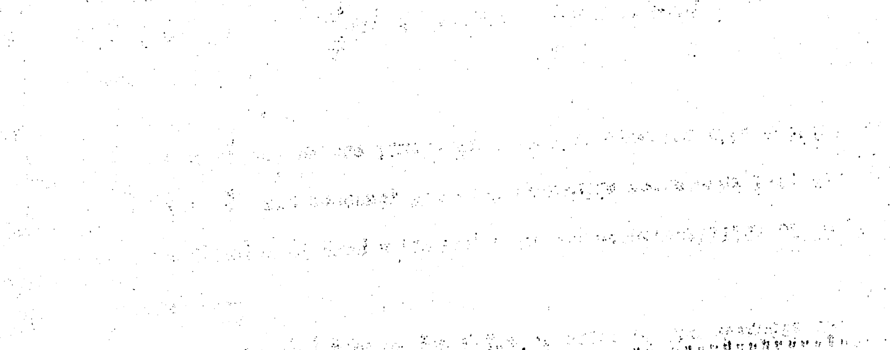

Figure 1. Diagram of a hydraulic transmission:

1 - pump; 2 - cylinder; 3 - choke.

In addition, the piston is acted on by the force of resistance, which is the result of friction of the mechanical parts and which may similarly be represented by the following equation:

$$\varphi_2(x') = \varphi_2(v_o) + (x' - v_o)\,\varphi_2'(v_o) + \dots$$

If we designate $\varphi_1'(v_o) = \alpha$ and $\varphi_2'(v_o) = \gamma$, the resulting force of resistance acting on the piston will be

$$f(x') = R_o + (\alpha + \gamma)(x' - v_o).$$

Figure 2. Curve of the velocity of oscillation of a piston.

1 - Zone of falling characteristic

Here α and γ are the tangents of the angle of slope of the characteristics of the forces of resistance.

The oscillation process of such a system, with the compressibility of the pressure fluid and leaks taken into account, but with hydraulic resistance left out of account, may be characterized by the linear differential equation with a right member:

$$x'' + \left(\frac{\alpha + \gamma + \frac{T_2}{T_1} M}{M}\right) x' + \frac{\frac{T_2}{T_1}(T_1 + \alpha + \gamma)}{M} x = \frac{\frac{T_2}{T_1}(T_1 + \alpha + \gamma)}{M} v_0 t +$$

$$+ \left(\frac{\alpha + \gamma + \frac{T_2}{T_1} M}{M}\right)(v_0 - \Delta v)$$

where $T_1 = \frac{dR_x}{dx'}$ — the rigidity of the hydraulic transmission according to velocity;

R_x is the load on the piston;

$T_2 = \frac{dR_x}{dx}$ — the rigidity of the hydraulic transmission according to displacement;

M is the mass of the parts being displaced;

v_0 is the velocity of displacement of the piston, corresponding to the load R_0;

Δv is the fall in velocity as a result of leakage of the pressure fluid upon change of load from R_0 to R.

If the characteristic of the forces of resistance in the v_0 field is falling, $\alpha + \gamma$ is negative, and if the characteristic falls steeply enough, an instance is possible where $\alpha + \gamma + \frac{T_2}{M_2} < 0$.

The system then falls into auto-oscillation, the intensity of which may be reduced by increasing the mass M, as well as by the inclusion of hydraulic resistance in the system. The latter may be achieved, for example, by setting the choke 3 in the rod recess of the cylinder 2 (Figure 1). In order to avoid unnecessary expenditure of power when the hydraulic transmission is operating at larger values of v_0, the choke is connected with the pump consumption change mechanism in such a way that when v_0 increases, the choke opening becomes larger.

Auto-oscillation processes frequently occur in hydraulic transmissions when various types of valves are installed in them. In this instance it is necessary to investigate oscillation of the systems with two or more degrees of freedom, the order of the differential equations being increased. The most typical examples should be subjected to analysis on the basis of which generalizing conclusions of great practical importance may be made. The solution to this problem should contain a general theory of vibrations in hydraulic transmissions, methods of calculations, and constructional measures for dealing with vibrations in the mechanisms of machines.

The following principal directions in the sphere of scientific research and experimental works may be distinguished for the piston, slide valve, and gear pumps employed in hydraulic transmissions, the majority of which are reversible:

1. Research of the phenomenon of wear and of the factors on which the efficiency of the pumps depends. This research should end in the creation of designs of pumps and motors which are wear-resistant under conditions of operation of the latter at high pressures and which are capable of insuring reliability of operation of automatic machines.

2. Research aimed at increasing the speed of pumps and motors with the aim of reducing their dimensions.

Thorough study of the processes of intake and forcing of pumps in order to insure their quiet operation at high speeds, as well as study of the degree of irregularity both of feed in pumps and of rotation for motors, should be provided for in the plan for such work.

3. Creation of compact and effective rotary motion hydraulic transmissions for main drives up to 100 kilowatts in power and for low-power drives (up to 3 kilowatts) which are employed for various purposes of automation (automatic clamping, tracking systems, rotation of articles and tables).

The development of automatic change of speed of rotary-motion hydraulic transmissions according to a specified law by means of regulation according to appropriate

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parameters should be included in the plan for such work.

Solution of the problems enumerated will require the perfection of certain types of pumps and motors and the investigation of new designs. One of the constructional solutions which insure increase in the resistance to wear of piston pumps and motors is slide-valve distribution accomplished directly by the pistons themselves. In this case, the law of change of opening of the slit through which the pressure fluid in the distributing piston passes is the same as the law of change of expenditure of fluid being forced or taken in by the piston, displaced 90°, and represents a cosine curve (Figure 3). For this reason, the speed of the pressure fluid passing through the slit of the controlling piston is constant, and, consequently, locking of the oil is theoretically eliminated.

Figure 3. Curve of the change of opening of the slit through which the pressure fluid in the distributing piston passes.

1 - total flow of pump; 2 - graph plotted for angle $\alpha = 10^{\circ}$; 3 - v, in meters per second.

There has been developed on this principle at the Experimental Scientific Research Institute for Lathes a design of a high pressure and low-consumption pump which has run for approximately 6000 hours at a pressure of 100 kg/cm^2. In the selection of the materials for the pump, it was found that a combination of 38KHMYUA steel for the block and SHKH-15 steel for the pistons is the most wear-resistant.

It may be assumed that such a design may be employed for high pressures up to 200 and more kg/cm^2 and for consumptions reaching 400 liters per minute.

The following circumstances hinder extensive employment of rotary motion hydraulic transmissions in mechanical engineering:

a) the difficulty of achieving a high efficiency, since the transmission consists of a pump and motor, production of the efficiency of which is determined by the overall efficiency of the transmission;

b) the large overall dimensions, which complicate the installation of hydraulic transmissions in machines.

A design of a hydraulic transmission utilizing the differential principle of operation (Figure 4) may be noted as one of the new solutions. A design of the spatial type is assumed as the basis of this hydraulic transmission, the body of the pump and motor being united in one block 1, which is rotated by the gears from the drive shaft. Depending on the slope of the washer 3 (the angle of slope of the washer 4 is constant), there may be two instances of operation of the transmission:

Figure 4. Diagram showing principle of operation of a differential hydraulic transmission:

1. with speed regulation for deceleration:

$$p_v = p\left(1 - \frac{tg\alpha_2}{tg\alpha_1}\right);\ M_k = M_m + M_p;\ M_p = M_k \frac{p_v}{p};\ N_p = \frac{M_k p_v}{97405}.$$

2. With regulation for acceleration:

$$p_v = p\left(1 + \frac{tg\alpha_2}{tg\alpha_1}\right);\ M_k = M_p - M_m;\ M_p = M_k \frac{p_v}{p};\ N_p = \frac{M_k p_v}{97405}.$$

a - from auxiliary pump; b - overflow to tank.

a) The pump is the mechanism nearest the exit shaft 2 of the transmission, the stream of oil forced by it being directed towards the motor, the effective torque (M_m) of which is added together with the torque (M_p) transmitted to the block by the drive shaft.

Regulation of speed in this case is possible only in the direction of reduction.

b) The pump is the mechanism separated from the exit shaft of the transmission, the stream of oil forced by it being directed towards the motor, the torque of which is compensated by the load M_k.

Regulation of speed in this case is possible only in the direction of increase.

The principle of the differential hydraulic transmission makes it possible to increase the efficiency in the zone of medium speeds, when the hydraulic transmission operates as a clutch, and to design a compact transmission capable of operation at a high speed, without the fear of accelerated wear of the distribution devices.

At the same time, some of the features of the principle of hydraulic transmissions of this type, to which the following belong, should be borne in mind:

1. The increase in the power transmitted through the pressure fluid, as well as the greatest possible degree of irregularity of rotation of the exit shaft when its speed is reduced (Figures 5 and 6); this power, without allowance for losses in the transmission, and the degree of irregularity of rotation are expressed by identical formulas, which have the following form: for operation of the hydraulic transmission for deceleration

$$N_g = N\left(\frac{p}{p_v} - 1\right),$$

$$\delta_g = \delta_n\left(\frac{p}{p_v} - 1\right);$$

for operation of the hydraulic transmission for acceleration

$$N_g = N\left(1 - \frac{p}{p_v}\right);$$

$$\delta_g = \delta_n\left(1 - \frac{p}{p_v}\right),$$

where N_g is the power transmitted through the pressure fluid;

N is the power transmitted by the transmission;

p is the number of revolutions of the block per minute;

p_v is the number of revolutions of the exit shaft per minute;

$\delta_g = \frac{\omega_{max} - \omega_{min}}{\omega_v}$ -- the greatest possible degree of irregularity of rotation of the exit shaft of a differential hydraulic transmission;

ω_{max} and ω_{min} are the greatest and smallest possible angular velocities of the exit shaft;

ω_v is the average angular velocity of the exit shaft;

δ_n is the greatest possible degree of irregularity of rotation of an ordinary hydraulic transmission having constructional parameters identical with the differential hydraulic transmission under consideration.

It follows from what has been stated that the utilization of differential hydraulic transmissions in the zone of low speeds is not advantageous due to the increase in the power transmitted by the pressure fluid and to the degree of irregularity of rotation.

2. The necessity of changing the direction of rotation of the entrance shaft in order to reverse the exit shaft, as well as the impossibility of installing the pump and the motor at a distance from one another.

The group of control-and-regulation, distribution, and reversible instruments includes valves, speed regulators, hydraulically and electrically controlled slide valves, and reversible slide-valve devices, as well as the most customary combinations of these units, hydropanels.

At the present time, this group of instruments has to a great extent been standardized by the Experimental Scientific Research Institute for Lathes, and this has made it possible to arrange for their serial production.

The employment of these instruments produced without pipes greatly simplifies assembly and disassembly; in the event of any defect, they can be rapidly replaced, this being particularly important in automatic machine lines.

The direction followed in domestic mechanical engineering of equipping machines with hydraulic devices with low speeds of displacement of the operating members (drilling and boring machines, lathes, milling, and other machines), which is based on the employment of continuous consumption pumps with choke regulation, requires the conduct of research and design work aimed at the achievement of a stable minimum consumption of 25 to 30 cm^3/min. Solution of this problem will insure reduction of the dimensions and weight of hydraulic motors.

Research must be provided for various types of valves which is based on a general theory of oscillations of hydraulic transmissions, with the aim of clarifying the constructional features of valves contributing to reduction of the intensity of oscillation. The most suitable methods of damping the valves and the most perfected designs for them must be found, and a method of calculation must be developed.

Fairly well-tested, reliably functioning hydraulic diagrams and functional assemblies are being employed for the hydraulic equipping of a group of automatic machines forming automatic machine lines. Control devices in which control by slide valves regulating the cycle proceeds from the camshaft should be employed in instances where, due to extensive saturation of a complex of automatic machines with hydraulic assemblies controlled by electric or hydraulic means, sufficient reliability of operation of the machines is not insured. This makes it possible to reduce the number of regulating slide valves and solenoids and to increase the reliability of operation of the system. There is need of further constructional development and perfection of such devices, the first models of which have been developed in the Experimental Scientific Research Institute for Lathes.

Figure 5. Graph of the relationship of the power of differential and ordinary hydraulic transmissions to the angular velocity of the exit shaft: 1 - Operation for deceleration; 2 - operation for acceleration; 3 - N_g; 4 - $p_v = p$; 5 - p_v.

Figure 6. Graph of the relationship of the degree of irregularity of rotation of differential and ordinary hydraulic transmissions to the angular velocity of the exit shaft:
1 - Curve of operation of differential hydraulic transmission for deceleration; 2 - Curve of operation of differential hydraulic transmission for acceleration; 3 - ordinary hydraulic transmission; 4 - δ_g; 5 - δ_n; 6 - $p_v = p$; 7 - p_v.

Reversal of the operation of hydraulic transmissions is frequently necessary in automatic machines. With insufficiently perfected reversible devices it causes hydra lic impacts, irregular displacement of the operating members, and marked STAT running.

Vital importance is assumed in connection with this by research work on the study of hydraulic impact, which occurs upon braking of the moving masses of a machine, the kinetic energy of which, depending on the reversal conditions, is to a greater or lesser extent transformed into potential energy of deformation of the compressed pressure fluid and pipelines.

Studies made at the Experimental Scientific Research Institute for Lathes of the relationship of the pressure of hydraulic impact to braking time for a closed progressive motion hydraulic system of transmission with an adjustable pump have shown that when the braking time is reduced, the pressure of the hydraulic impact increases, but cannot exceed the magnitude

$$\frac{M v^3}{2} = P_u^2 C,$$

where M is the mass of the moving parts of the machine;

v is the velocity of the parts of the machine;

P_u is the pressure of the impact;

C is the constant of the system which characterizes its resilience.

When the braking time $t_t = 0$, all the kinetic energy of the moving masses is transformed into potential energy of deformation of the pipelines and compressed fluid (Figure 7).

Figure 7. Curve of transformation of kinetic energy into potential energy in relation to the braking time:

$$P_u = 0.7\, v_o \sqrt{\frac{M}{C}}.$$

1 - P_u, in kg per cm^2; 2 - t_t, in seconds.

The work in this direction should be continued so as to culminate in the creation of a method of computing the magnitude of the pressure occurring upon hydraulic impact under various conditions (systems) of reversal, and in constructional measures for perfection of reversible devices, of great importance for which is research work

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on establishment of the permissible magnitudes of acceleration of the operating members of the machines and on study of the losses of energy upon reversal.

Hydraulic tracking systems are an important means of automation of production processes in mechanical engineering, since they posses the following very essential qualities:

a) slight inertness which does not affect the accuracy of operation;

b) the possibility of achieving, by comparatively simple means, complicated displacements of the operating members varying in accordance with the required law;

c) reliability of operation and resistance to wear.

Despite the fact that hydraulic tracking devices began to be employed for the driving of machines as early as the seventies of the last century (the first use of hydraulic boosters on the ships of the Russian Navy belongs to this period), yet at the present time there is still no valuable theory of these devices. In this respect the hydraulic engineers have lagged far behind the electrical engineers.

Questions of static calculation of hydraulic tracking devices, in the field of which significant experimental work has been conducted, have been subjected to more thorough study. For this reason, development of dynamic methods of calculation on the basis of a theory of automatic regulation and a theory of oscillations should be considered an important problem for hydraulic tracking devices. A theory of hydraulic tracking devices should be developed with allowance for the compressibility of the pressure fluid, since the modulus of elasticity of the latter is 100 times smaller than the modulus of elasticity of steel. This feature of the pressure fluid is the cause of the occurrence of various types of oscillations, which point was mentioned above.

The system of classification of hydraulic tracking devices shown in Figure 8 may be recommended at the present time.

Significant success has been achieved of late in domestic machine tool construction in the development and employment of slide-valve tracking devices. New, original designs of copying devices have been developed for lathes and milling and special machines. However, we do not yet have the exact characteristics and indices of all these systems, and this complicates selection of the best designs.

Questions concerning selection of the most rational constructional forms and dimensions of slide valves, stress analysis for their displacement and the design of damping devices, accuracy of operation, and optimum amounts of pressure in hydraulic tracking systems have not been studied with sufficient thoroughness. The well-known opinion that hydraulic tracking devices should be designed for low pressures (up to 25 kg/cm^2), so that significant oscillations may be avoided, cannot be considered substantiated, since there are tracking devices which operate with pressures up to 200 kg/cm^2.

Figure 8. Classification of hydraulic tracking devices.

1 - Hydraulic tracking devices; 2 - Devices for reproduction of a master form profile on the part being machined (copying devices); 3 - Devices with adjustable pump; 4 - Slide-valve devices; 5 - Continuous control; 6 - Intermittent control; 7 - Devices with tracking displacement of the operating members by one coordinate; 8 - Devices with tracking displacement of the operating members by two coordinates; 9 - Direct action; 10 - Indirect action; 11 - Devices for insuring the displacement of machine assemblies (boosters); 12 - Devices with adjustable pump; 13 - Slide-valve devices; 14 - Devices for synchronous rotation of two or more shafts; 15 - Devices with adjustable pump; 16 - Slide-valve devices.

As has been stated, slide-valve devices have been employed the most extensively in copying tracking devices. But the increase in speeds and operating forces caused by the further development of mechanical engineering will require the development of more powerful hydraulic tracking systems for which the slide-valve principle will prove to be unsuitable due to the significant losses of power in choking the pressure fluid. The development of hydraulic copying devices with an adjustable pump should be undertaken in connection with this.

Solution of this problem will require the surmounting of a number of difficulties among which we should count first of all the influence of leaks in the adjustable pump on the accuracy of operation of the tracking devices, particularly in operation at low tracking speeds. In slide-valve devices, leaks in the pump do not affect the operation of the system, since the pump operates on a valve.

Hydraulic tracking devices should be adopted more extensively in the future in heavy machine and machine tool construction. Devices for the synchronous rotation of two or more shafts in heavy machines may become industrially important.

Industrial employment of an "electric shaft" is observed in the field of electric drive, but almost no work is being conducted at the present towards creation of a hydraulic shaft.

In conclusion we shall note the necessity of further development of hydraulic transmissions for changing machine sppeds, the required magnitudes of which may be predetermined (preselective control), as well as for the blocking of mechanisms and automation of the operation of clamping, indexingk and conveying devices.

T. M. BASHTA

HIGH PRESSURE HYDRAULIC DRIVE IN AUTOMATIC CONTROL

The successful employment of high pressure hydraulic drives in aircraft construction may be of interest to various branches of machine construction.

Certain questions concerning the operation of hydraulic drives in automatic control systems are discussed in this article on the basis of data published in the foreigh and domestic literature.

Of importance for aircraft is the weight of the assemblies, since of the total weight of an airplane approximately 50% consists of the weight of the equipment and 50% of the weight of the design of the airplane; in other words, there is a unit of weight of the design for each unit of weight of an assembly.

If the auxiliary power devices employed on aircraft are evaluated from the viewpoint of these two basic indices, weight and bulk, then hydraulic transmissions will be seen to have indisputable advantages over other types of devices, and particularly over electric devices.

THE ADVANTAGES OF HYDRAULIC DRIVES

The development of hydraulic drives is proceeding in the direction of increase in fluid pressure and number of revolutions of the assemblies. For this reason, the pumps employed at the present in hydraulic aircraft transmissions operate within the limits of 4000 and 8000 rpm, and in some cases even 10,000 rpm. The prevalent fluid pressure in aircraft is 220 kg/cm^2; a pressure on the order of 300 atmospheres is in prospect for the next few years.

The employment of such speeds and fluid pressures in aircraft implies a high weight efficiency of hydraulic aircraft assemblies.

Figures 1, 2, and 3 are graphs of the relative weights of aircraft electric and hydraulic generators and motors which demonstrate that, the respective powers being equal, the weights of the electric generators and motors employed at the present time are approximately five times as great as the weights of hydraulic generators and motors designed for a pressure of 105 kg/cm^2, and eight times as great as the weight of hydraulic generators and motors designed for 210 kg/cm^2. Even a high-voltage alternating current generator cannot compare with hydraulic generators with respect to weight.

Figure 4 gives curves which characterize the gain in weight of a hydraulic aircraft system upon increase of pressure from 105 to 210 kg/cm^2.

The bulk of the hydraulic generator is eight to nine times smaller th STAT he bulk of an electric generator of equally great power.

The requirements for accuracy and quality of machining of the parts are increased in consequence of increase in pressure, since otherwise fluid leaks would increase, reducing the volumetric efficiency of the assembly, and in many cases possibly disrupting the operation of the hydraulic system as well. In view of this fact, the requirements have been increased for accuracy of production of the basic parts of hydraulic assemblies, which frequently must be held within the limits of two to four microns. However, with allowances of two to four micrens, the danger exists of wedging of mobile parts at low (-60°) and high ($+60^{\circ}$) temperatures. For this reason, both parts of a moving pair should be made of one material.

Figure 1. Weight characteristics of electric and hydraulic generators: 1 - Direct current electric generator; 2 - Alternating current electric generator; 3 - Hydraulic generator with pressure of 105 kg/cm^2; 4 - Hydraulic generator with pressure of 210 kg/cm^2; 5 - Weight, in kg; 6 - Power, in horsepower.

When the pressure is increased, various complications also occur which are related to the resilience of the fluid itself, which in certain instances may serve as a source of auto-oscillation of an assembly or its parts.

With pressures on the order of 150 kg/cm^2 and above, aluminum pipelines should be discarded and use made of thin-walled pipes of stainless steel having a wall thickness of 0.5 to 0.6 mm.

The fact must be taken into account that at high pressures of around 200 kg/cm^2 and above the allowance may be increased when the assembly is not sufficiently rigid and that fluid leakage increases. Increase in pressure is not always accompanied by a proportionate reduction in weight; it is often necessary to increase the weight in order to give the design the required rigidity and durability.

Hydraulic systems are suitable for operation at any altitude and within a wide temperature range, from minus 60° to plus 60° and above. Hydraulic transmissions are favorably distinguished from transmissions of other types by the extreme simplicity of accomplishing high degrees of reduction; regulation of the speed of hydraulic motors between 10 and 5000 rpm does not represent the limits for these assemblies. In addition, the stability of revolution of hydraulic motors greatly exceeds the stability of electric motors.

Hydraulic motors possess a high accelerating ability and slight inertness; thus, for example, reversal of the shaft of an 8-h.p. hydraulic motor from 2500 rpm to 2500 rpm in the opposite direction is accomplished during a time of 0.02 second.

An important advantage of hydraulic drive is the simplicity of accomplishing transmission with a high degree of reduction while maintaining a relatively high efficiency. Valuable qualities of hydraulic control are ease of control of pressure (stresses), number of revolutions (speed), bulk, reversal of the direction of motion (rotation) and other functions, as well as the speed of reaction of the performing member to the command impulse of the control emitter, on which depend the accuracy and sensitivity of the drive system.

Figure 2. Weight characteristics of electric and hydraulic motors:
1 - 200v electric motor at 2000 rpm; 2 - 200v electric motor at 5000 rpm; 3 - Electric motor at 20,000 rpm; 4 - Hydraulic motor with pressure of 105 kg/cm^2; 5 - Hydraulic motor with pressure of 210 kg/cm^2; 6 - Weight in kg; 7 - Power in hp.

Figure 3. Weight characteristics of electric and hydraulic motors: 1 - 24 v direct current electric motor at 7500 rpm for duration of operation under load of 1 min; 2 - 200 v three-phase intermittently operating electric motor at 12,000 rpm; 3 - Hydraulic motors at 3000 $\div$ 5000 rpm with pressure of 70 $\div$ 120 kg/cm^2; 4 - Pneumatic motors at 3000 $\div$ 8000 rpm with pressure of 16 $\div$ 18 kg/cm^2; 5 - Weight per hp, in kg; 6 - Power, in hp.

The simplicity of automation of the operation of controlling the aircraft and its assemblies should be added to the advantages of the hydraulic system.

The hydraulic system has simplified automation of prelanding operations such as extension of the brake and landing shields, preparatory loosening of the wheels in the air to eliminate "binding" of the wheels at the moment of their contact with the landing runway, etc.

Hydraulic systems insure minimum time lags in the response of the performing assembly from the time the command impulse is given, lags which in hydraulic systems can be reduced to around 0.005 to 0.02 second. Such a speed of reaction is due to the slight inertia of the units of the hydraulic assemblies and to the high speed of transmission through the oil-filled pipe of the hydraulic impulse, the speed of which for steel pipes and type MVP oil is as high as 970 to 1100 meters per second.

The great accelerating ability and small time lags of hydraulic assemblies have made it possible to employ them for automation of aircraft flight stabilization, and in particular have made it possible to accomplish automatic parrying of gusts of wind picked up by special emitters (receivers), which upon entry of the aircraft into an up-current give a command to the performing hydraulic booster of the aileron control system, which in turn deflects the ailerons in the proper direction. The STAT of such an automatic unit in practice does not exceed 0.05 second, owing to which fact the ailerons are capable of parrying even very abrupt gusts of wind in time.

Figure 4. Comparative weights of high and low pressure hydraulic systems:
1 - Full weight of hydraulic system, in kg; 2 - Gross weight of aircraft, in kg; 3 - Pressure of 70 kg/cm^2; 4 - Pressure of 210 kg/cm^2.

The employment of hydraulics has made it possible to make the aircraft control process automatic to a large extent, to perfect the control systems, to obtain stable speed drives for aircraft alternating current generators, and to obtain stable revolution drives for cabin superchargers, drives for radio sets, etc.

High reduction and a wide range of speeds, from 10-15 to 5000 rpm, are necessary for the driving of many aircraft assemblies; minimum irregularity of angular velocities at low speeds of revolution and maximum stability of revolution under various loads are required for this purpose. Hydraulic drives meet these requirements to the highest degree. Diagrams of the torque on the shaft of a standard 8-hp hydraulic motor as a function of n, as well as curves of the fluctuation of the angular velocity at low speeds of revolution, which confirm the high quality of hydraulic motors from this viewpoint, are given in Figure 5 in order to illustrate the performance of these drives.

Another of the superiorities of hydraulic systems is high efficiency, which for low-power (3 to 5-hp) generators and hydraulic motors in practice is 80 to 90%; with higher power, the efficiency increases. Figure 6 gives graphs which characterize the efficiency of electric and hydraulic generators. In many instances it is advisable to combine hydraulics with electricity, which in a number of instances is superior to hydraulics, particularly in rapidity of transmission of an impulse and in simplicity of conveying of energy. This principle may be formulated thus: it is expedient to employ hydraulics when it is necessary to achieve great force or to develop great power, and when it is necessary to transmit an impulse, it is expedient to employ electricity. In other words, hydraulics should perform in the aircraft the function

Figure 5. Characteristics of a hydraulic motor:
1 - Conditional speed; 2 - ω 1/sec; 3 - 12% irregularity; 4 - n_{av} = 15 rpm; M_{kr} = 102 kgcm; 5 - n_{av} = 11 rpm; M_{kr} = 0 kgcm; 6 - 8% irregularity; 7 - N_{SN}; 8 - Deflection angle of shaft; 9 - M_{kr} (kgcm); 10 - M_{kr} when P = 110 kg/cm^2; 11 - Power on shaft; 12 - N_{SN} when P = 110 kg/cm^2; 13 - n_{rev}; 14 - Volumetric efficiency; 15 - Pkg/cm^2; 16 - Torque; 17 - n rpm; 18 - Number of revolutions of shaft of hydraulic motor.

of a "muscular" system, and electricity the function of a "nervous" system.

Figure 6. Efficiency of electric and hydraulic generators:
1 - Hydraulic generator; 2 - Electric generator; 3 - Efficiency, in percent; 4 - Power, in hp.

However, instances of deviation from this principle are possible; even more probable is the advisability of employing combined electric and hydraulic systems, as mentioned above. In these instances, the superiorities of electricity (simplicity of conveying of energy, rapidity of transmission of an impulse, etc) and the superiorities of hydraulics (simplicity of accomplishment of power transmission with a high degree of reduction and high efficiency, reliability of functioning, etc) may be

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utilized to the full.

The principle of combining electricity and hydraulics is extensively employed particularly in hydraulic control assemblies. Thus, for example, the driving of hydraulic distribution devices is in the majority of cases performed electrically.

QUESTIONS CONCERNING THE DESIGNING AND OPERATION OF HYDRAULIC DRIVES

The principal assemblies of any hydraulic system are a hydraulic generator (pump), hydraulic motor, and distribution and safety devices.

One of the specific features of operation of aircraft hydraulic assemblies is the fact that these assemblies must insure operation under high-altitude conditions; this complicates their functioning, particularly the functioning of the pumps, owing to reduction of atmospheric pressure. The pumps then operate under conditions of cavitation, which causes erosion of the working pieces and causes the pump rapidly to become unserviceable because of back flow of the fluid and hydraulic impacts. For this reason, it is necessary in many instances to provide a special pressure feed of air into the tanks or employ auxiliary (booster) pumps for feeding the main pumps.

As has already been stated, modern pumps operate at pressures of 150 to 200 kg/cm^2. For the most part, gear pumps are employed for fluid pressures up to 150 kg/cm^2. Figure 7 shows the design of such a pump and, for purposes of comparison, the overall dimensions of an electric motor having the same power. So that a high volumetric efficiency will be insured, gear pumps (Figure 7) have a special side (face) sealing consisting of bronze disks a and b; disks b are fixed and disks a are movable and are tightened against the gear faces c by springs d and, in addition, by the operating pressure of the fluid moving into the cavities e.

Consequently, with increase in the operating pressure the force with which the disks a are tightened against the gears c, and hence also the tightness of contact, are increased. As we know, overflowing of fluid in a gear pump occurs along the radial gaps between the gear and body, as well as at the points of contact of the profiles of the meshing teeth and along the face gaps, between the gear faces and the side covers.

Experiments have shown that the amount of overflow along the radial gaps is negligible in comparison with overflow along the face gaps. This is explained primarily by the relatively great length of the path of fluid overflow along the radial gaps and by the countermovement of the cylindrical surfaces of the gears. Increasing the

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requirements in the matter of accuracy of execution of the tooth profile has made it possible to reduce overflow at the points of contact of the teeth. Reduction of overflow along the face gaps is achieved by means of the compensating device mentioned above, which reduces the amount of the gap with increase in the operating pressure of the fluid.

Figure 7. Gear pump:

1 - Volumetric efficiency, % - n_{rev}; 2 - Output, l/min - Q; 3 - n_{rev};
4 - From tank; 5 - To system; 6 - Boost, kg/cm^2; 7 - 21 kg.

Gear type pumps are usually constructed for outputs of up to 100 l/min when n = 4000 rpm.

Figure 7 gives a diagram of the output of a 30 l/min gear pump in relation to the amount of pressure of the fluid. Assuming the volumetric efficiency of the pump to be 100% at 0.4 kg/cm^2, we find that with a pressure of 150 kg/cm^2 it will be 94%. With increase in the service periods the volumetric efficiency of the pump usually

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decreases, but with a pump designed as described (See figure 7), the volumSTATic efficiency frequently even increases with the passage of time. However, this is possible only in the event of proper selection of the areas which are acted upon by the pressure of the fluid which tightens disks a against the gears and releases these disks from the gears. Obviously, if extreme tightening of the disks against the gears is permitted, scratching of the friction surfaces of the disks and gears may occur.

Figure 8. Multipiston pump.
1 - Conventionally drawn.

Figure 9. Diagram of an alternating output piston pump.

Experience has shown that the maximum pressure for gear pumps is 150 to 180 kg/cm^2. Further increase in pressure entails such complicated design and production technology that employment of these pumps becomes inadvisable. Multipiston pumps are employed for pressures around 150 kg/cm^2 and above. Structural diagrams of the most widespread pumps of this type are given in Figures 8 and 9. These pumps are produced both with non-adjustable (constant) output (Figure 8) and with automatic (Figure 9) or manual regulation of output.

The principle of operation of pumps of this type may be seen from Figure 8. Connected with a drive shaft 1, through a joint 2, is a multicylinder block 3. Upon rotation of the shaft 1 and block 3, pistons 4 accomplish reciprocating motion in the cylinders of the block, sucking in fluid from one of the ducts 5 and forcing it into the opposite duct.

The pump with automatic output regulation differs from the one described only in the regulation unit (Figure 9). The cylindrical block 1 of this pump may be ro- tated in the plane of the drawing (about the axis of its oil-pipe journals, not shown in Figure 9). A spring 2 strives to hold the block in the position of maximum angle of slope with respect to the axis of the drive shaft 7. However, after the pressure of the fluid fed along the duct 3 to the valves 4 overcomes the force of the spring 5, which has been adjusted to the required pressure, the valve 4 permits the fluid to pass into the cylinder 6. The pressure of the fluid, overcoming the force of the spring 2, will rotate the cylindrical block 1 towards reduction of its angle of slope with respect to the axis of the drive shaft 7. When the maximum pressure of the fluid is reached, the angle of slope of the block 1 equals zero and the feeding of the pump ceases.

Figure 10. Performance of an adjustable pump at 4000 rpm:
1 - Output of pump; 2 - Q, l/min; 3 - n_{tot} = f(P); 4 - n_{tot} = [illegible](P); 5 - n_{tot}; 6 - Total efficiency; 7 - n_{tot}; 8 - Volumetric efficiency; 9 - P, kg/cm^2; 10 - Fluid pressure.

Figure 10 gives the performance curves of such a pump. As may be seen from the graph, with increase in the pressure the output of the pump decreases somewhat; however, at a pressure of 190 kg/cm^2 the output curve falls sharply and at a pressure of 220 kg/cm^2 the output decreases to zero. The sharpness of the curve selected depends on actual needs. The accelerating ability of this pump in practice is computed in hundredths of a second, i.e., such a pump with automatic discharge is capable of developing and delivering practically instantaneously the maximum expenditure of fluid. Requirements for such a great accelerating ability are made in particular in the rudder control systems of aircraft and in the weapons control systems.

In certain instances, for the purpose of increasing the stability of the pump against autooscillation, the steepness of the dumping angle of the delivery Q (see Figure 10) is reduced by selection of appropriate springs 2 and 5 (See Figure 9).

The volumetric efficiency of a pump of this type is 97 to 98% at 200 kg/cm^2.

Figure 11. Hydraulic motor.

1 - B; 2 - Along AA; 3 - Along BB.

The pump just described may also be employed as a hydraulic motor. In practice, hydraulic motors with slide-valve distribution and a fixed cylinder block are widespread in hydraulic systems with pressures up to 150 kg/cm^2 (Figure 11). The drive shaft 1 of this hydraulic motor, which is connected with an inclined disk 4 has on its left tip a distributing slide valve 2 which is mounted on the eccentric pin 3 of the shaft 1. Upon rotation of the shaft 1, and together with it the inclined disk 4, the pistons 5 accomplish reciprocating motion in the cylinders of the fixed block 6;

the distributing slide valve 2, performing circular oscillatory motion, alternately connects the openings 8 of the cylinders with the corresponding ducts 7 and 9 (fluid delivery 7 and discharge 9).

The superiority of these hydraulic motors is the great accelerating ability achieved by virtue of the fact that their cylinder is fixed; aside from the pistons, the only moving part is the distributing slide valve 2, which here performs circular oscillatory motion. In practice, reversal of this hydraulic motor from a maximum speed of $n = 2500$ rpm to a maximum speed of $n = 2500$ rpm in the opposite direction of revolution is accomplished in 0.02 sec. The service life of these hydraulic motors, with proper use, may be extended to 5000 or more hours.

By connecting the pump and hydraulic motor with pipelines, we obtain a rotary type hydraulic drive (coupling). Regulation of the speed of revolution of the hydraulic motor in these drives is usually accomplished by changing the amount of fluid delivered to the hydraulic motor. The latter is accomplished either by regulation of the output of the pump by changing the length of the stroke of its pistons or by regulating (limiting) the delivery of the fluid moving into the hydraulic motor by means of a choke. The first type of regulation is generally employed in high power (over 2 or 3 hp) drives, and the second in low power drives.

Figure 12. Diagram of a stable revolution hydraulic drive.
1 - Hydraulic motor; 2 - Pump.

Of interest are hydraulic drives in which the number of revolutions of the exit shaft is automatically maintained in the required relationship to a given parameter. Thus, for example, it is frequently necessary to insure a constant speed of revolution of the exit (motor) shaft while the speed of the entrance (pump) shaft is variable. In drives of the type described this is accomplished by connecting the inclined disk in the pump (Figure 12) through a hydraulic booster b with a centrifugal governor a, which, upon disturbance of the established speed of revolution of the exit shaft for any reason, correspondingly changes the angle of slope of the pump disk and restores the speed of revolution of the exit shaft of the hydraulic motor.

As we know, the efficiency of the hydraulic drive described has its maximum value only for a specific performance and when these performance are employed, the efficiency of the drive decreases. When a pump and hydraulic motor of the same size are employed, the maximum delivery of the pump corresponds to this maximum efficiency

Hydraulic drives have been created on this basis in which power is transmitted to the driven unit in two ways: through the direct connection of the driving and driven shafts and supplementarily through the hydraulic drive, which in this instance transmits only a portion of the power. In these transmissions, the hydraulic drive merely adds (or takes away) the missing difference in speeds. By making use of this it is possible to change the arrangement of the drives so that the pump and motor will operate under conditions approaching the optimum.

Noteworthy among the drives of this type is the hydrodifferential drive, a diagram of the principle of operation of which is given in Figure 13. The entry (driving) shaft of the drive is shaft 2 and the exit (driven) shaft is shaft 13. In this drive, the pump (adjustable member) 14 and the motor (non-adjustable member) 15 either rotate as a whole, on bearings 1, or the hydraulic motor rotates at a lower or higher speed than the pump, or rotates in the direction opposite the direction of rotation of the pump.

When the angle of slope of the pump disk 4 is zero (neutral position), the hydraulic motor is rigidly connected to the pump through the fluid locked in the cylinders of the pump, the rotor of which is in turn connected through gears 5 and 6 and shaft 2 with the source of power, in this instance the aircraft engine.

Power in this case is transmitted from shaft 2 to the exit shaft 13 through the pump and hydraulic motor, which function here as the intermediate between these two shafts. The relative slip of shafts 2 and 13 is determined by the volumetric leakage of the fluid located in the cylinders of the pump and motor.

When the disk 4 is inclined in one direction or the other, the pump imparts either accelerating or decelerating motion to the motor. In the first instance, the pump 14 delivers the fluid into the operating cavity of the hydraulic motor 15, and, imparting relative motion to it, imparts additional revolutions to the exit shaft 13. In the second instance, the operating cavity of the hydraulic motor is connected to the non-operating (intake) cavity of the pump, as a result of which a portion of the

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fluid from the operating cavity of the hydraulic motor penetrates into the pump, i.e., the hydraulic motor operates as a "pump" for acceleration of the pump. In consequence of this the hydraulic motor "slips" with respect to the pump, as a result of which its speed of revolution decreases. It is easily seen that the amounts of accelerating or decelerating motions of the hydraulic motor depend on the amount of the positive or negative angle of slope of the disk 4, which amounts in a stable revolution drive are regulated by a centrifugal or other type regulator 10, which is connected to the exit shaft 13.

Figure 13. Diagram of a hydraulic drive.

The gear pump 8 serves for lubrication of the drive, and the pump 9 for delivery of the fluid through the chamber 12 into the main pump feed cavity 14 and for feeding the hydraulic system 7 of the speed governor. In the line along which the fluid is thrown into the tank is installed a valve 11 which maintains in the power line of the drive the pressure necessary for driving the pumppistons 14 and the inclined disk 4 turning gear 3.

In many instances it is necessary to achieve stable speed of a hydraulic motor which is fed from the common hydraulic system of the aircraft the output of which exceeds the needs of this hydraulic motor. In this case special governing devices are installed at the exit from the hydraulic motor 1 (Figure 14) which hold constant the consumption of fluid drawn off the hydraulic motor. It is obvious that disruption of the speed stability of the hydraulic motor when such a device is present can be caused only by change in fluid overflow (change in the volumetric efficiency of the hydraulic motor) due to fluctuation in the load of the hydraulic motor and in oil temperature.

Figure 14. Diagram of the speed governor of a hydraulic motor.
1 - From hydraulic system; 2 - To tank.

A throttling device (Figure 14) consisting of a throttling washer 2 and a regulator 3 which holds constant the pressure of the fluid in front of this washer may be used as a very simple regulator of the consumption of fluid leaving the hydraulic motor. The amount of this pressure is determined by the degree of tightening of the spring 4, which receives the pressure of the fluid on the piston of the valve 5.

Under the conditions established, the force of the pressure of the fluid against the piston 5 of the valve 6 and the force of the spring 4 are neutralized and the valve 6 with its tapered part throttles to some extent the fluid, thereby providing the required pressure. Upon change in the pressure of the fluid in front of the washer 2 due to any cause (change in pressure at the entrance to the throttling device, change in the viscosity of the fluid, etc), the valve 6 takes up a new position through which the extent of the fluid throttled by it changes as required to maintain an equilibrium of forces -- the force of fluid pressure against the piston 5 and the tension of the spring 4, i.e., to maintain the pressure established in front of the washer 2.

Experience has shown that a very simple governor such as this insures adequate speed stability of a hydraulic motor.

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However, in instances where particularly high speed stability (±0.5%) is necessary, to supplement this regulator use may be made of special speed governors which react to deviation in angular velocity. These governors may be of the mechanical (centrifugal) or electric type. In Figure 14 is a diagram showing the principle of operation of such a governor 7.

The governor 7 is a throttle-slide-valve device connected parallel to the pressure regulator 6 already discussed and which, depending on the position of the slide-valve 8, permits a greater or lesser quantity of fluid to pass through itself (bypassing the throttling washer 2). It is clear that reduction of this quantity of fluid causes reduction of the speed of the hydraulic motor and vice versa. The slide valve 8 is displaced by the speed governor of one or the other type. In particular, with an electric type governor, change (loss of adjustment) in the angular velocity of the shaft generally changes the voltage of the electric current, in consequence of which the slide valve 8, which is connected to an electromagnet 9, occupies a new position of equilibrium with which the loss of adjustment in angular velocity is eliminated.

In the diagram given in Figure 14, provision is also made for adjustment of variability of the pressure line, adjustment which is made by feeding this pressure into cavity 10.

DISTRIBUTING DEVICES

Manually driven (Figure 15) or electromagnetically driven (Figure 16) slide-valve devices are generally employed as distributing devices. Electromagnetically controlled slide-valve devices (Figure 16) have become widespread in the aircraft building industry. Here, electromagnets 3 displace an auxiliary slide valve 1 of small diameter (3--4mm), which feeds the fluid to the faces of the main distributing slide valve 2, setting it in the position corresponding to the direction of flow of the fluid.

When the electromagnets 3 are cut off, the main slide valve 2 is set in a central position, in which the cavities of the cylinder are connected to the overflow line.

A disadvantage of the slide-valve device just described is the difficulty of insuring the necessary hermetic sealing. For this reason, the use of flat slide valves (Figure 17) has begun in recent years. The function of the auxiliary slide valve is performed here by balls 6 which are acted on by electromagnets 5 overlapping one port or the other with the balls.

The fluid from the pump is fed along the port 9 and moves through the sleeve 8 to the distributing slide valve 1, from which it is directed, depending on the position of the latter, into the left or right port 10, which are connected to the power cylinders. In the central position of the slide valve 1 (shown in Figure 17), the port of the slide valve 1 is covered.

Displacement in the slide valve 1 is accomplished by feeding of the fluid into one or the other of the cavities 4 by pistons 2 and 3, the inner pistons 2 shifting the slide valve to the end position and the piston 2 setting the slide valve 1 in the central position (see position in Figure 17); the electromagnets are deenergized. In this instance, the fluid is fed into both cavities 4.

Figure 15. Slide-valve distributor.

Figure 16. Electromagnetic distributor with cylindrical slide valve: 1 - To tank; 2 - To power cylinder; 3 - From pump; 4 - To power cylinder.

The sleeve 8 is fitted against the flat slide valve 1 and is tightened against it by a spring 7 and the pressure of the fluid.

The flat slide valves described above are characterized by a high degree of hermetic sealing (practically complete); however, considerable production skill is required for their manufacture.

It was stated above that in the majority of modern aircraft use is made of booster (hydraulic booster) rudder control which operates in accordance with the following widespread system: the piston 6 (Figure 18) of the power cylinder 1 is connected to the rudder mechanism; inside the power piston is a slide valve 2 to which the

Figure 17. Electromagnetic distributor with flat slide valve:
1 - From pump; 2 - To tank.

Figure 18. Diagram of a hydraulic booster:
1 - Connected to control stick; 2 - Connected to drive unit.

fluid is fed through port 5. As may be seen from the diagram, when the slide valve 2 is moved from the neutral position in one direction or the other, the fluid moves through ports 7, 3, or 4 into one or the other cavity of the power cylinder 1 and displaces its piston 6, thereby eliminating the loss of adjustment of the positions of the piston 6 and slide valve 2 resulting from displacement of the latter.

It will easily be seen that if the required fuel consumption is provid STAT for and if the oil line ports are properly made, the power piston, and with it the rod connected to the drive unit, will follow the distributing slide valve with a minimum lag (loss of adjustment), which is computed in fractions of a millimeter, depending on the speed. This loss of adjustment depends to a large extent on the amount of overlap $S = \frac{a-b}{2}$ of the slide valve 2 of ports 3 and 4 (dimension figure b) by the shoulder 8 (dimension figure a). It is obvious that with zero overlap in the event $a = b$, the free travel of the slide valve is reduced to zero. In this case, the loss in travel adjustment, particularly at low speeds, will be negligibly small; the slide valve 2 is in a sense rigidly connected to the power piston 6, and the latter repeats the movement of the slide valve practically without lag. In view of the fact that it is difficult to achieve zero overlap, in practice it is made to equal $S = 0.05$mm. In certain cases it is sensible to place the slide valve not inside the piston but outside the power cylinder, connecting it in this instance with the moving piston by means of levers (Figure 19).

Figure 19. Diagram showing the principle of a booster drive.

If it is necessary for the "sensations" of the rudder to be transmitted to the control stick, the kinematic connection of the slide valve and stick with the power piston should be made as shown in the diagram in Figure 20. Depending on the relationship between shoulders a and b, it is possible to provide for "sensation" in the control stick; that generally selected equals one-twentieth to one-third of the force which would be transmitted to the control stick in the case of ordinary boosterless control. Hence, thanks to the employment of hydraulic boosters, we can provide for any force in the control stick, from zero to full force corresponding to boosterless control.

The utilization of hydraulics in the control of aircraft has made it possible to employ a special automatic device for imitation of the forces on the control stick not from the moment of swivel of the rudder but from the amount of flight speed or acceleration of the aircraft. In this case the forces which take into account the speed and acceleration of the aircraft are transmitted to the stick by special automatic devices. This has made it possible to create perfect aircraft control systems.

Figure 20. Diagram showing the principle of a booster drive with load on the control stick.

Noteworthy is the preliminary loosening of the wheels in the air prior to the landing of the aircraft which is employed in modern high-speed aircraft with the aim of reducing the load on the landing gear and of protecting the wheels from "binding". At the moment of contact between the wheels and the ground it may exceed the vertical load on the wheel; in addition, the wheel may, in consequence of the great moment of inertia, "bind" on the concrete for 40 to 50 meters and more, thereby causing increased wear of the landing gear. The positive effect of employing preliminary loosening of the wheels is especially noticeable in high-speed and heavy aircraft. When the landing gear is extended, a special hydraulic motor 2 (Figure 21) which serves to drive the wheel is automatically turned on. It is usually placed in the hollow axle 1 of the wheel, for which purpose a special slot is cut in the axle. The shaft 4 of the hydraulic motor 2 is connected to the wheel 3 by means of gears 5.

After the wheels of the aircraft touch the ground and the shock absorber is compressed, feeding of the hydraulic motor automatically ceases and the wheel slips on a roller acceleration device 6 with respect to the hydraulic motor, which is stationary in this case.

One of the problems of extreme importance to the aircraft industry is that of automatic control of the braking of aircraft wheels during the ground run of an aircraft. The importance of this problem consists of the fact that in effective manual braking the pilot always runs the risk of causing "binding" of the wheels, which is usually accompanied by ruining of the tires and frequently by damage to the aircraft.

In view of this, the pilot usually does not employ effective braking, and in consequence the landing run of the aircraft is lengthened.

Figure 21. Diagram of hydraulic drive for loosening the wheel of an aircraft.

Various automatic devices, of which devices of the inertia type have received the widest usage, are employed for more effective utilization of the brakes, and hence for reduction of the length of the run of the aircraft, while at the same time they eliminate the danger of destruction of the tires from "binding" of the wheels. The principle of operation of these devices is based on utilization of the kinetic energy of a pilot wheel which rotates together with the wheel, in order to transmit an impulse for releasing the brake of the wheel, in the event of loss of adjustment between the angular velocities of the flywheel and the wheel occurring during the initial stage of entry of the wheel into "binding" (when the wheel decelerates as it partially slips).

Figure 22. Diagram of an automatic braking device:

1 - To atmosphere; 2 - To brake; 3 - From valve; 4 - To brake;

5 - From braking valve; 6 - To overflow.

In figure 22 is a diagram showing the principle of such a device. The automatic device is installed on the fixed part of the wheel (the brake part) and is connected to the wheel 1 by means of gears 2 and 3; the flywheel 4 of the emitter of the automatic device is connected by means of a clutch 7 and sleeve 5 with a shaft 8 inside which is housed a push rod 6.

The sleeve is connected on one side with the flywheel 4 by means of the clutch 7 and on the other side is mounted loose on the shaft 8; however, the turning angle of the sleeve 5 on shaft 8 is limited by the size of the cam cut made in the outer end of the sleeve into which fits the perpendicular cam of the push rod 6 which at the

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same time prevents this push rod from turning with respect to the shaft 8.

The flywheel 4, which is connected with the wheel 1 by means of the clutch 7, rotates synchronously with the wheel until the moment the wheel is decelerated upon effective braking, in consequence of which loss of adjustment of the angular veloci-ties of the wheel and flywheel occurs, and the latter, continuing to rotate with the former angular velocity, turns the sleeve 5 on the shaft 8, hereby forcing out with the cam chamfers of the sleeve 5 the push rod 6, which, pressing the lever 9, closes the electric contacts of the switch 10; the latter in turn feeds current to an elec-tromagnet 11 which, displacing a valve 12 to the right, cutting off the delivery of fluid to the brakes from the operating line 13, and connecting the brake chambers with the overflow line, causes discontinuance of braking.

After the flywheel 4 has turned with the sleeve 5 to the angle corresponding to the width of the cam cut, the sleeve 5 is stopped by the twin cam of the push rod 6. The flywheel rotates, slipping on the clutch and through the lever 9 holding the con-tacts of the electric switch in the closed position.

However, after the braking of the wheel has been discontinued, it momentarily assumes an angular velocity corresponding to the progressive speed of the aircraft. By virtue of the fact that the flywheel loses a part of its speed during this time in consequence of braking of the clutch, the shaft 8 overtakes the sleeve 5, restor-ing its former position, and enables the push rod 6 to embed itself in the shaft 8 under the action of the spring 14 and disconnect the contacts of the switch 10. As a result of this the electromagnet 11 is deenergized and the valve 12 under the ac-tion of the spring 15 occupies the left position, which corresponds to braking.

In certain types of similar automatic devices, the flywheel acts directly on the brake release valve 16 of the wheel.

Experiments have shown that such an automatic device reliably prevents wheels from "binding" under the most unfavorable operating conditions on an ice-covered and partially ice-covered airfield. The sensitivity and speed of action of the device are such that it can insure up to 6 to 8 operations per second.

Another of the more important problems is that of providing for hermetic seal-ing of the units. This question is of such great importance to hydraulic assemblies that it may be stated without exaggerating that the level of constructional and pro-duction knowledge in the sphere of sealings determines the level of development of hydraulic assemblies, limiting or contributing to the adoption of these assemblies in engineering. The demands made of sealing devices have become especially high in connection with the employment of high and superhigh pressures.

Sealing devices are employed to eliminate leakage of fluid to the outside and to eliminate overflow of fluid from one (operating) cavity of a hydraulic assembly to another (non-operating) cavity.

A good-quality sealing is one which insures the required hermetic properties over a long period of service, does not cause wear of the moving parts sealed, has slight friction, is not affected by temperature, and is chemically inert towards the parts with which it comes in contact and towards the fluids employed. In addition, the sealing should be simple to produce and to use.

Internal sealing of the moving connections of slide valves and other distributing and safety assemblies, as well as of the parts of the piston group of pumps, is generally insured by accuracy of fit and high quality of machining of the parts of these connections. For example, in slide valve assemblies use is made of machining of working parts according to the tenth class of surface cleanness, with the condition that radial clearance of about 2 $\div$ 6 microns be insured. Aside from the aim of achieving small overflow, reduction of clearance contributes towards reduction of the forces of friction of the valves and slide valves and towards stabilization of these forces.

For the sealing of external connections, including the rods of slide valves and power cylinders and pump shafts, as well as for the sealing of parts which do not emerge to the outside of the assembly but which are of large diameter (more than 20 to 30 mm), use is generally made of various types of soft seals, the most widespread of which are O-shaped rubber rings a (Figures 23 and 24) which are used for sealing parts with rectilinear motion, and collars (Figures 25 and 26) for sealing rotating shafts.

Sealing by means of O-shaped rubber rings which are placed in the ring grooves of the piston or box (see Figure 24) is employed for pressures of up to 300 to 350 kg/cm^2. It is recommended that, beginning with pressures of 120 $\div$ 150 kg/cm^2, the rings be used together with leather spacing rings a (Figure 27) which prevent the ring from being forced out into the gap between the parts being sealed.

In the event the rubber rings are used without the leather spacers, the diametric gap (Figure 24) between the surfaces to be sealed must not be greater than 0.02 to 0.05mm with a fluid pressure of 100 to 200 kg/cm^2.

The possibility must be taken into account of increasing this gap at high pressures by means of elastic deformation of the parts and of insuring the required rigidity and durability of these parts.

In order that hermetic sealing will be insured, when fluid pressure is absent, the dimensions of the groove and ring selected are such that when installed in the groove the ring will be deformed (compressed) in cross section by 8 ÷ 12% of the diameter of the ring. For fixed connections, deformation of the cross section of up to 15 ÷ 20% of the diameter of the ring section is permissible. Further increase in deformation in moving connections is limited by the here increasing forces of friction and in fixed connections by installation difficulties. The ring must be installed on the neck of the groove with a tension such that the diameter of the bottom of the groove is 3 ÷ 5% greater than the internal diameter of the ring D (see Figure 23).

Figure 23. O-shaped sealing ring.

Figure 24. Location of sealing ring in groove.

Figure 25. Sealing collar:

1 - Spiral spring; 2 - Frame.

Figure 26. Reinforced type V sealing collar:

1 - Frame; 2 - Spiral spring.

Figure 27. Sealing ring with leather spacers.

If the quality of the rubber is properly selected and if the production is of good quality, the rings mentioned above will insure a high degree of hermetic sealing and a long service life, which at pressures of up to 100 kg/cm^2 may be increased to many hundred thousand runs; if the pressures are increased, the service life is shortened.

It should be noted that frequently excessively rigid demands are made of designers in the matter of hermetic sealing, demands not occasioned by practical necessity, without the fact being taken into account that meeting these increased requirements unjustifiably complicates and renders more expensive the production of hydraulic assemblies. The condition that drops of fluid not form on the rod throughout operation is often the interpretation of the requirement for complete hermetic sealing of rods emerging from an assembly. However, the formation of these drops while the assembly is in operation cannot be avoided because the sealing element scrapes a thin film of fluid from the rod during its strokes into the interior of the assembly.

For this reason, complete hermetic sealing of an assembly should be specified in requirements only for the occasions on which the aircraft will be standing. At the same time, the carrying out of fluid by moving parts emerging from the assembly should be specified; the amount which may be permitted being 5 ÷ 50cm^3 for the entire service life of the assembly.

Special attention should be devoted in production of the parts of the sealing unit to the smoothness of the surface of the groove into which the sealing ring fits, since a rough groove surface causes accelerated wear of the rings.

The outer edges of the grooves should not be rounded, since this contributes towards forcing of the ring a into the gap and destruction of it (see Figure 24); these edges should only be blunted to a radius not to exceed $0.02 \div 0.04$ mm.

Passage of a ring over any grooves, openings, etc, during operation, as well as pulling of the ring over threads, sharp edges, etc, are not permitted, since this might cause damage to the ring and destroy the hermetic sealing.

Installing bevels and curves should be provided in the parts (Figure 28). Rubber having a scleroscope hardness of 65 to 80 at 25°C is recommended. Its tensile strength should be 70 to 80 kg/cm^2. The rubber should not swell in oils by more than 3% of its volume and should not lose its elastic properties throughout the service life during operation under various temperature conditions.

Figure 28. Installation bevel of a sealing unit.

Metal springs and other means are frequently employed to increase the elasticity of sealing collars (See Figures 25 and 26).

In designing sealing connections it should be borne in mind that rubbers have a high coefficient of linear thermal expansion, which for the prevalent rubbers equals from $8 \cdot 10^{-5}$ to $11 \cdot 10^{-5}$, i.e., this coefficient exceeds practically by 10 times the coefficient for steels. In addition, in the designing of grooves, the change in the dimensions of the rings upon swelling in oil should be allowed for.

Mineral oil mixtures the components of which are selected so as to insure the necessary viscosity characteristics over the entire operating range of temperatures of the surrounding medium are employed as pressure fluids.

The requirements made of the fluids from this viewpoint should be thus formulated: at maximum positive temperatures, the fluids should have the viscosity as high as possible and at maximum negative temperatures, as low as possible. Due attention should be paid to proper oil utilization conditions.

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The requirements made of the fluids from this viewpoint should be thus formulated: at maximum positive temperatures, the fluids should have the viscosity as high as possible and at maximum negative temperatures, as low as possible. Due attention should be paid to proper oil utilization conditions.

At a temperature of plus 50°, the prevalent mixtures have a viscosity of 10 to 12 centistokes, and a viscosity of 1250 ÷ 1300 centistokes at minus 50°.

It is necessary to protect the oil from mechanical contamination by dust, moisture, air, etc., in order to avoid oxidation of the oil mixtures.

The fact should be taken into account that the intensity of oxidation of the oil depends on the temperature of contact between the oil and the oxygen -- oxidation is doubled when the temperature rises to 10°. Oxidation of the oil occurs upon local overheating of the oil due to dry friction, as well as upon compression of the air bubbles located in the oil. Oxidation is affected by the heat released upon the forcing of oil through the gaps in hydraulic assemblies.

One of the sources of contamination of oil by air is the liberation of air from a dissolved state and its transition to mechanical mixture with the oil in zones of reduced pressure. The curve of solubility of air in mineral oil shown in Figure 29 shows that the amount of air capable of being dissolved in oil is proportional to the pressure. In view of this fact, it is necessary to protect the oil from contact with air under excessive pressure. The presence of air in oil in any form -- in the form of a mechanical mixture or in a dissolved state -- is undesirable also for the reason that the operation of the pumps and individual hydraulic assemblies of the system may be impaired.

Figure 29. Air content in 1 cm^3 per 100 cm^3 of oil (P = 760 mm of mercury column and t = 0°);
1 - Volume of air, cm^3 (P = 760 mm of mercury column and t = 0°);
2 - Pressure in kg/cm^2.

It should be noted that the majority of fluids of mineral origin change their physical qualities to one extent or another during protracted kneading and mechanical action. These changes in the physical qualities are explained by the molecular structural changes taking place in the fluid during mechanical action.

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For example, instances have been noted in which the viscosity of the oil mixtures employed in aviation during a period of 500 hours of operation on a stand for testing pumps the loading of which to pressures of 150 to 200 kg/cm^2 was accomplished by throttling the fluid at its exit from the pump, comprised 50 to 60% of the viscosity which these mixtures had prior to the commencement of the experiments.

Increase in the accuracy of production of hydraulic assemblies, which is related to increase in pressure, increases the requirements for quality of filtration of the fluid. As experience has shown, improvement or deterioration of the quality of filtration of the fluid may increase or reduce by several times the service life of hydraulic assemblies. In view of this fact, use is made at the present time primarily of finely cleaned filters, the filter element of which is made from special filter paper. Because of the great resistance of such filters, they are installed only in the operating or the overflow line, or, more frequently, in both lines, and in front of important assemblies.

Such filters filter off hard micron-size particles and after repeated pouring through purify the fluid almost entirely of the hard particles.

The need for intensifying theoretical and scientific research work in the sphere of hydraulic drives has become urgent.

Increasing the volume and raising the level of these works would prove to be of vital assistance to industry in the matter of creating perfected hydraulic drives and systems.

Among the number of insufficiently studied questions one should count that of autooscillation of the units of hydraulic assemblies and systems, as well as questions concerning regulation. Questions concerning fluid filtration, erosion of parts upon cavitation of pumps, wear and increasing the service life of assemblies, etc, have likewise not been investigated with sufficient thoroughness.

The institutes of the Academy of Sciences could be of great assistance to industry in these questions by treating individual problems and questions conjointly with the interested industrial enterprises and on the basis of the latter.

I. I. PETROV AND V. G. ZUSMAN

SYSTEMS OF ELECTRIC CONTROL
OF AUTOMATIC MACHINE LINES
AND THE PRINCIPAL PROBLEMS OF THEIR FURTHER IMPROVEMENT
AND DEVELOPMENT

The following fourteen automatic machine lines have been studied at the Institute of Automation and Telemechanics of the Academy of Sciences of the USSR, in collaboration with the Experimental Scientific Research Institute for Lathes (ENIMS), for the purpose of generalization of materials dealing with the planning and operation of automatic machine lines:

(1) A261 - A268, A291 - A306, A421 - A434, and A413 - A417, for machining the cylinder blocks of trucks;

(2) A447 - A457, for machining automobile gear cases;

(3) 1S40A - 1S43 and 1S44 - 1A713, for machining the cylinder block heads of diesel tractors;

(4) 2A081 - 2A095, for machining crankcase blocks;

(5) 2A051 - 2A060, for machining cylinder blocks;

(6) I. P. Inochkin's system for machining the sealing hulls of tractors;

(7) those for machining piston pins;

(8) those for machining the rotors of electric motors;

(9) A901 - A911, A912 - A914, for machining automobile pistons and other assemblies made by automatic piston plants.

Analysis of the electric control systems of the lines named has made it possible to discover certain of their characteristics and disadvantages and has marked out a number of problems solution of which will aid in further development of systems of electric control of machine lines.

FEATURES OF SYSTEMS OF ELECTRIC CONTROL
OF AUTOMATIC MACHINE LINES

The system which has formed the basis of the automatic control systems of all the lines mentioned, with the exception of the Inochkin line, is the electroautomatic control system, which is at present the sole system permitting the most efficient automatic control of a complex of many cooperating machines; this is possible because of the following features of electroautomatic control:

(1) ease of insuring the assigned sequence of movements of the mechanisms and the technological operations;

(2) the absence of difficulties in effecting interconnections in the control of machines and mechanisms, regardless of great distances between them and peculiarities of their location with respect to each other;

(3) relatively simple accomplishment of centralization of control, supervision of sequences of operations, and observation of the operation of a complex of machines;

(4) ease of reconstruction of the working cycle of a machine and of changing the machining process to conform to the technological requirements which arise;

(5) great reliability of operation of the system and the possibility of eliminating trouble arising in the course of operation;

(6) simplicity of completing various control circuits and systems from a limited number of standard elements and devices.

As a rule, the electric control system is employed in automatic machine lines with the function of insuring performance of a specified movement by the working members of a machine, i.e., for the course function. In particular, this principle forms the basis of all the electric control systems of the machine lines under consideration. Construction of the systems for the course function is occasioned by the necessity of controlling the movement of the tool machining a part and movement of the part itself in transporting, locking, and other auxiliary mechanisms and devices. However, in addition to control for the course function, automatic control finds application also for the functions of pressure, time, and speed in individual assemblies.

Control for the pressure function is employed primarily for clamping devices and in lubrication systems. Control for the time function is generally utilized to accomplish "halting" of individual mechanisms of a line (e.g., the time needed for trimming an article during the machining process), as well as for control of the tempo of operation of the line. Control for the speed function is employed in automation of the braking processes of asynchronous motors for verification of the presence of motion of the cutting tool and verification of reduction of speed in indexing (fixed stop).

In course control systems, the basic command device is a path switch which reacts directly to the position of the controlled member of the machine or of the part being machined. In transporting devices, the position of the article moved frequently is verified by the condition of the electric circuit, which is closed by the article itself when it is in the verified zone. We may cite as an example a device for verifying the filling of an inclined slide between assemblies which is utilized for the transmission of articles (pistons) from assembly to assembly and for making slight stoppages between operations, thanks to which incomplete timing of interlinked assemblies is compensated for and a certain lag or lead of the assemblies with respect to each other is permitted.

Automation of such an inclined slide between assemblies (Figure 1) consists in the fact that in the event assembly Nr 2 stops and the slide is fully loaded, assembly Nr 1 must be stopped so that possible breakdown of the former may be avoided.

Two NVK switches are installed in the chute in one row in such a way that the piston closes them in the course of its travel (rocking) along the slide. The control circuit of the thermionic time relay ERV is therewith closed through an intermediate relay RP. In the event the NVK switches are closed (and consequently the intermediate relay engaged) for a brief period, this occurring in the event of flattening of a piston, the thermionic time relay will not have time to react. Hence, as soon as the NVK switches are opened, reading of the time lag ceases. In the event assembly Nr 2 stops for any reason, the slide is filled. Finally, a moment comes when the NVK switches have been closed so long that the time relay has had time to react, and its opening switches with the opening time lag connected in the control circuit of assembly Nr 1 open and this assembly stops.

If assembly Nr 2 starts operation and the slide is unloaded, then the NVK switches open, the intermediate and thermionic time relays are disconnected, and assembly

Nr 1 may be started again. Depending on the hook-up of the thermionic tiSTATelay switch, the assembly is started again by the adjuster or automatically.

In certain cases, indirect methods of controlling the position of the operating members are employed. For example, with hydraulic drive and in operation to support, increased pressure is utilized in the hydraulic system after the moved member reaches the assigned position. The command device in this case is a pressure relay.

Figure 1. Diagram of automation of an interassembly slide:
1 - article; 2 - interassembly slide; A - control circuit of assembly Nr 1; B - Assembly Nr 2; C - Assembly Nr 1; D - Assembly Nr 2; E - NVK switch; F - Assembly Nr 1; G - RL; H - TP; I - Thermionic time relay; J - intermediate relay.

The control systems with the course function are executed most frequently in the form of a successive action system, which is characterized by the fact that the command for each successive element of the cycle comes at the moment the preceding element has already been performed.

A feature of many automatic line control systems is the presence in them of closed control cycles for each individual machine and of interconnections between these systems, the control systems of the transporting and loading devices, and the control system of the line as a whole. The content of these interconnections is in the majority of cases determined by the giving of an initial impulse to the machines from the common control system and a return impulse from the machine which completes the cycle to the common control system. When the normal operating conditions at one of the machines are disrupted, these interconnections make possible the completion at all the remaining machines of the machining of parts which has been commenced, thanks to which the discovery of trouble occurring in the line is facilitated.

Characteristic of line control circuits is the presence in them of control units which provide for automatic, semiautomatic, and manually controlled operation. The last-named is necessary both for adjustment of individual machines and the line as a whole and for initial loading of the line with articles which have passed through the preliminary machining stages, before it is switched to automatic operation. "Cross-shaped switches" which insure observation of the rules of mnemonics are frequently employed as the controls in manually controlled operation.

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Figure 2. General view of a control panel of an automatic line for machining pistons

Figure 2 shows the control panel of the auxiliary mechanisms (horizontal and ver- tical conveyers, article clamping mechanisms, etc) of the A901 - A911 automatic line, in the operation of which the manually adjusted operation controls are disengaged, for which reason accidental or deliberate action on them does not disturb the running of the line.

Special units are provided in the connections of the most complicated automatic lines which facilitate the location of trouble in the electrical equipment. Thus, for example, on each of the machines of the type A901 - A911 line there is a signal light which indicates the machine the malfunctioning of which has caused trouble in the operation of the line.

Verification of the performance of the following operations is exercised through the signal light and the circuit unit shown in Figure 3: (a) movement of the power head from the initial position; (b) return of the head to the initial position upon completion of the cycle; (c) positioning of the head in the initial position at the moment the line is started; (d) switching on of the continuously revolving electric motor; (e) rotation of the spindles with the power head at operating speed.

When the line is switched on, as soon as the bars of the control circuit 1 and 2 are live, intermediate relay 1RP is switched on through the switches of the path switch of the initial position of the power head PV; the former, closing its normally open contacts at points 1 - 8, switches on another intermediate relay 2RP. Hence, the normally closed (n.c.) switches of this relay are open at points 1 - 13, and the safety relay 5RP is not switched on. In automatic operation, the electric motors of the power head are switched on from the intermediate relay O-1RP, which is common to the entire line. The latter with its normally open contacts switches in the contact- or K. In manually adjusted operation, this contactor is switched on and off by knobs 1KU and 2KU.

After fixing of the base plates and clamping of the centers have taken place, the "forward" command for starting the heads is given. At the same time, this command switches on the intermediate relay O-2RP, which, closing its normally open contacts

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Figure 3. Circuit for location of trouble in the electrical equipment of machine lines:

1 - Control circuit bars; 2 - 1KU; 3 - 2KU; 4 - D-1RP (intermediate relay); 5 - RT (thermal relay); 6 - Checking of continuously revolving electric motor; 7 - Path switch; 8 - 1RP (intermediate relay); 9 - Initial position; 10 - 1RP (intermediate relay); 11 - 2RP (intermediate relay); 12 - Initial position verification; 13 - 0-2RP (intermediate relay); 14 - 0-3RP (intermediate relay); 15 - 1RP (intermediate relay); 16 - 3RP (intermediate relay); 17 - Head start verification; 18 - 3RP (intermediate relay); 19 - 0-4RP (intermediate relay); 20 - 3RP (intermediate relay); 21 - 4RP (intermediate relay); 22 - 2RP (intermediate relay); 23 - 0-1RP (intermediate relay); 24 - 5RP (intermediate relay); 25 - Safety relay; 26 - RKS; 27 - 1RP (intermediate relay); 28 - 0-5RP (intermediate relay); 29 - 4RP (intermediate relay); 30 - spindle revolution verification; 31 - Return verification; 32 - 3KU; 33 - 5RP (intermediate relay); 34 - signal light; 35 - Head non-reaction verification; 36 - 5RP (intermediate relay); 37 - 5RP (intermediate relay); 38 - to preliminary stop circuit.

at points 1 - 8, shunts the normally open contacts of the 1RP relay at the same points.

As soon as the power head in its movement forward releases the path switch of the initial position, the intermediate relay 1RP is switched off and its normally open contacts at points 1 - 8 are opened. The intermediate relay 2RP remains switched on during this time only due to the closed normally open contacts of the 0-2RP relay. If the head were not in the initial position at the moment the line is switched on, the intermediate relay 1RP would not be switched off, and, consequently, its normally open contacts at points 1 - 8 would not be closed. For this reason, the intermediate relay 2RP is not switched on at the initial moment and its normally STATsed switches at points 1-13 switch on the intermediate safety relay 5RP. The latter with its normally open switches at points 15-17 shifts to self-feed, and with the other contacts at points 1-16 shunts the additional signal light resistance. In addition, the normally closed contacts of this same relay are switched in the "preliminary stop" circuit of the line, and for this reason, when the signal lamp is burning brightly, the line cannot be started "forward". In order to cut off the signal light and at the same time "permit" the line to operate, it is necessary to switch off the safety relay 5RP by means of the control knob 3KU. Since this control knob is located on the emergency machine, the adjuster is forced to approach and inspect it before pressing the knob.

The intermediate relay 2RP is switched off in the event the normally closed contacts of the thermal relay RT are opened. In this case, switching on of the contactor K and operation of the continuously revolving motor of the power head are checked.

Intermediate relays 3RP and 4RP serve for checking the starting of the head. As soon as the head leaves its initial position and the intermediate relay 1RP is switched off, its normally closed contacts at points 9-10 close and switch on the intermediate relay 3RP. The latter shifts to self-feed and simultaneously at points 11-12 breaks the circuit of the intermediate relay 4RP. The normally open contacts of the intermediate relay 0-4RP close briefly at points 1-11 at the end of the line cycle, but since the normally closed contacts of the 3RP relay are open, intermediate relay 4RP is not switched on. At the beginning of the following cycle, the switches of the intermediate relay 0-3RP are opened at points 1-9; hence the intermediate relay 3RP is removed from self-feed and prepares for operation for the following start of the head. In the event the latter for any reason does not move from the initial position, the intermediate relay 3RP is not switched on, and, consequently, at the end of the cycle when the normally open contacts of the 0-4RP relay are closed at points 1-11, the intermediate relay 4RP is switched on, and, closing its normally open contacts at points 1-17, also gives a command for switching on of the safety relay 5RP.

The system of signalling described is utilized not only for checking the operation of individual machines, but also for centralized checking of a line as a whole.

A very characteristic feature of the automatic machine lines under consideration is the limited number of functions performed by the electric drives of the machines and mechanisms forming part of them. These functions amount essentially to starting of the electric drive motors, operation of the latter at a set rate, and, in certain cases, to electric braking and stepwise changing of the speed of the motors. It is this which predetermines the utilization of the simplest and most reliably functioning electric machines, three-phase asynchronous electric motors with short-circuited rotors, as the main electric drive motors for machine lines.

For the majority of the lines in question, the average power of the main drive electric motor is 5 to 6 kilowatts, and the maximum power does not exceed 30 kilowatts.

Electronic equipment has been put to limited use in the control systems of machine lines. It is employed in circuits primarily in the form of individual devices:

thermionic time relays, intermediate thermionic relays, etc.

For the feeding of control circuits, use is made as a rule of 127-volt alternating current derived from special monophase transformers with short-circuit voltage of reduced value ($e_k = 3 - 3.5\%$). Thanks to the use of these transformers, the control circuits are isolated from the common electric circuit, and this increases the reliability of the electroautomatic line circuits. Accidental grounding of one of the conductors does not disturb normal operation of the circuit. The introduction of ground fault control makes it possible to discover and eliminate it at once. In individual instances, for example, for feeding computing devices, telephone and track relays, signalling devices, etc., low voltage current is employed.

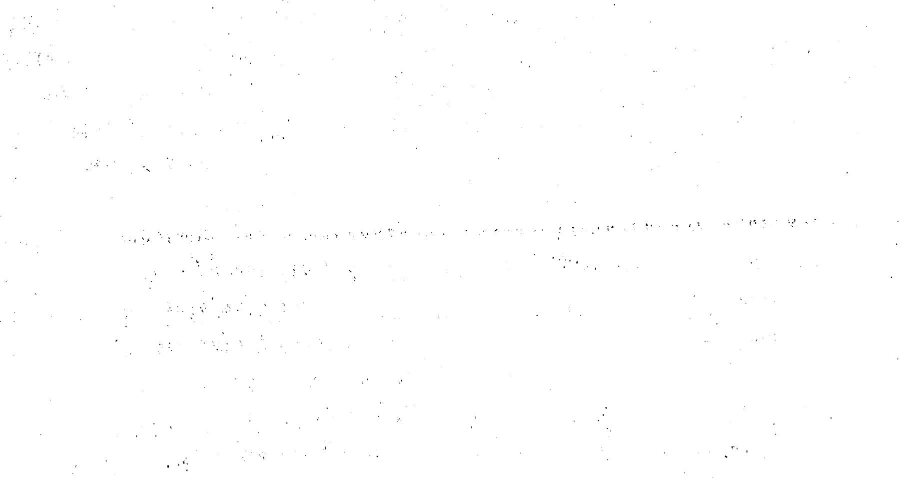

Figure 4. Skeletal diagram of the control of a centerless grinding machine:

Conventional designations:

1 - electric connections and equipment; 2 - pneumatic connections and devices; 3 - mechanical connections and operating members; 4 - pneumatic slide valve; 5 - terminal switch; 6 - pneumatic cylinder; 7 - electromagnet; 8 - electromotor; a - lug; b - forward; c - reverse; d - 1PV (path switch); e - 2PV (path switch); f - from A923; g - switch-on; h - 3PV (path switch); i - 5PV (path switch); j - 12PV (path switch); k - 6PV (path switch); l - start of A923 cycle; m - carriage; n - forward; o - reverse; p - 4PV (path switch); q - from A923 assembly; r - 11PV (path switch); s - tool holder, up, down; t - 1E1 (electromagnet); u - 2E1 (electromagnet); w - switch-off; x - 1E2 (electromagnet).

In many machine line assemblies there are mechanical, hydraulic, and pneumatic connections which operate in succession, in addition to the electric control circuits (connections). Hence the need has arisen for the creation of a unified control circuit for machine lines which includes all the forms of control. Such circuits, which have been given the name "skeletal", greatly simplify clarification of the operation

of complicated assemblies. In combination with the principle (element) electric con-
trol circuit, the skeletal circuit enables the operating personnel rapidly to dis-
cover the piece of equipment which has not responded at a given moment, and, if ne-
cessary, to make the appropriate correction.

The skeletal control circuit of a centerless grinding machine (model 3A82P) with
automatic piston starter is cited below as an example, and a description of it is
given (Figure 4). The machine is designed for the final grinding of the surface of
piston skirts. At the same time four pistons, moving two at a time from the machine,
are ground with the object of matching their weight.

Prior to commencement of the cycle, the mechanisms of the machine must be in the
following initial positions:

(a) The rod of the lug which moves the pistons along the straightedge into the
machining zone (towards the stones) is in the reverse position, and the path switch
2PV is depressed;

(b) The carriage of the piston receiver is in the forward position, and the
path switch 12PV is released;

(c) The tool holder is in the down position, and the path switch 3PV is depress-
ed;

(d) The path switched of the piston receiver 5PV and 6PV are released;

(e) The electromotor 5M serving to drive the camshaft rotates, but the catch
acting on the path switch 4PV does not depress the latter.

The operating cycle of the machine takes place as follows.

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1. The first group of pistons (two pieces) enters. Hereupon the path switches
5PV and 6PV are depressed and the electromagnet 1E - 1 is switched off through the
2PV - 5PV - 6PV circuit. The carriage is moved to the "reverse" position. The path
switches 5PV and 6PV are released and the path switch 12PV is depressed.

2. The second group of pistons enters; it also depresses the path switches 5PV
and 6PV, and switches on the electromagnet 2E - 1 through the circuit 2PV - 5PV -
6PV - 12PV. The tool holder is displaced "up", whereupon the path switch 3PV is re-
leased and the pneumatic valve of the lug cylinder is depressed.

3. Grinding of the pistons of the preceding cycle is completed. Hereupon the
valve, acting from the camshaft, through the prepared valve of the tool holder, acts
on the cylinder of the lug and the latter moves "forward", moving the next set of
pistons (four pieces) with the carriage toward the grinding stones. The path switch
2PV is released, and path switch 1PV is depressed.

4. By this time the camshaft catch is depressing the path switch 4PV and switch-
es off the electromotor 5M.

5. Upon completion of the operating cycle of the A923 machine, which is next in the chain of operat-
ions, a command impulse is given through the A923 - 1PV circuit for switching on of
the motor 5M; the camshaft starts to rotate, and the path switch 4PV is released.

6. A command is given through the circuit A923 - 1PV - 5M - 4PV to the A923 for
the beginning of a new cycle.

7. The electromagnet 2E - 2 is switched on through the circuit A923 - 1PV, and
the tool holder moves down. Hereupon the valve of the lug cylinder is released, and
at the end of the stroke the path switch 3PV is depressed.

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8. The path switch 3PV switches on the electromagnet 1E - 2 and returns the carriage to the reverse position. The path switch 3PV is released, and the carriage is prepared for the following cycle.

9. The camshaft, turning to a certain angle, acts on the pneumatic valve of the lug cylinder and returns the lug to the reverse position. Hereupon the path switch 1PV is released, the 2PV switch is depressed, and the circuits are prepared for the following cycle.

10. Blocking. In the event the A923 does not respond, at a specified moment the catch of the camshaft depresses the path switch 11PV and switches off the motor 5M, interrupting the cycle.

THE PRINCIPAL DEFECTS

OF THE ELECTRIC CONTROL SYSTEMS OF MACHINE LINES

The principal electric devices of relay-contactor line control circuits are intermediate relays, contactors, path switches, relays for various purposes, and command devices.

Analysis of the control circuits in question of automatic machine lines shows that shifting from lines designed for the machining of housing parts with simpler technological operations (drilling, threading, boring, and counterboring) to lines for the machining of parts requiring more complicated mechanical processing (turning, grinding) causes an increase in the number of electric devices allotted for one machine of a line. Thus, the average numbers of devices per machine on the lines in question for the machining of housing parts are: intermediate relays, 3.5; contactors, 3.3; path switches, 11.0; miscellaneous relays, 3.

The following are allotted per machine in lines for more complicated machining of parts, e.g., in lines for machining automobile pistons (solids of revolution): intermediate relays, 17; contactors, 7; path switches, 12; miscellaneous relays, 8.8.

Table 1

Data on the number of switch responses of the electric equipment of the 2A051 - 2A060 automatic line (machining of cylinder blocks)

Designation of switch devices	Number of devices of each designation (pieces)	Number of switches		Number of responses of devices	
		in power circuits	in control circuits	per cycle	per hour
Contactors	34	93	176	26	678
thermal relays	27	—	27	—	—
time relays	9	—	15	6	156
Intermediate relays	29	—	111	20	522
Pressure relays	31	—	31	17	444
Path switches	104	—	162	104	2713
Command devices	3	—	62	18	470
Total......	237	93	584	191	4983

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Table 1 (continued)

Designation of switch devices	Number of responses of the switches				Total number of responses of switched in power circuits and control circuits per hour
	in power circuits		in control circuits		
	per cycle	per hour	per cycle	per hour	
Contactors	78	2035	138	3600	5635
Thermal relays	--	----	--	----	----
Time relays	--	----	12	313	313
Intermediate relays	--	----	77	2009	2009
Pressure relays	--	----	17	444	444
Path switches	--	----	162	4226	4226
Command devices	--	----	74	1930	1930
Total......	78	2035	480	12522	14557

Table 2

Data on the number of switch responses of the electric equipment of the A901 - A911 automatic machine line (machining of automobile pistons)

Designation of switch devices	Number of devices of each designation (pieces)	Number of switches		Number of responses of devices	
		in power circuits	in control circuits	per cycle	per hour
Contactors	41	120	53	12	1200
Thermal relays	22	---	22	--	----
Time relays	8	---	10	7	700
Intermediate relays	127	---	430	64	6400
Pressure relays	6	---	7	6	600
Path switches	63	---	93	40	4000
Total......	267	120	615	129	12900

Designation of switch devices	Number of responses of the switches				Total number of responses of switches in power circuits and control circuits per hour
	in power circuits		in control circuits		
	per cycle	per hour	per cycle	per hour	
Contactors	70	7000	50	5000	12000
Thermal relays	--	----	--	----	----
Time relays	--	----	18	1800	1800
Intermediate relays	--	----	380	38000	38000
Pressure relays	--	----	14	1400	1400
Path switches	--	----	114	11400	11400
Total........	70	7000	576	57600	64600

The most important result of analysis of the control circuits of machine lines is the conclusion as to the uninterrupted increase in the number of switch responses of the electric devices per unit of time apace with the increasing complexity of the technological operations on machine lines. In illustration of this, the number of switch responses of the equipment on a line for machining body parts (the 2A051 - 2A060 line) is given in Table 1, and in Table 2, the same data for a line for the machining of more complex parts, specifically, solids of revolution (the A901 - A911 line). It will be seen from these tables that the number of switch responses per

hour is 14,557 on the first line, but as high as 64,600 on the second line, STAT, the
number increases by 4.5 times. Moreover, the 2A051 - 2A060 line consists of 11 ma-
chines, and the A901 - A911 of 7 machines.

At an automatic automobile piston plant, the total number of switch responses of
the equipment for all the lines of the plant reaches 300,000 per hour.

The sharp increase in the number of switch responses in lines in which machining
of the parts is more complex is occasioned by the increase in the amount of electric
equipment and the number of machining cycles per unit of time. The reliability of
operation of the electric control systems of automatic machine lines is reduced in
consequence of this.

An analysis, made by the ENIMS, of the probability of lost time on a line, in-
cluding that caused by the electric equipment, shows that the duration of lost time
due to the electric equipment increases precisely in those lines which provide for a
more complex technological process of machining parts (turning, grinding).

Together with this, because of their very nature, switch devices cannot insure
uninterrupted operation of a circuit when there are a large number of intensively
operating devices.

Thus, according to statistical data, for each million responses of a path switch
with a perfectly sound switch system, about ten non-responses are caused by incident-
al, unforeseen circumstances.

In addition, wear of the switches is a factor which limits the requirements for
the permissible number of responses of the electric equipment or its service life.
Nevertheless, the maximum number of responses of a series of devices employed in the
control circuits of machine lines is already inadequate.

In many instances it is not possible to achieve complete hermetic sealing of
switch devices which are manufactured to be waterproof. The "wick-like" property of
the leads connected to the electric devices, despite the fact that they are cased in
gas pipes or metal hoses, together with insufficiently careful sealing of the points
of connection of the leads and devices, etc., cause breaking of the hermetic sealing
of the devices. Penetration of cooling emulsions, oil, and metal dust into the cas-
ing of the devices impairs the reliability of operation of a switch system. The dif-
ficulty of achieving complete hermetic sealing of electric devices is not infrequent-
ly the reason they are installed only in places where they are not subjected to the
direct action of cooling emulsions, oil, etc. In a number of cases this results in
complication of the kinematic connections between the moving members of the devices
and the machines (for example, between the clamping devices of path switches and the
moving supports of line mechanisms).

Among other defects of the control systems operating with the course function
(with successive action) we should number the complexity of the electric circuits and
of their adjustment, the necessity of installing a large number of path switches in
the zone of operation, the necessity of for consecutive execution of commands, lead-
ing in some cases to a marked protraction of the cycle, and hence to reduction of out-
put, the unwieldiness of the control panels, and the great extent and complexity of
the electric communication lines. In addition, the necessity for periodic inspections
cleaning, and replacement of the switches of the electric devices, in view of the

location of them in places difficult of access, has a negative effect on the STAT operation of line control systems. For this reason, relay-contactor systems of automatic machine lines require not only more rational methods of construction, but also development of new principles of electric control of machine lines.

PROBLEMS OF FURTHER IMPROVEMENT AND DEVELOPMENT OF ELECTRIC CONTROL SYSTEMS OF AUTOMATIC MACHINE LINES

Employment of switchless electric devices in the control circuits of machine lines

It was established above that as the processes of mechanical working of articles in machine lines grow more complex and as the tempo of their operation is stepped up, the number of switch responses of the equipment in the control circuits increases sharply, in consequence of which the reliability of the lines is lowered. This defect may be eliminated to a great extent by the employment of switchless electrical equipment.

The predominant types of devices in control circuits are path switches and intermediate relays. It is precisely on the reliability of these devices that the accuracy of operation of the entire control system as a whole depends. Reliability of the path switches, the number of which in some cases reaches the hundreds, is of particular importance for uninterrupted operation of a line. For this reason, the gradual replacement of them by switchless displacement emitters is one of the essential tasks for increasing the reliability of the electric control systems of machine lines.

We may number among switchless emitters those with changing resistance, capacity, inductance, mutual inductance, and electromotive force. However, up to the present time not one of the emitter systems mentioned has been employed in the electric control circuits of machine lines. Hence it is necessary carefully to investigate and evaluate each system from the viewpoint of its suitability for operation in machines. In evaluation of switchless displacement emitters designed for operation as path switches, it is recommended that the following indices be taken as a basis:

(a) frequency of change of resistance ($k = \frac{z_{max}}{z_{min}}$), on which depends the reliability and accuracy of operation of an emitter, as well as the possibility of employing relays with a small coefficient of return;

(b) the possibility of changing, within fairly wide limits, the nature of the relationship of the resistance of the emitter to displacement

$$z = f(x);$$

(c) the power dissipated in the emitter, on which depends the possibility of operation of the emitter with an intermediate amplifier or without one;

(d) the possibility of automatic return of the emitter to the initial position when the action on it ceases;

(e) the magnitude of the time constant of the emitter;

(f) stability of the characteristics upon change of voltage and frequency of the supply current;

(g) constancy of the characteristics and reliability of operation of the emitter in a medium containing metallic and abrasive dust, moisture, cooling emulsions, oil, etc.;

(h) stability in operation under impacts, percussion, and [redacted]s;

(i) the presence or absence of special sources of current;

(j) simplicity and reliability of design;

(k) the overall dimensions of the emitter;

(l) the electric characteristics of the path switches needed for conformity with the parameters of the circuits which they commutate.

Particular attention in research should be devoted to the group of emitters with changing inductance, and with normal and resonance wiring diagrams. These types of emitters have a relatively great controlled power which makes it possible to wire an electromagnetic relay directly into the emitter circuits, and have an adequate resistance change frequency. Their time constant is small, they are reliable in operation under impacts, percussions, vibrations, and not sensitive to the influence of moisture, cooling emulsions, etc. In point of design, they may be made sufficiently simple and reliable, with relatively small overall dimensions. Inductive emitters are suitable for operation in alternating current circuits of both normal and increased frequency.

It is expedient to construct wholly switchless displacement emitters and switchless relays. The greatest practical interest is offered by a combination of an inductive displacement emitter and a switchless electromagnetic (choke or transformer) relay.

The employment of switchless emitter systems and speed relays in electric control circuits may also be of much interest. The group of switchless emitters and relays for continuous speed control with pickup members based on mechanical, electromechanical, and electric conversion deserve particular attention in this connection. Emitters and speed relays containing the following should be investigated and studied:

(a) a switchless pickup member and intermediate and performing members with switches;

(b) switchless pickup, intermediate, and performing members;

The employment of emitters and speed relays with switchless pickup members would apparently be expedient in all cases, since the presence of switches in a member situated directly on a machine or other mechanism of a line is undesirable. Intermediate and amplifying members may obviously be left in the form of switch relays and emitters, provided they are constructionally not connected to the pickup member and may be located in another place.

It is desirable to replace all members with switches by switchless ones for emitters and speed relays designed for operation with high frequency, with the aim of increasing the reliability and lengthening the service life of the latter.

In certain cases, it may prove expedient to employ an emitter with two switchless members (pickup and intermediate) and a performing member with switches.

The question of the possibility and advisability of employing switchless emitters and acceleration relays in the circuits of machine lines should also be investigated, in addition to that of switchless emitters and speed relays.

<u>Employment of multivalue and multicommand impulse-distribution devices in the control systems of machine lines</u>

The employment of remote control principles may be of vital importance for reduction of the extent of electric lines of communication and for increasing reliability. Worthy of particular attention are impulse-distribution devices, as well as systems employing multivalue and multicommand control devices and equipment.

In the existing control systems of machine lines, a specific function is assigned to each emitter and intermediate relay. This occasions the presence in the control circuits of a large number of impulse emitters and intermediate devices which act on the performing members (contactors, electromagnets, etc). For this reason it is advisable to shift to the new principle of control, in particular to impulse-distribution systems employed in combination.

The essence of this principle is that one emitter transmits over one connection channel successive impulses (commands) to a special distribution device which produces the necessary effect on the performing members. A system of this type when applied to the individual control units of a line can reduce considerably the number of electric devices in the circuit, as well as the extent of the electric lines of communication. The greatest effect from employment of this system should be expected in those cases in which it is necessary to send a large number of successive control impulses in order to accomplish the required working cycle of an individual unit. In addition, such a system can when necessary insure simple and rapid readjustment of the control circuit of lines. Readjustment of lines is at the present time a very intricate procedure entailing alteration of the working cycle of the lines. But this problem must be solved, since utilization of automatic machine lines in mechanical engineering will of necessity require greater universalization of them.

Reduction of the amount of relay equipment and of the number of its switch responses may also be achieved through employment of special command devices in the form of multivalue and multiposition devices the commutation position of which is determined by the number of impulses received from the path command devices.

The diagram of the mechanism and the installation of a command device are very simple. Motion is transmitted from an electromotor having a power of 0.125 kilowatt through a worm gear and change gears to a distributing drum having the necessary number of cams. The latter, acting on levers, close the appropriate control circuit. A control cycle is completed in one turn of the distributing drum. Depending on the number of command impulses, the drum turns to a certain angle (for example, 45° or 30°), this being fixed by the change gears.

Intermittent movement of the command device is accomplished as follows: After a command is given for switching on the motor of the command device, a drive shaft completes one full rotation, independently of the angle to which the drum turns, and presses a cam against a path switch which switches off the motor and engages the brake. The command device remains in this position until the following impulse is received, whereupon a rotation of the distribution drum and the corresponding switchings in the control circuit are again completed. However, when there is a large number of impulses in one control cycle, the dimensions of the device are increased considerably due to the necessity of increasing the diameter of the distribution drum. Since this is so, it is advisable to employ a command device with continuous rotation of the distribution drum, which switches the control circuits (electric, hydraulic, or pneumatic) in the required sequence.

Verification of the execution of commands given, which is accomplished in the following manner, is provided for in the operation of the control system to insure the necessary reliability. Cams which close the switches of the command circuit of the drive of the command device at those instants when the individual elements of the cycle are to end and the following to begin, are mounted on a drum seated on the same shaft as the command distribution drum. The normally open contacts of the path switches, which are located immediately by the moving members and which verify their position, are thrown on accordingly, parallel to the switches being closed by the verifying drum. The contacts of these switches are closed at the moment the contacts controlled by the drum open, if the corresponding elements of the cycle have been executed, and, consequently, rotation of the distributing drum is not interrupted. Otherwise, if an element of the cycle has not been executed by the specified time, the command device drive control circuit is opened and remains open until this element has been completed. Hence, with normal operation of the line, rotation of the command device is uninterrupted. Disruption of the cycle can be easily detected and eliminated by the introduction of signals showing interruption of the continuous rotation of the command device.

This control system possesses a number of important merits, among which we may number the following:

(1) acceptable overall dimensions of the command device, even with a large number of command impulses in the cycle;

(2) the possibility of greatly reducing the number of path command devices, thanks to the repeated use of the same path switch at various moments of the cycle;

(3) the possibility of markedly reducing the time needed for switching in the control system, by giving command impulses with a certain lead, parallel to execution of a previously given command;

Figure 5. Principle of the control circuit of a continuous-rotation command device:

1 - command verifier; 2 through 6 - path switches 1 through 5; 7 - LS (signal light).

(4) considerable reduction of the number of intermediate relays in the circuit.

Comparative analysis of the two versions of the control circuit of a two-machine automatic line with a complex cycle has shown that upon shifting from the system of consecutive control in which relay-contactor equipment is used to the system in which a continuous-rotation command device is used, the number of path command devices is cut in half, and the number of intermediate relays is reduced by more than two-thirds. Figure 5 shows the principle of the control circuit of the drive command device.

Employment of electronic equipment in the control circuits of machine lines

Electronic devices are used only to a very limited extent in present-day machine lines. They are employed primarily as individual instruments -- thermionic time relays and potentiometers. However, the sphere of their rational employment can be greatly expanded, for example, for verifying the dimensions of articles being machined, the quality of a surface being machined, and the position and proper setting of articles being machined.

Great possibilities come to light in connection with the intensive development and adoption of semiconductor germanium and silicon diodes and triodes. These new instruments, in combination with magnetic and automatic devices, make it possible to create switchless, reliable control systems of small dimensions.

M. A. GAVRILOV

APPLICATION OF THE THEORY OF RELAY-CONTACTOR CIRCUITS TO DEVELOPMENT OF ELECTRIC CONTROL SYSTEMS OF MACHINE LINES

Control of modern machine lines is accomplished in the majority of cases by means of relay circuits having the function of positioning the operating members of machines and assemblies. The principle function of these circuits is to pickup actions from these members (terminal and path switches, contacts, and various types of relays) and to transform them into actions on various types of performing members which control other members of the machines (relays, contactors, electromotors, signal lamps, electromagnets, etc). In machine lines, relay circuits also perform the functions of protecting, signalling, establishing a specific functional relationship between the actions of individual parts of a line, etc. They are characterized by the large number of devices which form part of them, by the great extent of the electric control circuits, and by the large number of switchings of the contacts within a unit of time. For this reason, correct construction of relay circuits, insuring a minimum number of elements in them, and rational selection of a structure of relay control circuits such as will insure the action necessary by them under all assigned operating conditions, are of particular importance.

Practice in planning modern relay circuits has until lately been founded on intuitive methods the basis of which has been furnished by investigation of technological requirements and theoretic building of the structure of a circuit.

Soviet scientists have developed a scientific theory of relay circuits which makes it possible to create such circuits by planning on the basis of specific, substantiated rules which permit transition from one circuit structure to another without changing their action. Application of this theory for the solution of a number of practical problems has shown that the structure of a circuit is thereby designed, and its accuracy verified, with a much smaller expenditure of time and with superior results.

Three basic problems are solved in the theory of relay circuits:

(a) the problem of synthesis, i.e., arriving at the structure of the relay circuit in accordance with the operating conditions set for it (for example, sequence in time of the operation of the members);

(b) the problem of equivalent conversions, i.e., transition from one circuit structure to another, simpler or more complex, while maintaining exactly the conformity of the structure to the operating conditions set for the circuit;

(c) the problem of analysis, i.e., determination of the operating conditions of the members of a circuit already completed (for example, determination of the sequence of their action in time, clarification of the action of the circuit in the event of damage, etc).

The basis of the theory of relay circuits is analytical recording of the latter in the form of analytical expressions (structural formulas) which characterize the relationship of the presence or absence of a closed circuit within the overall circuit as a whole to the closed or open condition of its individual members, together with a number of laws and relationships which determine the specific nature of relay circuits

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and make it possible to effect equivalent conversions of the structural formulas mentioned above.

Figure 1. Circuit of a relay impulse generator:

1 - L; 2 - b; 3 - c.

The analytical recording is based on designation of the members of relay circuits and the nature of the connections between them by specific symbols. For instance, we designate a parallel connection by the sign of algebraic addition, a series connection by the sign of algebraic multiplication, the reacting members of the circuit elements (the relay and contactor coils, signal lamps, etc) by the capital letters of the alphabet, and their contacts by the corresponding small leters. In addition, we designate closing (mormally open) contacts by a letter without a superposed dash (for example, x), and opening (normally closed) contacts, by a letter with a superposed dash (for example, $\bar{x}$). In graphic representation we show the circuit in the form of lines preceeding from a point of entry to a point of exit (the circuit supply points, for example, may be the entry and exit points), with letters corresponding to the reacting members or contacts of the elements of the circuit placed in breaks in the lines.

Figure 1 gives the circuit of a relay impulse generator (pulse pair) with the designations adopted for weakcurrent (a) and heavycurrent (b) circuits, as well as the simplified designations mentioned above (c).

By use of the symbols given above for analytical recording of circuits, the circuit (Figure 1) may be recorded in the form of the following expression:

$$F = F_{(X_1)}X_1 + F_{(X_2)}X_2 + F_{(L)}L = a\bar{x}_2\bar{x}_1 + x_1X_2 + x_1L,$$

where F is the symbol designating the entire circuit as a whole, and $F_{(X_1)}$, $F_{(X_2)}$, and $F_{(L)}$ are the symbols designating the circuits which act on the individual reacting members of the circuit.

In Figure 2 is given, in the ordinary (a) and simplified (b) designations, the stopping, starting, and reversal circuit of an asynchronous motor with short-circuited rotor. In this diagram, N and V are the "reverse" and "forward" movement contactors; k_n and k_v, the contacts of the "reverse" and "forward" start button; k_s, the contacts of the "stop" button; and T, the thermal shield. The analytical recording of this circuit has the following form:

$$F = \bar{k} \cdot [k_v(\bar{k}_n + N)\bar{v}N + \bar{k}_n(k_v + v)\bar{N} \cdot B]\,\bar{t}.$$

The structural formulas of a circuit make it possible to reach a conclusion as to the composition of the circuits present in the entire circuit as a whole or acting on its individual elements; however, they do not give an idea of the sequence of action of the circuit elements in time, and this is necessary for circuits in which

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such sequence is provided for.

Figure 2. Diagram of the starting, stopping, and reversal of an asynchronous short-circuited electric motor:
$1 - K_s$; $2 - K_v$; $3 - K_n$; $4 - N$; $5 - K_n$; $6 - K_v$; $7 - V$; $8 - V$; $9 - N$; $10 - N$; $11 - V$; $12 - \bar{K}_s$; $13 - \bar{K}_v$; $14 - \bar{K}_n$; $15 - N$; $16 - K_n$; $17 - K_v$; $18 - v$; $19 - \bar{v}$; $20 - \bar{N}$; $21 - N$; $22 - V$; $23 - b$.

For the data of so-called "multiple cycle" circuits, it is convenient to record the sequence of switching of the elements in the form of a table in the columns of which is given the condition of the circuit elements for the various cycles of its operation. The operating cycle ("takt") of a circuit is defined as that condition of its elements which differs from the preceding condition in change of position of at least one element.

We shall designate the response of an element by the plus sign and release by the minus sign. The sequence of operation of the elements of a circuit (Figure 1) may then be recorded in the form of a table of switchings (Figure 3), from which it may be seen that when element A is switched on, elements X_1 and X_2 begin to respond and release in succession, sending a pulsating current to the signal lamp L.

0	1	2	3	4	5	6
$-A$	$+A$					
$-X_1$		$+X_1$		$-X_1$		$+X_1$
$-X_2$			$+X_2$		$-X_2$	
$-L$			$+L$		$-L$	

etc.

Figure 3. Table of switchings of the circuit shown in Figure 1.

The sequence of operation of the elements of the circuit in Figure 2 must be recorded in the form of several switching tables, since it may vary in relation to the sequence of action on the pickup elements of the circuit. A table of switchings for the event of action on the button K_n is shown in Figure 4a. As may be seen from this table, when the contacts of this button are closed, the reverse movement contactor N is switched on and remains switched on until the button K_s has been acted on. Such a sequence also takes place in the event of action on the button K_v (Figure 4c and 4d). It will be seen from these tables that in the event of action on the button for movement opposite that taking place at a given moment, first the contactor in the "on" position must be switched off, and then the contactor of the opposite direction of movement switched on. The sequence of switching upon response of the thermal shield need not be recorded, since the action of the circuit elements in this case is identical with that upon switching on of the button K_s.

The structural formulas and switching tables give a fairly clear picture both of the structure of the circuit itself and of the sequence of operation of its elements.

0	1	2	3	4	5	6	1	2	3	4	5	6
$-K_N$	$\not- K_N$		$-K_N$									
$-K_V$							$\not- K_V$		$-K_V$			
$-K_S$				$\not- K_S$		$-K_S$				$\not- K_S$		
$-N$		$\not- N$			$-N$							
$-V$								$\not- V$			$-V$	

a b

0	1	2	3	4	5	6	7	1	2	3	4	5	6	7
$-K_N$	$\not- K_N$		$-K_N$								$\not- K_N$			$-K_N$
$-K_V$				$\not- K_V$			$-K_V$	$\not- K_V$		$-K_V$				
$-K_S$														
$-N$		$\not- N$			$-N$								$\not- N$	
$-V$						$\not- V$			$\not- V$			$-V$		

c d

Figure 4. Table of switchings of the circuit shown in Figure 2.

In addition to this, the possibility of analytical recording of the structure of a circuit in the form of a certain algebraic expression makes it possible to effect conversion of the structure of the circuit into another structure without disrupting the action of the circuit itself. This makes it possible in synthesis of the circuit not to be concerned over obtaining the simplest circuit at the very beginning. The only important thing is to arrive at a structure, so long as it conforms to the assigned operating conditions. It is then easy to shift to any desired structure by making equivalent conversions of the structure.

The table of switchings of the elements of the circuit represents in essence the operating conditions of the latter assigned in the form of sequence of action of the individual elements. Such a sequence may obviously be recorded preliminarily on the basis of analysis of the actions, both those taken on the circuit and those effected by the circuit itself. This suffices for reaching a decision as to the possibility of creating the circuit, the structure of its individual circuits acting on specific elements, and the entire circuit as a whole.

We shall discuss in greater detail the features of structural formulas and switching tables.

The structural formula as an analytical expression has the characteristic that each of its members may have only two values. The specific character of relay circuits based on the employment of elements with relay action lies in the presence of only two conditions of the circuits affecting them: one in which the element in question reacts, and another in which it does not. This appears the most clearly in cases in which there are only electric contacts in the circuits acting on a given element. In this case, the structural formula and each of its members can have only two values: one corresponding to the closed circuit and the other to the open circuit. In the case of relay action of the performing members of a circuit, the action of any other circuit is reduced to these two values. Let us say, for example, that there is in the circuit of a relay element resistance the amount of which changes smoothly from zero to infinity. Then all the values which this resistance assumes may be divided into two groups. To the first group we must assign all the amounts of resistance with which the reacting member of the relay element in question, being acted on by the circuit, does not respond; and to the second, all the amounts of resistance with which this element does respond. The latter amounts may be assigned the same status as the completely closed circuit of the element, since the element will respond with any of them.

This feature of relay circuits -- the fact that the members of the structural formulas characterizing them have only two values -- determines the specific nature of the relationships arising in relay circuits and forming the peculiar algebra of relay circuits.

Let us examine the principal relationships of this algebra. We find that

$$x \cdot x = x, \tag{1}$$

$$x + x = x. \tag{1a}$$

This follows from the fact that the circuit, consisting of identical, series or parallel connected contacts of the same relay element, is equivalent in its action to one contact of this element (Figure 5a and b).

Figure 5. Diagrams illustrating the equivalency of (1) and (1a):

1 - b.

Relationships (1) and (1a) attest to the fact that there are neither powers nor coefficients in the algebra of relay circuits. The laws of transposition, combination, and distribution of ordinary algebra are valid for this algebra. For example, the laws of transposition of ordinary algebra recorded in the form of structural formulas will appear thus:

$$xy = yx, \tag{2}$$

$$x + y = y + x. \tag{2a}$$

The circuits corresponding to these expressions are shown in Figure 6a and b.

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They are mutually equivalent, since with parallel or series connection of the e-
ments, the order of this connection is of indifference for the action of the circuit.
Apart from the question of dependency on the order in which the contacts will be con-
nected in the circuit of an element, when they are connected in series, the circuit
is closed only when all the contacts are closed, and when they are connected in paral-
lel, when at least one of them is closed.

The laws of combination of ordinary algebra are recorded thus in the form of structural formulas:

$$(xy)z = x(yz), \quad (3)$$

$$(x + y) + z = x + (y + z). \quad (3a)$$

Figure 6. Diagrams illustrating the equivalency of (2) and (2a): 1 - b.

Figure 7. Diagrams illustrating the equivalency of (3) and (3a): 1 - b.

The corresponding diagrams are given in figure 7a and b. The equivalency of
these circuits derives from the fact that with parallel or series connection of the
three elements in the circuit, it makes absolutely no difference how we regard its
formation, whether from the closing of the first two elements and then of the third,
or from the closing of the first element and then of the other two.

The structural formula of the law of distribution of ordinary algebra assumes the following form:

$$x(y + z) = xy + xz. \quad (4)$$

Figure 8. Diagrams illustrating the equivalency of (4) and (4a): 1 - b.

The diagram corresponding to it is shown in Figure 8a. It will readily be seen
that the circuits depicted in the left and right parts of this figure act in exactly
the same manner. In both of them the circuit is closed only when contacts x and y or
x and z are closed.

All the rules for placing in parentheses and removal of parentheses employed in
ordinary algebra are valid for the transposition of structural formulas. However, in
contrast to the former, the specific nature of relay connections stipulates its own
laws, to which belong, besides the law of "repetition" (relationships 1 and 1a), the
distributive law of addition and/or multiplication and the laws of inversion. The
first law is the inverse with respect to the distributive law of multiplication and/
or addition (4) and is derived from it by replacement of all the laws by the inverse

ones; it is recorded as follows:

$$x + yz = (x + y)(x + z). \qquad (4a)$$

The corresponding connections are shown in Figure 8b. They are mutually equivalent, since in the right and left connections a closed circuit is formed when x or y and z is closed.

The laws of inversion in the algebra of relay connections make it possible to shift from one connection to another having the opposite effect, one closed when the initial connection is open, and vice versa. Such an inverse connection is designated by a dash over the structural formula characterizing the original connection. This sign has already been used earlier to designate the closing contacts of the elements of a connection, since the latter represent the most elementary inverse connection with respect to closing contacts.

Figure 9. Connections illustrating the laws of inversion reflected in formulas (5) and (5a):
1 - opposite; 2 - b.

Figure 10. Connections illustrating relationships (6) and (6a):
1 = b.

The laws of inversion are recorded as follows:

$$\overline{xy} = \bar{x} + \bar{y}, \qquad (5)$$

$$\overline{x + y} = \bar{x} \cdot \bar{y}.$$

The connections corresponding to these equations are shown in Figure 9a and b. The connections corresponding to the expressions under the dash in the left part of the equations (the connections on the left in Figure 9a and b) have an effect opposite that of the connections corresponding to the right parts of these equations (the connections on the right in Figure 9a and b).

The laws discussed above are valid both for individual contacts and for groups of them, as well as for parts of connections connected in parallel or in series. All these laws are symmetrical with respect to the operations of addition and multiplication and each of them marked by a numeral and the letter "a" can be derived from the law designated by a numeral without a letter. It is possible in them to replace all the signs of addition by signs of multiplication and vice versa.

The algebra of relay connections is the algebra of two numbers, one of which must correspond to the closed condition of the circuit and the other to the open condition. Let us determine what these numbers must be. In ordinary algebra, one is the coefficient of multiplication, the number which does not change its value when multiplied by any other:

$$x \cdot 1 = x, \qquad (6)$$

and zero is the coefficient of addition:

$$x + 0 = x. \qquad (6a)$$

Let us determine what this corresponds to in relay connections. For this purpose, let us imagine expressions (6) and (6a) to be the structural formulas of certain connections. The connection (Figure 10a) corresponding to expression (6) means that if any element (or group of elements) is connected in series with an unknown circuit corresponding to one, then the action of the circuit in which the element in question is connected does not change. This will hold true only in the event that the circuit corresponding to one is constantly closed.

The connection (Figure 10b) corresponding to expression (6a) means that if any element (or group of elements) is connected in parallel with an unknown circuit corresponding to zero, then the action of the circuit in which the element in question is connected does not change. This can happen only if the circuit corresponding to zero is constantly open.

Hence the two values of the members of structural formulas can be expressed by one, which corresponds to the closed circuit, and by zero, which corresponds to the open condition of the circuit. The following relationships occur here:

$$x + 1 = 1, \qquad (7)$$

$$x \cdot 0 = 0, \qquad (7a)$$

from which it follows that any circuit connected in parallel with a constantly closed circuit will also be closed, and that any circuit connected in series with a constantly open circuit, will also be open.

Two other relationships hold true:

$$x\bar{x} = 0, \qquad (8)$$

$$x + \bar{x} = 1. \qquad (8a)$$

The first follows from the fact that with a series connection of the closing and opening contacts of one and the same element, the circuit will always be open, since with any position of this element one of the contacts will be open. The second relationship is determined by the fact that with a parallel connection of the closing and opening contacts of one and the same element, the circuit will always be closed, since with any position of this element one of its contacts will be closed.

Using relationships (6), (6a), (7), and (7a) as a guide, we can obtain the following equations:

$$0 \cdot 0 = 0, \qquad (9)$$

$$0 + 0 = 0, \qquad (9a)$$

$$1 \cdot 1 = 1, \qquad (10)$$

$$1 + 1 = 1, \qquad (10a)$$

$$0 \cdot 1 = 0, \qquad (11)$$

$$0 + 1 = 1. \qquad (11a)$$

From the very definition of zero and one, as well as of the concept of inversion, it follows that

$$\bar{0} = 1, \qquad (12)$$

$$\bar{1} = 0. \qquad (12a)$$

The relationship established above between the closed and open condition of the circuits of a connection and the numerical values of the members of the structural formulas, as well as the formulas obtained above for operations with these values, make it possible to determine through computations the presence of a closed or open circuit in the connection. If there is at least one closed circuit in the STAT ction, then the value of its structural formula as a whole must equal one. If there is not a single closed circuit in the connection, then its value must equal zero. In principle this makes it possible to solve the problem of analysis of relay connections, i.e. determine from a given connection the conditions of its operation, by compiling a table of the switchings of its elements.

Let us say, for example, that the connection shown in Figure 11 has been assigned. The structural formula for it will appear in the following form:

$$F = \bar{x}_4(x_2X_3 + aX_1) + x_3(X_1 + X_4) + x_4x_2X_4 + x_1X_2.$$

Removing the parentheses and grouping the members with X_1, X_2, X_3, and X_4, we obtain

$$F = (a\bar{x}_4 + x_3)X_1 + x_1X_2 + x_2\bar{x}_4X_3 + (x_3 + x_2x_4)X_4.$$

The members contained in parentheses with the symbols of the reacting members of the elements of the connection represent the circuits acting on these elements. Hence,

$$F_{(X_1)} = a\bar{x}_4 + x_3; \quad F_{(X_2)} = x_1;$$
$$F_{(X_3)} = x_2\bar{x}_4; \quad F_{(X_4)} = x_3 + x_2x_4.$$

Figure 11. Diagram of a multicycle pulsing device.

Let us determine the initial position of the elements of the connection. For this purpose let us assume that the connection is dead and all its elements in the released condition. All the closing contacts of the elements will then be open (equal to zero) and the opening contacts closed (equal to one). Substituting these values in the structural formulas of the circuits acting on the elements of the connection, we obtain

$$F_{(X_1)} = 0 \cdot 1 + 0 = 0; \quad F_{(X_2)} = 0;$$
$$F_{(X_3)} = 0 \cdot 1 = 0; \quad F_{(X_4)} = 0 + 0 \cdot 0 = 0.$$

This attests to the fact that in the released condition a closed circuit is not formed for any of them and that when the connection is switched on, all the elements remain static. Thus, in the zero cycle of the connection (Figure 12) all its elements are switched off.

Let us now switch on element A (cycle 1 in the switching table, Figure 12). The structural formulas of the circuits acting on the elements of the connection hereupon assume the following values:

$$F_{(X_1)} = 1 \cdot 1 + 0 = 1; \quad F_{(X_2)} = 0;$$
$$F_{(X_3)} = 0 \cdot 1 = 0; \quad F_{(X_4)} = 0 + 0 \cdot 0 = 0.$$

The transition of $F_{(X_1)}$ from 0 to 1 means that a closed circuit is created for element X_1 in the first cycle, and it is thus switched on in the second cycle. The circuit $F_{(X_2)}$ hereupon becomes equal to one, causing the switching on of the element X_2 in the third cycle, etc. The subsequent values of the structural formulas of the circuits acting on the elements of the connection are given in the upper part of the switching table in Figure 12 and the corresponding conditions of the elements of the connection in the lower part.

CYCLES		0	1	2	3	4	5	6	7	8	9	10
Values of structural formulas: $F(X_1) = a\bar{x}_4 + x_3 =$		0	1	1	1	1	1	0	0	0	1	1
$F(X_2) = x_1 =$		0	0	1	1	1	1	1	0	0	0	1
$F(X_3) = x_4x_2 =$		0	0	0	1	1	0	0	0	0	0	0
$R(X_4) = x_3 + x_2x_4 =$		0	0	0	0	1	1	1	1	0	[illegible]	0
Conditions of the elements of the connection	$-A$	$-A$	$+A$									
	$-X_1$	$-X_1$		$+X_1$					$-X_1$			$+X_1$
	$-X_2$	$-X_2$			$+X_2$					$-X_2$		
	$-X_3$	$-X_3$				$+X_3$		$-X_3$				
	$-X_4$	$-X_4$					$+X_4$				$-X_4$	

Figure 12. Table of switchings of the elements of the connection shown in Figure 11.

The laws of relay connections and the relationships with zero and one make it possible to obtain certain general formulas for conversion of these connections. For example, let there be a structural formula

$$F = x + xy.$$

Removing x from parentheses, we obtain

$$F = x(1 + y) = x \cdot 1 = x.$$

In this instance this is equivalent to the condition in which the element x in the second member of the structural formula equals zero.

Let there be a somewhat different structural formula:

$$F = x + \bar{x}y.$$

Resolving it in accordance with the distributive law of addition and/or multiplication (4a), we obtain

$$F = (x + \bar{x})(x + y).$$

However, $x + \bar{x} = 1$; hence,

$$F = x + \bar{x}y = x + y.$$

The element x in the second member of the structural formula thus equals one.

In the most general instance, when there is a connection $f(x, y, \ldots, w)$ variously containing the elements $x, y, \ldots, w$, and connected in parallel to a closing contact x, all the circuits in it containing x may in the same way (i.e., by removal of x from parentheses) be reduced to one x, and all the circuits containing $\bar{x}$ may be replaced by circuits which do not contain this member by application of the distributive law of addition and/or multiplication. We may thus record in a general form:

$$x + f(x, y, \ldots, w) = x + f(0, y, \ldots, w). \quad (13)$$

This means that if a connection is connected in parallel to the closing contact of any element, then all the closing contacts of a given element within this connection can be made to equal zero (opened) and all the opening contacts to equal one (closed). Similarly, we can write:

$$\bar{x} + f(x, y, \ldots, w) = \bar{x} + f(1, y, \ldots, w). \quad (14)$$

Let us consider an instance in which a connection is connected in series to a contact x. For example, let there be a structural formula

$$F = x(x + y).$$

Removing the parentheses, we obtain

$$F = x \cdot x + xy = x + xy = x.$$

The contact x here could thus have been assumed to equal zero.

Let there be a structural formula

$$F = x(\bar{x} + y).$$

Then, removing the parentheses, we have

$$F = x\bar{x} + xy = xy.$$

In this case, the contact $\bar{x}$ within the parentheses may thus be made to equal zero. Drawing an analogy with respect to a certain general connection $f(x, y, \ldots, w)$, we obtain

$$xf(x, y, \ldots, w) = xf(1, y, \ldots, w). \quad (15)$$

This means that if a connection is connected in series to the closing contact of any element, then all the closing contacts of a given element within this connection can be made to equal one (closed) and all the opening contacts to equal zero (closed). Similarly, we can write:

$$\bar{x}f(x, y, \ldots, w) = \bar{x}f(0, y, \ldots, w). \quad (15a)$$

The formulas obtained permit fairly rapid and effective simplification of relay connections.

Let us take, for example, the connection shown in Figure 13a. The structural formula for it will have the following form:

$$F = x[(y + b)x + y(z + \bar{x}y)] + \bar{x}[(y + d)x + (a + \bar{a}\bar{x})y].$$

Using formulas (15) and (15a), we find:

$$F = x[(y + b) \cdot 1 + y(z + 0 \cdot y)] + \bar{x}[(y + d) \cdot 0 + (a + \bar{a} \cdot 1)y] =$$
$$= x(y + b + yz) + \bar{x}(a + \bar{a})y,$$

and, employing formula (13) for the first parenthesis and taking into account the fact that $a + \bar{a} = 1$ and $x + \bar{x} = 1$, we obtain

$$F = x(y + b) + \bar{x}y = (x + \bar{x})y + xb = y + xb.$$

The connection of Figure 13a is thus equivalent in its action to that of the connection of Figure 13b, which has a much fewer number of contacts.

Let us examine the problem of synthesis of relay connections, which consists of the ability of shifting from the operating conditions assigned for a connection to a connection structure satisfying these conditions. The availability of the switching tables makes the obtaining of such a structural formula a fairly simple matter, since the condition of the elements in each of the cycles of the switching table is the condition of closing of the circuit for the element responding in the next cycle. Thus, for example, the element x_1 in the switching table in Figure 12 responds in the second cycle, hence the closed circuit for it must be created in the first cycle, in which the condition of the elements is recorded in the form of the product

$$f_1 = \overline{ax_1}\bar{x}_2\bar{x}_3\bar{x}_4,$$

which represents the structural formula of the circuit which closes in the first cycle. If such products are written out for a number of cycles and their total taken, they will give the structural formula of the circuits closed in all these cycles.

Figure 13. Illustration of a connection before and after simplifications are made.

Thus, for the switching of any element in all the cycles of a table it is adequate to write out the products of the symbols of the contacts of the elements of the connection corresponding to their conditions in all the cycles in which the circuit of the element in question must be closed (for example, these cycles are 1, 2, 3, 4, and 5 for the element X_1 of the switching table in Figure 12). Hence one of the main tasks of synthesis is compilation of a table of switchings on the basis of the operating conditions assigned for the connection. This is easily accomplished in the case of its pickup and performing elements, since the sequence of their action follows directly from the list of the actions exercised on the connection from without, and the list of these actions which the connection itself must exercise on the mechanisms controlled by it. For example, the table of switchings in Figure 4 for the starting, stopping, and reversal connection of an asynchronous motor can easily be compiled on the basis of analysis of the order of the actions on the buttons K_n, K_v, and K_s for switching the contactors N and V on and off. However, the sequence of switching of the elements of a connection cannot always be secured solely by means of the pickup and performing elements. In a number of cases conversion of the actions exercised on the pickup elements of a connection from without into a set sequence of action of the performing elements necessitates the switching of intermediate elements into the connection. Their number and point of connection are readily determined by analysis of the feasibility of the switching table. The latter may be realized in the form of a connection only in the event that, for each of its elements, not one of the combinations corresponding to those cycles in which the circuit is closed, is repeated in the cycles in which it is open.

Identical combinations of the elements may conveniently be determined by computation of their ordinal numbers in a system of numerals with the basis of 2 (the system of binary numbers). For example, we designate the individual elements in the table of switchings in Figure 12 by the numbers 2^0, 2^1, 2^2, 2^3, and 2^4 and shall designate the condition of the elements of the connection in each of the cycles by the sum of these numbers, which correspond to those elements which are switched on in this cycle (see the table of switchings in Figure 14). A comparison of the numbers obtained, which are given in the bottom line of this table, shows that there are no contradictory conditions for any one of the elements, since not one of the numerals characterizing the condition of the elements in the various cycles is repeated.

CYCLES		0	1	2	3	4	5	6	7	8	9	10	
BINARY NUMBERS	2^0	$-A$	$\not-A$										
	2^1	$-X_1$		$\not-X_1$					$-X_1$			$\not-X_1$	
	2^2	$-X_2$			$\not-X_2$					$-X_2$			etc.
	2^3	$-X_3$				$\not-X_3$		$-X_3$					
	2^4	$-X_4$					$\not-X_4$				$-X_4$		
NUMBER OF COMBINATIONS		0	1	3	7	15	31	23	21	17	1	3	

Figure 14. Table of switchings of the elements of the connection shown in Figure 11, with the combinations of the elements determined by the system of binary numbers.

CYCLES		0	1	2	3	4	5	6	7	8	9	10
BINARY NUMBERS	2^0	$-A$	$\not-A$		$-A$			$\not-A$		$-A$		
	2^1	$-X_1$		$\not-X_1$			$-X_1$		$\not-X_1$		$-X_1$	
	2^2	$-X_2$				$\not-X_2$						$-X_2$
NUMBER OF COMBINATIONS		0	1	3	2	6	4	5	7	6	4	0

a

CYCLES		0	1	2	3	4	5	6	7	8	9	10
BINARY NUMBERS	2^0	$-A$	$\not-A$		$-A$			$\not-A$		$-A$		
	2^1	$-X_1$		$\not-X_1$			$-X_1$		$\not-X_1$		$-X_1$	
	2^2	$-X_2$				$\not-X_2$						$-X_2$
	2^3	$-X_3$							$\not-X_3$			$-X_3$
NUMBER OF COMBINATIONS		0	1	2	2	6	4	5	15	14	12	0

b

Figure 15. Example of non-feasibility and feasibility of a table of switchings of elements.

Another situation arises in the table in Figure 15a, in which there are repeating combinations (6 in cycles 4 and 8; 4 in cycles 5 and 9). For the element X_1, they fall in cycles in which its circuit must be open, hence the switching table is feasible for this element. However, for the element X_2, the combination 4 falls once in a cycle in which the circuit of this element must be closed (fifth cycle), and another time in a cycle in which the circuit must be open (ninth cycle). For this reason the table of switchings is not feasible for the element in question. In order to render it feasible, it is necessary to switch in a supplementary element, compelling it to respond and release in such a way as to eliminate the recurrent numbers of the combinations of the elements. Usually the very arrangement of these numbers indicates where the supplementary element should be switched in. For example, the table in Figure 15a is rendered feasible by the switching in of a supplementary element X_3 in cycles 7, 8, and 9 (see the table of switchings in Figure 15b).

Let us examine the question of separation from a table of switchings tables from which it would be possible to obtain the structural formulas of circuits which affect each of the elements.

At the minimum, those elements which change their position in the cycle preceding both the response and release of a given element should enter into the composition of the elements affecting this element. The elements the contacts of which participate in the circuit of the element in question should create for it a feasible table of switchings, for the compilation of which the following rules may be formulated.

1. With the assigned switching table for the entire connection as a whole used as the basis, a table is compiled which contains only those elements which change their position in the cycles f_{res} and f_{rel} for a given element.

2. If the table thus obtained is not feasible, one or more elements are sought out, from among those in the assigned general table, which will render it feasible when written out in the same sequence of switching on and off as is given in the general table.

3. If there are no such elements in the general table, then the switching in of supplementary, intermediate elements is resorted to. In this case it is advisable to study jointly all the elements for which non-feasible conditions are obtained, in order to achieve feasible switching tables.

Let us examine the synthesis of relay connections through the example of the switching table in Figure 14. We write out the sequence of switching of the elements A and X_3, which change their positions in cycles f_{res} and f_{rel} for the element X_1. The corresponding table of switchings is shown in Figure 16a, from the bottom line of which it will be seen that it is not feasible, since it has recurrent numbers of combinations in cycles 1 and 7, 2 and 6, 3 and 6. In order to render it feasible, it is necessary to add to it the sequence of action of an element which through its being switched in would bridge over the cycles having recurrent numbers of combinations (6 - 7 or 1 - 3). X_4 (Figure 16b) is such an element.

In order to determine the structural formula of the circuits acting on the element X_1, we write out the condition of the elements in the table shown in Figure 16b, in all the cycles in which the circuit of this element must be closed (cycles 1, 2, 3, 4, and 5):

$$F_{(X_1)} = a\bar{\bar{x}}_1\bar{\bar{x}}_3\bar{\bar{x}}_4 + a\bar{\bar{x}}_1\bar{\bar{x}}_3\bar{\bar{x}}_4 + a\bar{\bar{x}}_1\bar{\bar{x}}_3\dot{x}_4 + a\bar{\bar{x}}_1\bar{\bar{x}}_3\bar{\bar{x}}_4 + a\bar{\bar{x}}_1\bar{\bar{x}}_3\bar{\bar{x}}_4.$$

Placing the common members outside the parentheses and making the appropriate simplifications, we have:

$$F_{(X_1)} = a[\bar{\bar{x}}_3\bar{\bar{x}}_4(\bar{\bar{x}}_1 + x_1) + x_1\bar{\bar{x}}_4(\bar{\bar{x}}_3 + x_3) + x_1x_3x_4] = a(\bar{\bar{x}}_3\bar{\bar{x}}_4 + x_1\bar{\bar{x}}_4 + x_1x_3x_4) =$$

$$a[\bar{\bar{x}}_4(\bar{\bar{x}}_1 + \bar{\bar{x}}_3) + x_1x_3x_4]$$

The sequence of switching on and off of the elements of a connection makes it possible to effect additional simplifications, since with a set sequence of switching of the elements certain combinations are never encountered; hence the corresponding circuits are always open. Thus, for example, if it is known from the switching table

CYCLES		0	1	2	3	4	5	6	7	8	9	10
BINARY NUMBERS	2^0	-A	-/-A									
	2^1	$-X_1$		$-/-X_1$					$-X_1$			$-/-X_1$
	2^3	$-X_3$				$-/-X_3$		$-X_3$				
NUMBER OF COMBINATIONS		0	1	3	3	11	11	3	1	1	1	3

a

CYCLES		0	1	2	3	4	5	6	7	8	9	10
	2^0	-A	-/-A									
	2^1	$-X_1$		$-/-X_1$					$-X_1$			$-/-X_1$
	2^3	$-X_3$				$-/-X_3$		$-X_3$				
	2^4	$-X_4$					$-/-X_4$				$-X_4$	
NUMBER OF COMBINATIONS		0	1	3	3	11	27	19	17	17	1	3

Figure 16. Example of the non-feasibility and feasibility of a table of switching of elements.

that an element X_1 is switched on before an element X_2 and is switched off after it or at the same time, then the condition of element X_2 being switched on and X_1 switched off at the same time will not be encountered in the table, owing to which

$$\bar{\bar{x}}_1 \cdot x_2 = 0. \tag{16}$$

Taking the inversion of this expression, we find:

$$x_1 + \bar{\bar{x}}_2 = 1. \tag{16a}$$

By adding expression (16) (as a circuit always open) to the various other combinations of the contacts of elements X_1 and X_2, one may obtain a number of relationships which permit simplification of the structural formulas when there is an assigned frequency of their switching.

Thus, for example:

$$x_1x_2 = x_1x_2 + \bar{\bar{x}}_1x_2 = x_2(x_1 + \bar{\bar{x}}_1) = x_2. \tag{17}$$

Taking the inversion of this expression, we find:

$$\bar{\bar{x}}_1 + \bar{\bar{x}}_2 = \bar{\bar{x}}_2. \tag{17a}$$

Similarly,

$$\bar{\bar{x}}_1\bar{\bar{x}}_2 = \bar{\bar{x}}_1\bar{\bar{x}}_2 + \bar{\bar{x}}_1x_2 = \bar{\bar{x}}_1(\bar{\bar{x}}_2 + x_2) = \bar{\bar{x}}_1. \tag{18}$$

The inversion of this expression has the following form:

$$x_1 + x_2 = x_1. \tag{18a}$$

Remark.

In the theory of relay connections there are the following relationships for the different sequences of switching of the elements:

1. Closing in the order of the indices, opening simultaneously or in the reverse order:

STAT

$$\bar{x}_1 x_2 = 0, \qquad (16)$$

$$x_1 + \bar{x}_2 = 1, \qquad (16a)$$

$$\bar{x}_1 x_2 = \bar{x}_2, \qquad (17)$$

$$\bar{\bar{x}}_1 + \bar{x}_2 = \bar{x}_2, \qquad (17a)$$

$$\bar{x}_1 \bar{x}_2 = \bar{x}_1. \qquad (18)$$

$$x_1 + x_2 = x_1. \qquad (18a)$$

2. Opening in the order of the indices, closing simultaneously or in the reverse order:

$$x_1 \bar{x}_2 = 0, \qquad (19)$$

$$\bar{x}_1 + x_2 = 1, \qquad (19a)$$

$$x_1 x_2 = x_1, \qquad (20)$$

$$\bar{x}_1 + \bar{x}_2 = \bar{x}_1, \qquad (20a)$$

$$\bar{x}_1 \bar{x}_2 = \bar{x}_2, \qquad (21)$$

$$x_1 + x_2 = x_2. \qquad (21a)$$

3. Absence of simultaneous switching on:

$$x_1 x_2 = 0, \qquad (22)$$

$$\bar{x}_1 + \bar{x}_2 = 1, \qquad (22a)$$

$$\bar{x}_1 x_2 = x_2, \qquad (23)$$

$$x_1 + \bar{x}_2 = \bar{x}_2, \qquad (23a)$$

$$x_1 \bar{x}_2 = x_1. \qquad (24)$$

$$\bar{x}_1 + x_2 = \bar{x}_1. \qquad (24a)$$

4. Absence of simultaneous switching off:

$$\bar{x}_1 \bar{x}_2 = 0, \qquad (25)$$

$$x_1 x_2 = 1, \qquad (25a)$$

$$\bar{x}_1 x_2 = \bar{x} \qquad (26)$$

$$x_1 + \bar{x}_2 = x, \qquad (26a)$$

$$x_1 \bar{x}_2 = \bar{x}_2, \qquad (27)$$

$$\bar{x}_1 + x_2 = x_2. \qquad (27a)$$

It will be seen from the table in Figure 14 that in the instance under consideration, the element X_1 is always switched on earlier than, and switched off later than the element X_3, and that the element A is always switched on earlier than the element X_3. Hence, in accordance with (16a) and (17), in the structural formula we have:

$$x_1 + \bar{x}_3 = 1; \quad x_1 x_3 = x_3, \text{ and } ax_3 = x_3,$$

and we obtain the final structural formula of the circuits acting on the element X_1 in the following form:

$$F_{(X_1)} = a[\bar{x}_4(x_1 + \bar{x}_3) + x_1 x_3 x_4] = a(\bar{x}_4 + x_3 x_4) = a(\bar{x}_4 + x_3) =$$

$$= a\bar{x}_4 + ax_3 = a\bar{x}_4 + x_3.$$

For the element X_2, as may be seen for the switching table in Figure 14, only the element X_1 changes its position in cycles f_{res} and f_{rel}. In this case, the switching table is always feasible when there are present in it only the element in question and the element through which the former responds and releases (Figure 17). Writing out from the switching table in Figure 17 the condition of the elements in cycles 2, 3, 4, 5, and 6, we find:

$$F_{(X_2)} = x_1 \bar{x}_2 + x_1 x_2 = x_1.$$

CYCLES		0	1	2	3	4	5	6	7	8	9	10
BINARY	2^1	$-X_1$		$+X_1$					$-X_1$			
NUMBERS	2^2	$-X_2$			$+X_2$					$-X_2$		
NUMBER OF COMBINATIONS		0	0	2	6	6	6	6	4	0	0	0

Figure 17. Example of the feasibility of the table of switching of the element X_2.

CYCLES		0	1	2	3	4	5	6	7	8	9	10
	2^2	$-X_2$			$+X_2$					$-X_2$		
BINARY NUMBERS	2^3	$-X_3$				$+X_3$		$-X_3$				
	2^4	$-X_4$					$+X_4$				$-X_4$	
NUMBER OF COMBINATIONS		0	0	0	4	12	28	20	20	16	0	0

Figure 18. Example of the feasibility of the table of switching of the element X_3.

For the element X_3, as follows from the table of switchings in Figure 18, feasible conditions are obtained if a table is written out for the elements which change their position in the cycles f_{res} and f_{rel}. Writing out the condition of the elements from this table in cycles 3 and 4, we find

$$F(X_3) = x_2\bar{x}_3\bar{x}_4 + x_2x_3\bar{x}_4 = x_2\bar{x}_4.$$

The same table of switchings (Figure 18) may be utilized for the element X_4. Writing out the condition of the elements in cycles 4, 5, 6, and 7, we find

$$F(X_4) = x_2x_3\bar{x}_4 + x_2x_3x_4 + x_2\bar{x}_3x_4 = x_2x_3 + x_2\bar{x}_3x_4.$$

It will be seen from the table in Figure 14 that the element X_2 is always switched on earlier than the element X_3 and switched off later than the latter.

Hence $x_2x_3 = x_3$, and, consequently

$$F(X_4) = x_2x_3 + x_2\bar{x}_3x_4 = x_3 + x_2\bar{x}_3x_4 = x_3 + x_2x_4.$$

The common structural formula of the connection is:

$$F(a\bar{x}_4 + x_3)X_1 + x_1X_2 + x_2\bar{x}_4X_3 + (x_3 + x_2x_4)X_4.$$

A graphic representation of the connection is given in Figure 19a. Opening the parentheses and grouping the members somewhat differently, we have

$$F = \bar{x}_4(aX_1 + x_2X_3) + x_3(X_1 + X_4) + x_4x_2X_4 + x_1X_2.$$

The corresponding connection is given in Figure 19b; it is similar to the connection shown in Figure 11.

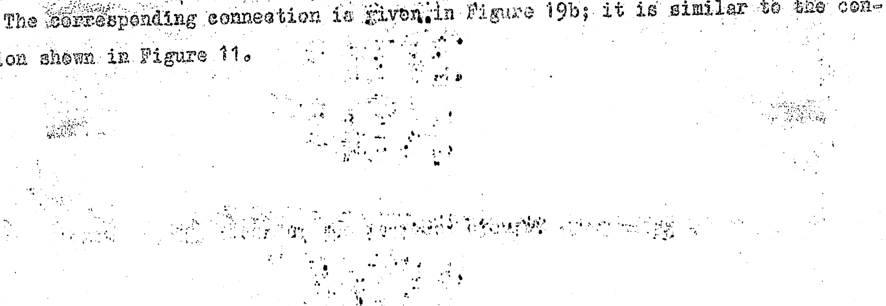

Figure 19. Variants of the connection of a multicycle pulsing device, shown in Figure 11:
1 - b.

The method discussed of transition from a switching table to structural formulas of the circuits acting on the individual elements of a connection requires the writing out of the condition of the elements in all the cycles in which the circuit of a given element is closed. In a number of cases this is a rather unwieldy process. The theory of relay connections has several formulas for transition from switching tables to initial structural formulas, and several of these furnish a simpler solution to the problem.

First of all it must be pointed out that when the condition of the elements in the cycles in which the circuit of a given element must be closed (in the method discussed above) is written out, there is no need to record in the cycle f_{res} the condition of the contacts of the element for which the structural formula is being recorded. This simplifies the conversion somewhat. The formula for transition from the switching table in the structural formula of the circuits acting on any element X in this case has the following form:

$$F_{(X_1)} = f^{o}_{res} + xF^{o}_{3}, \qquad (28)$$

where f^{o}_{res} is the condition of the elements in the cycle f_{res} without allowance for the contacts of the element X;

F^{o}_{3} is the condition of the elements in the cycles in which the circuit of the element X must be closed, likewise without allowance for the contacts of the element X, and, in addition, excluding the cycle f_{res}.

For transition from the switching tables to initial structural formulas, it is very convenient to make use of the expression

$$F_{(X)} = f^{o}_{res} + xf^{o}_{rel}, \qquad (29)$$

where f^{o}_{res} is the same as indicated above;

f^{o}_{rel} is the inversion of the condition of the elements in the cycle f_{rel}, again without allowance for the element's own contacts.

Formula (29) is convenient for the reason that when it is utilized for arriving at the structural formula of the circuits acting on a given element, it is sufficient to write out the condition of the elements in the two cycles f_{res} and f_{rel} only, instead of in all the cycles in which the circuit of a given element must be closed. In the event any element has several cycles of response and release according to the switching table, when f^{o}_{res} and f^{o}_{rel} are written out in formulas (28) and (29), it is necessary to include in them the total of the conditions of the elements in cycles f_{res} and f_{rel} for all the cycles of switching on and off of the element X.

Let us examine the synthesis of a relay connection in the light of one of the concrete problems of systems of electric control of machine lines[1] in the loading connection of a relay mechanism for pistons.

This mechanism (Figure 20) operates as follows. Pistons (two each) move from the interassembly slide onto the tray of the relayer. When the first two pistons are delivered, a path switch is depressed which, acting on the connection of the automatic system, switches on an electromagnetic drive E_1, which advances the cylinders horizontally (along arrow A). At the end of movement of the cylinders, the path switch P switches off the electromagnet E_1. When the next pair of pistons move from the slide onto the tray, the path switch P is again depressed which, acting again on the

[1] This example was worked out by Engineer I. V. Ivanov.

connection of the automatic system, switches on the electromagnetic drive E_2, which displaces the tray vertically (along arrow B). Then the pistons, in groups of four, are delivered to the next assembly, the path switch is switched off, and the automatic system connection and tray return to the original position.

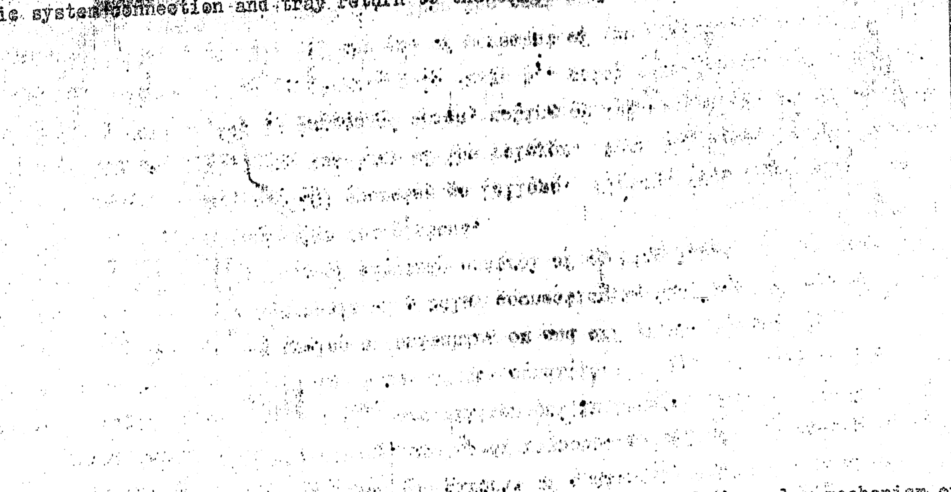

Figure 20. Diagrammatic representation of the loading of the relay mechanism of machine lines for machining pistons: 1 - interassembly slide; 2 - cylinder valve "up"; 3 - cylinder valve "forward"; 4 - B; 5 - E_2 (electromagnet); 6 - E_1 (electromagnet).

According to the conditions of operation, the connection of the automatic system must have one pickup element (the path switch P) and two performing electromagnetic drives E_1 and E_2. We introduce two intermediate elements, X_1 and X_2, for the control of the latter. In accordance with the conditions of operation of the connection, we draw up a table of sequence of switching of the elements (Figure 21) and verify its feasibility. As will be seen from the table, there are no recurring numbers throughout the cycles of operation of the connection; hence the connection is feasible.

Let us determine the switching circuits of the individual elements of the connection:

$$F_{(X_1)},\ F_{(X_2)},\ F_{(E_1)},\ \text{and}\ F_{(E_2)}.$$

First let us find feasible conditions for each element. Element X_1 is switched on in cycle 2 and off in cycle 8, in both instances after the element P has been switched on. Since these conditions are contradictory for the operation of X_1, we introduce into $F_{(X_1)}$ the contacts of the intermediate element X_2. Now we verify the feasibility of the table for the element X_1 (Figure 22). The table is not feasible, since in cycles 0 and 4 identical values of the combination numbers (0) correspond to the opposite condition of the switching circuits of the element X_1. In cycle 0, the circuit $F_{(X_1)}$ must be open and closed in cycle 4. We introduce the contacts of the element X_1 into the connection $F_{(X_1)}$ and obtain a feasible table (see the number of combinations in the bottom row of the table in Figure 22). The table will be feasible for element X_2 if the same elements are used as for X_1.

In order to render the table of switchings feasible for the elements E_1 and E_2, it is sufficient to use the elements X_1 and X_2, which cause the switching on and off of both electromagnets.

CYCLES		0	1	2	3	4	5	6	7	8	9	10	11	12
BINARY NUMBERS	2^0	−P	+P			−P			+P			−P		
	2^1	$-X_1$		$+X_1$						$-X_1$				
	2^2	$-X_2$					$+X_2$						$-X_2$	
	2^3	$-E_1$			$+E_1$			$-E_1$						
	2^4	$-E_2$									$+E_2$			$-E_2$
NUMBERS OF COMBINATIONS		0	1	2	11	10	14	6	7	5	21	20	16	0

Figure 21. Table of the sequence of switching of the elements of the connection shown in Figure 20.

CYCLES		0	1	2	3	4	5	6	7	8	9	10	11	12
BINARY NUMBERS	2^0	−P	+P			−P			+P			−P		
	2^1	$-X_1$		$+X_1$						$-X_1$				
	2^2	$-X_2$					$+X_2$						$-X_2$	
NUMBERS OF COMBINATIONS		0	1	1	1	0	4	4	5	5	5	4	0	0
		0	1	3	3	2	6	6	7	5	5	4	0	0

Figure 22. Determination of the conditions of feasibility of the table shown in Figure 21 for the elements X_1 and X_2.

The table given in Figure 23 shows the conditions of feasibility of the switching circuits $F_{(E_1)}$ and $F_{(E_2)}$. We determine the circuits acting on the elements X_1 and X_2 by formula (29):

$$F_{(X_1)} = \bar{p}x_2 + x_1(\overline{\bar{p}\bar{x}_2}) = p\bar{x}_2 + x_1(\bar{p} + \bar{x}_2);$$

$$F_{(X_2)} = \bar{p}x_1 + x_2(\overline{\bar{p}\bar{x}_1}) = \bar{p}x_1 + x_2(p + x_1).$$

CYCLES		0	1	2	3	4	5	6	7	8	9	10	11	12
BINARY NUMBERS	2^1	$-X_1$		$+X_1$						$-X_1$				
	2^2	$-X_2$					$+X_2$						$-X_2$	
	2^3	$-E_1$			$+E_1$			$-E_1$						
	2^4	$-E_2$									$+E_2$			$-E_1$
NUMBERS OF COMBINATIONS		0	0	2	2	2	6	6	6	4	4	4	0	0

Figure 23. Table characterizing the conditions of feasibility of the switching circuits of the elements E_1 and E_2.

In order to simplify the expressions obtained for $F_{(X_1)}$ and $F_{(X_2)}$, we isolate in them the switching contact of the element P, representing them in the form:

$$F_{(X)} = (p + F_{(X)})(\bar{p} + F_{(X)}).$$

It is easy to demonstrate that it does not have the original expression for the circuits $F_{(X_1)}$ and $F_{(X_2)}$. Indeed, opening the parentheses, we find:

$$F_{(X)} = pF_{(X)} + \bar{p}F_{(X)} = (p + \bar{p})F_{(X)} = F_{(X)};$$

$$F_{(X_1)} = [p + p\bar{x}_2 + x_1(\bar{p} + \bar{x}_2)][\bar{p} + p\bar{x}_2 + x_1(\bar{p} + \bar{x}_2)] = (p + x_1)(\bar{p} + \bar{x}_2);$$

$$F_{(X_2)} = [p + \bar{p}x_1 + x_2(p + x_1)][\bar{p} + \bar{p}x_1 + x_2(p + x_1)] = (p + x_1)(\bar{p} + \bar{x}_2).$$

We determine the circuits acting on the electromagnets E_1 and E_2 by the cycles of their switched on condition (Figure 22):

$$F_{(E_1)} = x_1\bar{x}_2 + x_1\bar{x}_2 = x_1\bar{x}_2; \qquad F_{(E_2)} = \bar{x}_1x_2 + \bar{x}_1x_2 = \bar{x}_1x_2.$$

The general analytic expression of the entire connection has this form:

$$F = F_{(X_1)} \cdot X_1 + F_{(X_2)} \cdot X_2 + F_{(E_1)} \cdot E_1 + F_{(E_2)} \cdot E_2 = (p + x_1)(\bar{p} + \bar{x}_2)X_1 +$$
$$+ (p + x_1)(\bar{p} + \bar{x}_2)X_2 + x_1\bar{x}_2E_1 + \bar{x}_1x_2E_2 = (p + x_1)[(\bar{p} + \bar{x}_2)X_1 +$$
$$+ (\bar{p} + x_2)X_2] + x_1\bar{x}_2E_1 + \bar{x}_1x_2E_2.$$

If x_2 and $\bar{x}_2$ respectively are added to the element p in the parentheses with X_1 and X_2, and doing so does not change the action of the connection, it will be seen that the switching circuits of the elements X_1 and X_2 represent an elementary bridge connection.

In this case, the analytic expression of the connection assumes the following form:

$$F = (p + x_1)[(\bar{p}x_2 + \bar{x}_2)X_1 + (\bar{p}\bar{x}_2 + x_2)X_2] + x_1\bar{x}_2E_1 + \bar{x}_1x_2E_2.$$

The element P is the bridge element in the connection.

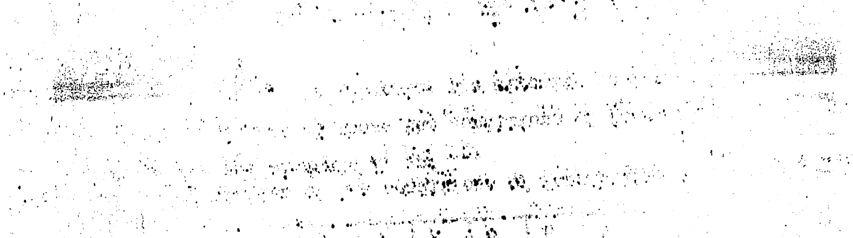

Figure 24. Switching connection of the elements of the group of mechanisms shown in Figure 20:
1 - p; 2 - $\bar{p}$; 3 - E_1; 4 - E_2.

A graphic representation of the analytic expression of the connection obtained, which satisfies all the operating conditions assigned for it, is given in Figure 24.

B. S. SOTSKOV

EMPLOYMENT OF THE ELEMENTS OF SWITCHLESS CONTROL IN THE AUTOMATIC ELECTRIC DRIVE OF MACHINES AND MACHINE LINES

Of the various types of switchless emitters, inductive emitters possess the greatest simplicity and reliability, an adequate frequency of change of current in the control circuit, and great controlled power. These positive features have insured, even at the present time, relatively extensive employment of inductive emitters in the connections of automatic devices for various purposes.

Figure 1. Constructional diagrams of inductive emitters: 1 - b; 2 - c; 3 - d; 4 - e.

An account is given in this article of the principal considerations of the characteristics and methods of calculation of an emitter, with the aim of facilitating selection of the constructional forms and calculation of the parameters of emitters in the construction of switchless systems of control of machine lines.

Figure 1 gives the constructional diagrams of various types of inductive emitters, of which 1a and 1b have been put to the widest use. Only these types of emitters will be dealt with subsequently in the article, although the methods of calculation proposed may be applied to the other groups also.

In comparing emitters with variable inductance ($L = var$) with those with variable mutual inductance ($M = var$), it should be noted that the former are more economical insofar as expenditure of copper and steel is concerned. However, in inductive systems with $M = var$ it is possible to achieve greater frequency of change of current in the control circuit. This is due to the fact that in emitters with $L = var$, the greatest value of $L = L_{max}$ is conditioned by the value of the sum of the permeances for the main flux Φ_r and leakage flux Φ_u, and the smallest [redacted] ned chiefly by the conductance for the leakage flux Φ_u, since the conductance for the main flux Φ_u is here very small.

STAT

In the case of an emitter with M = var, the greatest and smallest values of the electromotive force in the secondary (controlled) circuit are determined by the maximum and minimum conductances of the operating flux Φ_r.

CHARACTERISTICS OF AN INDUCTIVE EMITTER

The principal characteristics of an inductive emitter are the following:

(a) the working characteristic $z_e = f(x)$, which gives the resistance of the emitter and the displacement of its armature;

(b) the voltampere characteristic $u = f_2(I)$, which gives the connection between the voltage drop on the emitter and the current in it;

(c) the traction characteristic $P = f_2(\delta, I)$, which gives the dependency of the tractive forces in the emitter gap with the function of displacing the armature and the current intensity.

The working characteristic of an emitter depends on the whole on the laws of change of the permeance of its air gap. Indeed, the full resistance of an emitter z_d can be represented as

$$z_d = R_d + j\omega L_d,$$

where R_d is the resistance of the emitter winding and L_d is the inductance of the emitter.

The resistance of the emitter winding is determined by the formula

$$R_d = \frac{\rho \cdot l_{av} \cdot w}{\frac{\pi d^2}{4}} = \frac{\rho \cdot l_{av} w^2}{Q \cdot f_0} = k_1 w^2,$$

where ρ is the resistivity;

l_{av} is the average length of the turn (m);

Q is the area of the cross section of the winding window (mm^2);

f_0 is the filling coefficient;

w is the number of turns of the winding.

The inductance of an emitter is determined by the following formula:

$$L_d = \frac{0.4\pi \cdot w^2 \cdot 10^{-8}}{R_m} = \frac{0.4\pi \cdot 10^{-8} \cdot w^2}{R_{ms} + R_{mv}} = \frac{0.4\pi \cdot 10^{-8} \cdot w^2}{1 + R_{ms} G_{mv}} \cdot G_{mv} = k_2 w^2,$$

where $R_{mv} = \frac{1}{G_{mv}}$ is the reluctance of the air gap;

G_{mv} is the permeance of the air gap;

R_{ms} is the reluctance of the magnetic circuit.

Usually $G_{mv} \cdot R_{ms} \ll 1$, hence $1 + R_{ms} \cdot G_{mv} \approx 1$. Thus we have:

$$z_d = (k_1 + jk_2\omega)w^2$$

or

$$z_d = k_2 \omega w^2 \sqrt{\left(\frac{k_1}{\omega k_2}\right)^2 + 1} = \omega k_2 w^2 \cong \omega \cdot 0.4\pi \cdot 10^{-8} G_{mv} w^2.$$

The values of ω and w for an emitter are constant magnitudes and when the emitter is in operation, the value of the permeance G_{mv} changes, i.e., $G_{mv} = \varphi(x)$, STAT con- nection with which

$$z_d = 0.4\pi \cdot 10^{-8} w^2 \omega\varphi(x) = f(x).$$

Figure 2. Graphs of the function:

$a = z_d = f(x)$; $b = z_d + z_r = f(x)$;

$1 - Z'_d$; $2 - 2\Delta Z_d$; $3 - Z_{d_2}$; $4 - Z_{d_1}$; $5 - 2\Delta Z_d$; $6 - \frac{V}{I_{rel}} = Z''_r + Z_{d_2}$; $7 - \frac{V}{I_{res}} = Z'_r + Z_{d_1}$; $8 - Z_d$; $9 - b$; $10 - Z_v$; $11 - Z''_r$; $12 - Z'_r$.

The function $z_d = f(x)$ is shown in Figure 2a. For an emitter with varying gap, for which

$$g_{mv} = \frac{S_v}{\delta} = \frac{(a + k\delta)^2}{\delta} \approx \frac{a^2}{\delta} + 2ak.$$

It is necessary to take into account the permeance for leakage fluxes, which equals

$$G_{mu} = \frac{1}{2} g l_s,$$

where l_s is the length of the core of the emitter;

g is resistivity for leakage fluxes.

Then we obtain

$$G_{ms} = G_{mv} + G_{mu} = \frac{a^2}{\delta} + (2ak + \frac{1}{2} g l_s) = \frac{a^2}{\delta} + G_0.$$

If the necessary limits of change of current in the emitter have been assigned (lower, I_1, and upper, I_2), then the required change of resistance of the emitter is determined by the formulas

$$z_{d_1} = \frac{i + \Delta i}{I_1} - z_0,$$

$$z_{d_2} = \frac{i - \Delta i}{I_2} - z_0,$$

where i is the voltage in the emitter circuit;

Δ_i are the possible fluctuations of the voltage;

z_0 is the resistance switched into the emitter circuit.

With z_{d_1} and z_{d_2} known, it is possible to compute the corresponding values of δ_1 and δ_2 and their variations upon change of voltage. Figure 2a shows the computation of δ_1 and δ_2 by the characteristic $z_d = f(x)$.

In a number of cases, for example, when the emitter is connected in series with the relay, the resistance of the latter is variable, changing from the value of z_r' with the armature released to the value z_r'' with the armature attracted. If the response and release currents of the relay are designated by I_{res} and I_{rel} respectively, the values of the resistance of the emitter necessary for the response and release of the relay are determined as follows:

$$z_{d_1} = \frac{i + \Delta i}{I_{res}} - z_r';$$

and

$$z_{d_2} = \frac{i + \Delta i}{I_{rel}} - z_r''.$$

With the values of $z_d = z_{d_1}$ and $z_d = z_{d_2}$, it is easy to compute by the function $z_d = f(x)$ the magnitudes of δ_1 and δ_2 corresponding to them.

Figure 2b shows the functions $z_d + z_r = f(x)$ for the two values $z_r = z_r'$ and z_r''. By plotting the values of $\frac{i}{I_{res}}$ and $\frac{i}{I_{rel}}$, it is possible to find the magnitudes of the gaps δ_1 and δ_2, which correspond to the instants of response and release of the relay. If the armature of the relay is attracted (upper curve), with subsequent reduction of δ (from point b), $z_d + z_r$ increases, until at point c, which corresponds to $\delta = \delta_2$, it reaches the value of $z_d = z_r''$, which determines the magnitude $I = \frac{i}{z_{d_2} + z_r''} = I_{rel}$. The relay releases its armature and the value of its resistance falls to $z_r = z_r'$ (i.e., the operating point shifts from point c to point d). Further reduction of δ does not produce any changes in the position of the relay. With subsequent increase of δ (in accordance with the lower curve), $z_d + z_r$ decreases until at point a, which corresponds to $\delta = \delta_1$, it reaches the value $I = \frac{i}{z_r' + z_d} = I_{res}$, with which the relay attracts its armature and by a jump changes its position from the value $z_r = z_r'$ to the value $z_r = z_r''$ (the operating point shifts from point a to point b of the upper curve). Further increase corresponds to displacement of the operating point to the right of point b.

Adherence to the following conditions is necessary for proper operation of an inductive emitter with a relay, for the avoidance of pulsing operation of the latter:

$$\frac{i}{I_{res}} < [z_b = z_r'' + z_{d(\delta = \delta_1)}] < \frac{i}{I_{rel}}$$

and

$$\frac{i}{I_{res}} < [z_d = z_r' + z_{d(\delta = \delta_2)}] < \frac{i}{I_{rel}}.$$

The voltampere characteristic of an emitter is determined by the expression

$$i = z_d \cdot I \approx \omega \cdot 0.4\,\pi \cdot 10^{-8} \cdot w^2 \cdot G_{mv} I,$$

i.e., for a given G_v (or δ), is is also proportional to the magnitude of the current. If G_v varies between G_{v1} and G_{v2}, the limiting voltampere characteristics may be defined as follows:

$$i_1 = \omega \cdot 0.4\,\pi \cdot 10^{-8} \cdot w^2 \cdot G_{mv1} I,$$

$$i_2 = \omega \cdot 0.4\,\pi \cdot 10^{-8} \cdot w^2 \cdot G_{mv2} I.$$

Figure 3 shows the limiting voltampere characteristics and the voltampere characteristic of an inductive load, expressed in approximation by the formula

$$i = z_0 I \approx \omega \cdot 0.4\,\pi \cdot 10^{-8} \cdot w_0^2 \cdot G_0 \cdot I$$

Figure 3. Graphs:

a - of a limiting voltampere characteristic; b - of the voltampere characteristic of an inductive load; 1 - G_{mv2}; 2 - G_{mv1}; 3 - b.

(i.e., R_0 is assumed to equal zero). The points of intersection of the limiting voltampere characteristics of an emitter with the characteristic

$$E - i = E - \omega \cdot 0.4\pi \cdot w^{-8} \cdot w_0^2 \cdot G_0 \cdot I$$

determine the limits of change of current in the circuit of the load of I_1 and I_2.

In the event of connection in series with an emitter of a relay the voltampere characteristics of which in the attracted and released position of the armature may be expressed as

$$i_r' = z_r' I \approx \omega \cdot 0.4\pi \cdot 10^{-8} \cdot w_r^2 \cdot G_r' I$$

and

$$i_r'' = z_r'' I \approx \omega \cdot 0.4\pi \cdot 10^{-8} \cdot w_r^2 \cdot G_r'' I,$$

the limiting values of the current I_1 and I_2 may be found from the conditions that

$$i' = E - i_r' \text{ and } i'' = E - i_r''$$

or from Figure 3b. For reliable operation of a relay with an emitter, the response current I_{res} and the release current I_{rel} of the relay must lie within the limits

$$I_1 < I_{res} < I_1$$

and

$$I_2 < I_{rel} < I_3.$$

The traction characteristic of an emitter may be determined from the condition

$$P_e = \frac{(0.4\,\pi \cdot IW_v)^2}{8\,\pi} \cdot \frac{dG_v}{d\delta} \cdot \frac{1}{981}$$

or

$$P_e = \frac{(0.4\pi w)^2}{98 \cdot 18\pi} \cdot \frac{dG_v}{d\delta} \cdot \left[\frac{1}{\omega L_0 \omega + 0.4\pi \cdot 10^{-8} w^2 G_{mv}}\right]^2.$$

Figure 4. Graphs of the function

$$P_e = \varphi(\delta);$$

a - when the inductive load is invariable; b - when a relay is connected; 1 - P_e; 2 - b.

Figure 4 shows the function $P_d = \varphi(\delta)$ for the event of invariable inductive load (a) and for the event of connection of a relay (b), in which at the moments of response (when $\delta = \delta_1$) and release (when $\delta = \delta_2$) of the relay, an abrupt change of the current occurs, and, consequently, abrupt changes of the tractive forces must also take place. Curve 1 (Figure 4b) corresponds to the released position of the armature of the relay and curve 2 to the attracted position.

CALCULATION OF THE MAGNETIC SYSTEM OF AN EMITTER

We shall examine two methods of calculation of the magnetic system of an emitter which make it possible to determine with the accuracy necessary for engineering calculations the dependency of the current in the emitter circuit and of the load on displacement of the armature.

A. Calculation of the magnetic system by means of the family of curves $B_m = f(I_m \delta_x)$. The following equations are valid for calculation of the magnetic system.

(a). For magnetomotive forces:

$$(Iw)_m = (Iw)_{mv} + (Iw)_{ms}$$

or, multiplying by $\frac{0.4\pi}{l_s}$:

$$f_m = H_{mv} + H_{ms},$$

where f_m is the magnetomotive force per unit of length;

H_{mv} is the portion of the magnetomotive force from 1 cm of the length of the active part of the magnetic circuit required for carrying the magnetic flux through the air gap;

H_{ms} is the portion of the magnetomotive force lost in carrying the flux through the magnetic circuit;

(b) For magnetic induction in the magnetic circuit: $B_m = f(H_{ms})$;

(c) For magnetic induction in the air gap:

$$\Phi = H_{mv} l_s G_{mx}$$

or

$$B = G_{mv} \frac{l_s}{S_c} H_{mv}.$$

Figure 5. Graphs for calculation of the magnetic system of an emitter.

Drawing a straight line from the point $H_m = f_m$ at an angle $\theta = \operatorname{arc\,tg} G_{mv} \frac{l_s}{S_s}$, we find point B_m and the values of H_{mv} and H_{ms} (Figure 5a). Bringing point 1 to the line $f_m = \text{const}$, we obtain the point 1'; repeating the construction for the various values of f_m, we obtain the curve $B_m' = F(H_m)$. Setting ourselves a number of values $\delta_x = \delta_1$; $\delta_x = \delta_2$; $\delta_x = \delta_3$ ...., we find the corresponding values of G_{mvx} and $tg\theta_x$. It is a simple matter to construct a curve $B_m' = F(H_m)$ for each value of $tg\theta_x$ (i.e., each δ_x), as shown in Figure 5a.

For the electrical part we have:

$$i_m = I_m(R_r + R_d) + j(\omega w S B_m \cdot 10^{-8} + \omega \cdot L_r I_m),$$

whence it follows that

$$i_m^2 = I_m^2(R_r + R_d)^2 + (\omega w \cdot S B_m 10^{-8} + \omega \cdot L_r I_m)^2.$$

Dividing the right and left members of the equation by i_m^2, we find

$$1 = \frac{I_m^2}{I_{mm}^2} + \frac{\left(B_m + \frac{L_r}{w \cdot S 10^{-8}} \cdot I_m\right)^2}{B_{mm}^2}$$

or

$$1 = \frac{I_m^2}{I_{mm}^2} + \frac{B_{mx}^2}{B_{mm}^2},$$

where

$$I_{mm} = \frac{i_m}{R_r + R_d}; \quad B_{mm} = \frac{i_m}{\omega \cdot w \cdot S \cdot 10^{-8}}; \quad B_{mx} = \frac{L_r I_m}{wS \cdot 10^{-8}}.$$

In addition, the function

$$H_m = \frac{0.4\pi \cdot W}{l_s} \cdot I_m = s \cdot I_m.$$

is valid.

Let us construct the function $B_m = f_1(I_m)$ for various values of δ_x. For this purpose it is necessary to change by s times the scale along the abscissa in the graph of the family of curves shown in Figure 5a. We plot on the graph of the family $B_m = f_1(I_m)$ with various values of δ_x the curve corresponding to the equation

$$1 = \frac{I_m^2}{I_{mm}^2} + \frac{B_{mx}^2}{B_{mm}^2},$$

which the equation of an ellipse with semiaxes I_{mm} and B_{mm}.

We plot on this same graph the function

$$\frac{L_r' I_m}{S \cdot w \cdot 10^{-8}} = tg\alpha_1 I_m,$$

for which purpose we draw from the origin of the coordinates a straight line at the angle $\alpha_1 = \text{arc tg} \frac{L_r'}{S \cdot w \cdot 10^{-8}}$. We subtract from the ordinates of the ellipse the values of the ordinates of the straight line $\frac{L_r' I_m}{S \cdot w \cdot 10^{-8}}$, and obtain the curve $B_{mm} 1' 2' 3' \ldots$, which is indicated by a broken line in Figure 5b. The points of intersection of the curve thus obtained with the curves $B_m = f_2 X (I_m, \delta_x)$ determine the amount of current in the circuit of the relay and emitter windings.

In order to arrive at the relationship of the current in the circuit of the relay and emitter windings to the size of the operating gap δ_x, we plot below the origin of the coordinates the values of δ_x corresponding to the curves of the graph $B_m = f(I_m, \delta_x)$ and bring the structures of the values of the currents obtained as a result down to the lines corresponding to the values of δ_x, as shown in the lower portion of Figure 5b. Then we obtain the curve $I_m = \varphi_1(\delta_x)$ (curve a). By similar plotting it is possible to find a second curve $I_m = \varphi_2(\delta_x)$ for the event of an attracted relay armature, for which purpose it is necessary only to plot a straight line at an angle

$$\alpha_2 = \text{arc tg} \frac{L_x''}{w \cdot S \cdot 10^{-8}}$$

instead of the former line corresponding to the angle α_1. In this case we obtain the curve $I_m = \varphi_2(\delta_x)$ (curve b).

It should be noted that the power losses in the steel of the magnetic circuit are not taken into account in the plotting discussed above, and for this reason in calculations we use the vector diagram shown in Figure 5d instead of the vector diagram in Figure 5c. If the magnetic induction in the magnetic circuit of the emitter does not exceed 8000 to 10,000 gausses, the error in the amount of current resulting from disregarding of the power losses in the steel of the magnetic circuit does not exceed 10 to 15%, and the method indicated of calculation of an inductive emitter may be successfully employed.

Given below is another method, which makes it possible to take into account the losses in the steel of the magnetic system.

B. Calculation of the magnetic system by means of the curves $\mu = \mu_L - j\mu_R$. For the magnetic materials employed in alternating current magnetic systems, use is made at the present time of expressions of magnetic permeability in the form

$$\mu = \mu_L - j\mu_R ,$$

where μ_L is the component of the magnetic permeability which determines the magnitude of the magnetic flux;

μ_R is the component of the magnetic permeability which determines the losses in eddy currents and hysteresis.

In Figure 6 are shown graphs for the function

$$\mu = \mu_L - j\mu_R \quad \left(\text{more accurately, } \frac{\mu}{\mu_n} = \frac{\mu_L}{\mu_n} - j\frac{\mu_R}{\mu_n}\right)$$

with various amplitudes of intensity of the magnetic field H_m and various frequencies f of the supply current.

Figure 6. Graph of the function $\mu = \mu_L - j\mu_R$:
1 - 50 cycles per second; 2 - 1 cycle per second.

The vector diagram in Figure 7a is valid for the conditions of connection of an emitter (Figure 7); the diagram corresponds to the equation

$$i_m = I_m(R_d + j\omega L_{S\,d}) + (e_{m_2} + je_{m_1}) + I_m(R_0 + j\omega L_0),$$

where R_d is the ohmic resistance of the winding;

$L_{S\,d}$ is the dispersion inductance of the emitter, which is determined by the leakage fluxes; $e_m = e_{m_2} + je_{m_1}$ is the electromotive force, which is induced by the main magnetic flux in the winding of the emitter;

R_0 is the ohmic load resistance;

L_0 is the load inductance.

In addition, we have

$$e_m = j\omega \cdot w \cdot S \cdot B_m$$

and

$$e_{m_2} = e_m \cdot \sin \varepsilon ; \; e_{m_1} = e_m \cdot \cos\varepsilon ,$$

where ω is the angular frequency;

w is the number of turns;

S is the winding section;

B_m is the amplitude of the magnetic inductance.

We find the connection between the angle ε and the components of the magnetic permeability:

$$\mu = \mu_L - j\mu_R .$$

Hence,

$$B_m = H_m \mu = H_m \mu_L - jH_m \mu_R .$$

Occasionally it is more convenient to use the inverse function:

$$H_m = H_{m_1} + jH_{m_2} = B_m (\alpha_L + j\alpha_R),$$

or

$$B_m = \frac{H_{m_1}}{\alpha_L} \text{ and } B_m = \frac{H_{m_2}}{\alpha_R} .$$

Figure 7. Diagrams of the conditions of connection of an emitter: 1 - $y_2 = U$ = constant; 2 - $y_1 = f_1(H_{m_1})$ when $G_b = G_{b_2}$.

The connection between (α_L), (α_R) and (μ_R), (μ_L) is determined from the conditions

$$(\mu_L - j\mu_R)(\alpha_L + j\alpha_R) = 1$$

and

$$\frac{\mu_R}{\mu_L} = \frac{\alpha_R}{\alpha_L} ;$$

hence,

$$\alpha_L = \frac{\mu_L}{\mu_L^2 + \mu_R^2} \text{ and } \alpha_R = \frac{\mu_R}{\mu_L^2 + \mu_R^2} .$$

It follows from Figure 7c that

$$\cos\varepsilon = \frac{H_{m_1}}{H_m} = \frac{\mu_L}{\sqrt{\mu_L^2 + \mu_R^2}} = \frac{\alpha_L}{\sqrt{\alpha_L^2 + \alpha_R^2}} ;$$

$$\sin\delta = \frac{H_{m_2}}{H_m} = \frac{\mu_R}{\sqrt{\mu_R^2 + \mu_L^2}} = \frac{\alpha_R}{\sqrt{\mu_R^2 + \mu_L^2}} .$$

We proceed to calculation of the magnetic system of the emitter. For the fluxes in the magnetic system we have

$$\Phi = \Phi_v + \Phi_y ,$$

where Φ_v is the flux in the air gap, equalling $\Phi_v = G_v \cdot H_v \cdot l_m$;

Φ_y is the leakage flux, equalling $\Phi_y = \frac{1}{2} g \cdot l_m \cdot H_v$;

H_v is the intensity of the magnetic field required for carrying the flux through the air;

G_v is the total conductance of the operating air gaps;

l_m is the length of the active portion of the magnetic system;

g is the resistivity of the leakage fluxes.

Since

$$\Phi = S_m \cdot B_m = S_m \frac{H_{m_1}}{\alpha_L},$$

it follows that

$$\frac{H_v}{H_{m_1}} = \frac{S_m \cdot \frac{1}{\alpha_L}}{(G_v + \frac{1}{2} g l_m) l_m} . \qquad (1)$$

The magnitude of the specific magnetomotive force is

$$f = \frac{0.4\pi \cdot w}{l_m} \cdot I_m . \qquad (2)$$

On the other hand,

$$f^2 = (H_{m_1} + H_v)^2 + H_{m_2}^2 = H_{m_1}^2 \left[\left(1 + \frac{H_v}{H_{m_1}}\right)^2 + \left(\frac{H_{m_2}}{H_{m_1}}\right)^2 \right].$$

Using formula (1), we have

$$f = H_{m_1} \left[\left(1 + \frac{S_m}{(G_v + \frac{1}{2} g l_m) \dfrac{l_m \cdot \mu_L}{\sqrt{\mu_L^2 + \mu_R^2}}} \right)^2 + \left(\frac{\mu_R}{\mu_L}\right)^2 \right]^{\frac{1}{2}} . \qquad (3)$$

The values of e_1 and e_2 may be expressed thus:

$$\left.\begin{aligned} e_1 &= e_m \cdot \cos\delta = \omega w S_m \cdot \frac{H_{m_1}}{\alpha_L} \cdot \frac{H_{m_1} + H_v}{f} = \\ &= \omega w S_m \cdot \frac{\dfrac{H_{m_1}^2}{\mu_L}}{\sqrt{\mu_L^2 + \mu_R^2}} \left[\frac{1 + \dfrac{S_m \sqrt{\mu_L^2 + \mu_R^2}}{(G_v + \frac{1}{2} g l_m) l_m \mu_L}}{f} \right] \\ e_2 &= e_m \sin\delta = \omega w S_m \frac{H_{m_1}}{\alpha_L} \cdot \frac{H_{m_2}}{f} = \\ &= \omega \cdot w \cdot S_m \cdot \frac{H_{m_1}^2 \cdot \sqrt{\mu_L^2 + \mu_R^2}}{\mu_L} \cdot \frac{\dfrac{\mu_R}{\mu_L}}{f} . \end{aligned}\right\} \qquad (4)$$

It follows from the main equation of the emitter circuit that

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$$I_m = \left\{\left[e_2 + \frac{f l_m}{0.4\pi w}(R_0 + R_d)\right]^2 + \left[e_1 + \frac{f \cdot l_m}{0.4\pi w}\cdot(L_{sd} + L_0)^2\omega\right]\right\}^{\frac{1}{2}}.$$

The calculation is performed as follows. H_m is worked out, the values of μ_L and μ_R are found from the graphs, and $H_{m_1} = H_m \cos\varepsilon = H_m \frac{\mu_L}{\sqrt{\mu_L^2 + \mu_R^2}}$ are determined. Then the values of f, e_1, and e_2, which are substituted in the right member of the last equation, are computed from formulas (3) and (4). A series of values of H_m is computed, the values of the right member corresponding to them are found, and the following graph is constructed in accordance with the latter:

$$Y_1 = \left\{\left[e_2 + \frac{f \cdot l_m}{0.4\pi \cdot w}\cdot(R_0 + R_d)\right]^2 + \left[e_1 + \frac{f l_m}{0.4\pi w}(L_{sd} + L_0)\omega\right]^2\right\}^{\frac{1}{2}}.$$

The point of intersection of this graph with the straight line (Figure 7b)

$$Y_2 = i_m = \text{const}$$

determines the value of H_{m_1}, by which it is possible to find f, and then

$$I_m = \frac{f l_m}{0.4\pi \cdot w}.$$

Similar calculation must be repeated for the series of values of the permeances of the air gap G_v.

This method of calculation is of particular importance at higher frequencies of the supply current (on the order of 2000 to 5000 cycles per second).

SELECTION OF THE DIMENSIONS OF AN EMITTER

The number of ampere turns of the winding of an emitter is related to the dimensions of the latter by the following function:

$$(Iw)^2 = I^2 R \frac{w^2}{R} = P \frac{w^2 q}{\rho l_{ave} w} = \sqrt{P} \cdot \sqrt{\frac{Qf}{\rho \cdot l_{ave}}}$$

or

$$Iw = P \cdot \frac{Q \cdot f_0}{\rho \cdot l_{ave}},$$

where Q is the area of the cross section of the window of the winding ($Q = LH$);

L is the length and H the height of the winding;

l_{ave} is the average length of a turn, equalling $l_{ave} = 2(b_1 + b_2) + \pi H$;

ρ is the resistivity of the winding;

f_0 is the coefficient of filling of the area of the window with copper.

The necessary area of the cooling surface is

$$S_{cool} = \sigma \cdot \Sigma P,$$

or

$$S_{cool} = \sigma \cdot (P_{cu} + P_{Fl}),$$

where P_{cu} is the loss of power in the winding;

P_{Fe} is the loss of power in the magnetic circuit;

σ is the specific cooling surface (with an assigned overheating temperature of 50°C);

S_{cool} is the area of the cooling surface.

Usually $P_{cu} \gg P_{Fe}$ and the principal elimination of heat is accomplished over the lateral face of the winding; hence, in the majority of cases it is permissible to assume that

$$S_{cool} \approx S\delta \approx \sigma \cdot P_{cu}.$$

Assuming that $Q = LH$; $l_{ave} = 2(b_1 + b_2) + \pi H$; $S\delta = 2(b_1 + b_2 + 4H)L$, we find that the maximum number of ampere turns permissible in protracted operation of an emitter must not exceed

$$Iw = L \cdot \sqrt{\frac{f_0}{\sigma\rho}} \cdot \sqrt{\frac{H \cdot (b_1 + b_2 + 4H)}{b_1 + b_2 + \frac{\pi}{2} H}}.$$

When calculating the magnetic surface of air gaps it is necessary to take into account the swelling of the flux.

For a rectangular pole shoe situated against a surface of much greater extent, the permeance of the air gap may be determined by the formula

$$G_{mv} = \frac{(b_1 + 2\Delta b_1)(b_2 + 2\Delta b_2)}{\delta},$$

where δ is the length of the air interval (all dimensions in centimeters).

The values of Δb_1 and Δb_2 are a function both of the length of the air interval and of the extent (a) of the surface opposite the pole and the chamfer angle of the pole α^o. The value of Δb may be expressed in approximation by a function in the form

$$\Delta b = \left[a \cdot \delta \cdot \frac{1}{\alpha}\right]^m = 0.58 \cdot \sqrt{a \cdot \delta \cdot \frac{100}{\alpha^o}}.$$

It is generally assumed that $a \approx (7.5 \div 8.0)\delta$, hence

$$\Delta b \approx 0.58 \sqrt{(7.5 \div 8.0) \cdot \frac{100}{\alpha^o}} \approx \frac{15}{\sqrt{\alpha^o}} \delta.$$

If there is another similar pole opposite this pole, the permeance is determined by the relationships

$$G'_v = \frac{G_v}{2} = \frac{1}{2} \frac{(b_1 + 2\Delta b'_1)(b_2 + 2\Delta b'_2)}{\delta/2} = \frac{(b_1 + 2\Delta b'_1)(b_2 + 2\Delta b'_2)}{\delta}$$

and

$$\Delta b' = \frac{15}{\sqrt{\alpha^o}} \cdot \frac{\delta}{2},$$

by use of which it is possible to determine in approximation the functions $G_{mv} = f(x)$ in the case of the basic constructional forms shown in Figure 1.

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I. M. GRADUSOV

CLASSIFICATION OF THE COPYING DEVICES EMPLOYED IN METAL CUTTING MACHINES

The copying devices employed in metal cutting machines are highly diversified. The mechanically, electromechanically, and hydromechanically controlled devices are the most widespread. Pneumomechanical, photoelectromechanical, electrohydraulic, and other devices are also encountered. In addition, mechanisms with command devices and the so-called electric former in the form of a setting rheostat, functional potentiometer, etc, are counted among copying devices. Analysis of the operation of the copying devices mentioned makes it possible to effect a classification of these mechanisms, which are so diverse in design. The proposed classification is shown in Table 1.

The first subdivision has been selected on the basis of the principle system of copying devices employed in metal cutting machines. This system consists of the following basic units: a former; a copying-measuring instrument, i.e., a clearance gage (impulse converter); an amplifier; a performing member in the feed drive; and a performing member in the machine, i.e., a tool rest.

The units listed perform the following functions:

(1) measurement of an article along the former and conversion of the impulses obtained in measurement of the former-article;

(2) amplification of the impulses;

(3) conversion and transmission of the impulses to the performing member in the feed drive;

(4) transmission of impulses to the performing member, the tool rest;

(5) securing of regenerative coupling.

Conversion of the impulses obtained in measurement of the former-article is absent from mechanically controlled copying devices, and the performing member is accordingly excluded from the feed drive. Nevertheless, they on the whole retain the structure of a copying device, differing from the usual devices only in the degree of automation. Mechanically controlled copying devices are assigned to the category of devices without impulse converters.

Mechanisms controlled by program command devices in the form of drums with cams, a perforated tape, setting rheostats, etc, are also assigned to an independent category, in accordance with the degree of automation. In these copying devices, measurement of the article is not accomplished during the copying process, but before it. The proper functioning of the devices is insured by appropriate adjustment of a program command device in accordance with the dimensions of the article. Measurement of the former-article is the basic and most complex function of the usual copying devices. Despite its complexity, this function is performed by man in the mechanisms under discussion.

It is comparatively easy to exercise programm control of the operation of a machine on the simplest profiles of an article, particularly in the machining of stepped shafts. An example of a mechanism of this type is the well-known device for control of a lathe in the machining of stepped shafts of the L. M. Kaufman design.

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Table 1

Basic criteria of classification of copying devices	Classification of copying devices
1 - By degree of automation	a - Copying devices without conversion of command impulses (mechanical); b - copying devices with a copying and measuring instrument; c - copying devices without a copying and measuring instrument.
2 - By control of feeding speed	a - copying devices with control of a single feeding speed from the clearance gage; b - copying devices with stepless control of feeding speed from the clearance gage; c - copying devices with manual control of feeding; d - copying devices with a command device.
3 - By the system of regulation	a - copying devices of the static system; b - copying devices of the astatic system; c - copying devices of the static system; d - copying devices of the astatic system.
4 - By the method of measurement of loss of adjustment	a - copying devices with measurement in rectangular coordinates; b - copying devices with measurement in polar coordinates; d - in rectangular coordinates; d - in polar coordinates; e - in rectangular coordinates; f - in polar coordinates; g - in rectangular coordinates; h - in polar coordinates.
5 - By provision for regenerative coupling	
6 - Single coordinate, contour, or three-dimensional copying	
7 - By the form of energy employed	

The employment of program control in the machining of complicated profiles is much more complex. However, there are such devices, as, for example, copying devices with functional potentiometers (in particular, with sine potentiometers). In this case, the advisability of freeing the copying machine from performance of its basic function of measurement of an article cannot always be proved. The complexity and lack of necessity of performance of this operation by man limits the employment of such devices only in major serial and mass production.

Manually controlled devices also are included in the category of copying devices without a copying and measuring instrument. In this instance, measurement of an article takes place during the copying process, but it is performed by man. Hence, copying and measurement instruments are absent from them, just as in the case of the copying devices functioning through a program apparatus.

The criterion deciding the classification of copying devices is the number of feeding speeds controlled by means of a clearance gage. They may be divided into two classes in accordance with this criterion, those with a single feeding speed controlled through the clearance gage and those with stepless control of the feeding speed through the clearance gage. The well-known copying devices with electrocontact clearance gage and with electromagnetic couplings in the feed drive of the machine may serve as an example of the first group. To the second group may be assigned devices with an inductive copying and measuring instrument and with electronic and electromechanical amplifiers, as well as the majority of hydraulic copying devices.

The difference in the number of feeding speeds controlled through the clearance gage creates the difference in the possibilities of machining various profiles of an article. In the case of copying of sections of a profile which do not coincide with the direction of cross-feed and longitudinal feed in the machine, the simplest electrocontact copying devices with control of a single feeding speed through the clearance gage can create only a certain step-like form of the trajectory of the cutting tool. The complex copying devices, i.e., those with control of intermediate feeding speeds through the clearance gage can in this case create an approximately rectilinear trajectory of the cutting tool.

The advantage of machining with the aid of a complex copying device such as this manifests itself chiefly in smooth transitions from one form of profile of the former-article to another and in rare changes in the profile being machined. In the case of abrupt and frequent changes of the profile of the former-article, the advantage remains with the simplest copying device with control of a single feeding speed through a clearance gage. In this case, smooth change of speed is not required. Hence, the possibilities of machining both with the aid of the simple and the complex copying device are at any rate identical in the case of machining of the complex profile of an article. When we take into account the fact that the complex copying device, owing to the large number of its members, has a correspondingly greater inertia run-out of the tool rest and hence a larger copying error, the advantage of the simple copying device in the machining of an article with abrupt change of profile becomes obvious. The extensive use of the simplest copying devices, particularly electrocontact devices with electromagnetic couplings in the feed drive is conditioned by this factor.

Nevertheless, in evaluation of copying devices, aside from the question of dependency on possible accuracy and that achieved in the machining of an article, it must be acknowledged that the copying devices with control of intermediate feeding speeds through a clearance gage are more universal. In this case, an approximately uniform tangential feeding speed, which is adopted during adjustment of the machine, is insured for each surface of an article with any inclination.

The subdivision of copying devices next in significance must be made in relation to the system of regulation adopted -- control of the copying process.

Analysis of the operation of electrocontact copying devices indicates the existence of two basic copying systems. The most widespread is the system which utilizes for control of the process of copying the deviation of the stud of the clearance gage, which is proportional to the loss of adjustment between the former and the article. Use is made in such a system of a three-position clearance gage (Figure 1a) which requires regulation of the gap between the contacts upon each change of feeding speed selected in the machine. This complicates considerably the operation of the device.

The electrocontact copying device of the second system, provided with a two-position clearance gage (Figure 1b), does not require regulation of the gap between the contacts of the gage, and this is its superior feature. The first system may be termed static in accordance with its performance, and the second system, astatic.

Figure 1. Diagram of the design of electrocontact clearance gages: a - three-position gage; b - two-position gage.

The copying movement of the clearance gage or cutting tool along the contour of the former or article, as we know, is performed owing to the geometric addition of two feeds perpendicular in direction of movement; the reversible tracking feed, approximately perpendicular to the profile of the former, and the main feed, which is constant in direction, approximately along the contour of the former profile. If control of one of the two feeds indicated, i.e., main or tracking, is provided for in each position of inclination of the stud of the clearance gage of the contact copying device, then we obtain a "step-like" form of the trajectory of the axis of the clearance gage - cutting tool. Such a form is shown in Figure 2 (machining of a recess). In order to obtain a step-like form of the trajectory when the contour of the former is within the limits of two quadrants, it is necessary to employ a three-position clearance gage. The step-like form of the trajectory corresponds to the first of the two control systems indicated.

Figure 2. Step-like form of the trajectory of the axis of a clearance gage: 1 - Tracking; 2 - Main; 3 - Tracking; 4 - Direction of main feed on a milling machine.

Figure 3. Serrate form of the trajectory of the axis of a clearance gage: 1 - Tracking; 2 - Main; 3 - Tracking; 4 - Direction of main feed on a milling machine.

If control of the feed of one direction or the other is provided for in each position of the stud of the clearance gage, this making it necessary to provide for control of the main and tracking feeds simultaneously in one of the positions, then we obtain a "serrate" form of the trajectory (Figure 3). A two-position clearance gage is adequate for obtaining this form of the trajectory. The serrate form of the trajectory corresponds to the second of the two copying systems indicated above. It appears from comparison of the performance of the simple electrocontact copying devices of the different systems that the quality of the copying device with the steplike form of the trajectory, the copying conditions being equal, is superior to the quality of the copying device with the serrate form of the trajectory.

The data derived from the analysis made permit the conclusion that the contact copying devices with a two-position clearance gage, owing to the simplicity of design and to the adequate range and quality of the functions performed, are superior to the similar copying devices with a three-position gage, in the case of slight automation of lathes, milling machines, vertical lathes, and planing and slotting machines.

The performance of the copying devices of both systems is similar, independently of the number of feeding speeds controlled through the clearance gage. This may be illustrated by the example of a static system copying device with control of intermediate feeding speeds through a clearance gage, for example, the device with inductive clearance gage and with electronic and electromechanical amplifiers (T. N. Sokolov design). Despite control of intermediate feeding speeds, in particular through replacement of the clearance gage contacts by coils with inductive reactance, the clearance gage even in this case remains three-positional in design, as are the simple copying devices of the same system. A diagram of the positions of the stud of the clearance gage of such a device is given in Figure 4.

The disadvantages of the system of the simple copying device with a three-position clearance gage are repeated in the copying device with smooth regulation of the feeding speed. In particular, each time the feeding speed selected in the machine is changed, it is also necessary to adjust the gap in this device, in this case between the cores of the coils of the clearance gage.

Figure 4. Diagram of the positions of the stud of the clearance gage: 1 - curve of values of tracking feed; 2 - curve of values of main feed; 3 - threshold of sensitivity of clearance gage; 4 - intermediate position; 5 - gap between the extreme positions in the clearance gage; 6 - positions between the extreme positions in the clearance gage.

The general characteristics of the copying device of any system are retained upon change in the type of energy employed for the drive of the tool rest feed and for control of the copying process. Thus, for example, one may cite hydraulic copying devices with smooth regulation of the feeding speed (of the same astatic system) in which control of the copying process is achieved by a single slide valve STAT edge, i. e., according to the type of two-position clearance gage the stud of which has no intermediate position (Figure 5).

The hydraulic copying system of the static system retain all the characteristics of this system, aside from the type of energy employed. In Figure 6 is given a diagram of such a device with a copying slide valve for three positions, which correspond to the position of the three-position, electromechanically controlled copying device.

Figure 5. Hydraulic copying device of the astatic system.

Copying devices should be further classified as to the method of measurement of the adjustment between the former and the article, the measurement being made in rectangular or polar coordinates. In accordance with this, any copying device, either of the static or the astatic system, may be assigned to one group or another, depending on the method of measurement adopted.

Measurement of loss of adjustment in polar coordinates is most frequently made for the purpose of automation of contour copying at 360°. Consequently, accomplishment of complete automation of contour copying usually corresponds to this criterion of the method of measurement in polar coordinates. An example of copying devices with measurement in polar coordinates is offered by such devices in which the stud of the copying and measuring instrument is not located centrally, but with a certain eccentricity with respect to the axis of the clearance gage.

Measurement of loss of adjustment in polar coordinates is characterized by the fact that gaging of the profile of the former-article takes place at a point which does not coincide with the cutting edge of the tool, at the distance of constant eccentricity of the stud of the clearance gage. This latter circumstance is the essential distinction between measurement of loss of adjustment in rectangular coordinates.

On the whole, one may limit oneself to the criteria listed above when classifying copying devices. Other possible criteria, but ones of secondary significance, may be the design of the mechanism insuring regenerative coupling in the functioning of the copying device and the function of the copying device, for single-coordinate, contour, or three-dimensional copying. The type of energy employed determines only to a slight extent the quality of the copying device and is thus utilized as an auxiliary criterion in the classification proposed.

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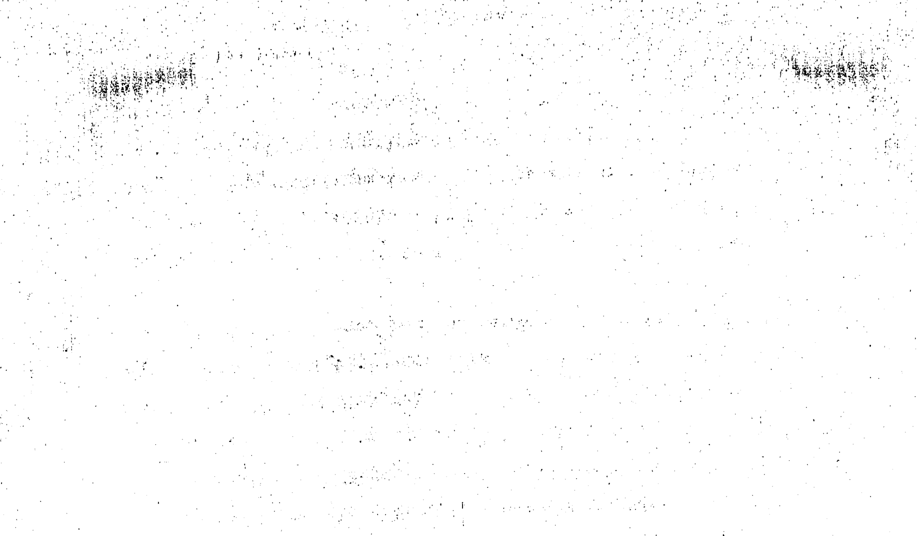

Figure 6. Hydraulic copying device of the static system:
1 - under pressure from pump.

P. S. ARZAMASTSEV

EMPLOYMENT OF TRACKING DRIVE SYSTEMS

FOR THE AUTOMATION OF COPYING MACHINES

An account is given in this article of the results of the work conducted at the NIAT on electromechanical and electrohydraulic tracking systems of copying machines which operate both with a rigid former and with a drawing.

Figure 1. Graphs of the feeding speeds of a contour copying system: S - deviation of the shaft of the copying lug in the normal direction toward the contour (error); S_0 - double maximum deviation; V_x, V_y - speeds of longitudinal (x) and lateral (y) feeds; I, II, III, IV - quadrants of circle; 1 - former.

The most important problem in this field was mechanization of the production of flat steel master forms and their duplication. The tracking system of the machine was to insure (1) automatic copying of closed profiles; (2) high copying accuracy ($\pm$ 0.1 mm); (3) high output (high feeding speeds).

The existing copying systems could not insure meeting of the requirements with respect to automatic copying of closed profiles and output.

The scientific research and project works conducted at the NIAT have made it possible to create a new profile copying system with a rotary copying head, characteristic of which is the presence in it of elements which track the angle of inclination of the profile being copied.

The necessity for additional tracking of the angle of inclination of the profile may be elucidated from the graphs of the feeding speeds of the systems given in Figure 1. This system has no elements which track the angle of inclination of the profile and operates only through the deviation (error) of the copying lug from the neutral position in the direction of the normal to the profile being copied.

In the copying of profiles in the form of a closed circle, each of the graphs of feeding speeds shown (graphs a and b) can provide only for the copying of a semicircle

(180-degree profiles), graph a for the upper semicircle, and graph b for the lower.

Figure 2. Diagram of the NIAT rotary copying head:
1 - axis of copying instrument; 2 - central copying lug; 2' - auxiliary lug; 3 - brushes of sine potentiometer; 4 - sine potentiometer; 5 - flat spring compressing auxiliary lug against former; 6 - inductive measuring instrument; 7 - amplifier; 8 - former

The copying instrument system shown in Figure 1, without additional switches thus provides for only one of the graphs shown and consequently can copy profiles not exceeding 180°. This is due to the fact that for transition from one feeding speed graph it is necessary each time that angles which are multiples of 180° are passed to change simultaneously the directions of both speeds and the law of their change.

This is a typical three-dimensional copying system.

Figure 2 shows a diagram of the NIAT rotary copying head, which makes it possible to track simultaneously the error and angle of inclination of the profile. The system has two copying lugs, a central 2 and an auxiliary 2'. The former is connected to an inductive measuring instrument 6 and the latter, which is constantly in contact with the master form, during copying rotates the brushes 3 of a sine potentiometer 4 to the angle corresponding to the angle of the profile at a given point. Such an arrangement makes it possible to achieve displacement of the signal at the sine potentiometer so that the signals at its exit will be

$$u_x = F_1 (S) \sin \alpha ,$$

$$u_y = F_2 (S) \cos \alpha ,$$

where S is the magnitude of the error;

α is the turning angle of the brushes (profile angle).

The exit signals and feeding speeds are in this case sinusoidal functions of the angle of inclination of the profile with an amplitude depending on the magnitude of the error. This principle has served as the basis for construction of the NIAT pro-

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file systems.

NIAT PROFILE COPYING SYSTEMS OPERATING BY RIGID MASTER FORM AND BY DRAWING

A. The electromechanical profile milling layout machine (KSF-1M)

The simplest example of employment of a rotary copying head for automatic copying of closed profiles by a rigid master form is the KSF-1M profile drilling-milling machine, designed for the laying out of duralumin sheets.

Figures 3 and 4 show an overall view and a skeletal diagram respectively of this machine. The longitudinal carriage 1 (Figure 3) moves along the bed, along guides, by means of rollers set in rotation by the longitudinal feed electromotor. The rigidly secured copying head 4 and milling head 5 on the lateral carriage 2 are displaced by the electric motor 3, also by means of rollers.

The copying head of the KSF-1M machine (Figure 5) has a rotary member 1 and a fixed body 2. The rotary member has an eccentrically located auxiliary lug 3. The main copying lug 4 is located in the lower part of the moving body of the head in such a way that its axis coincides with the axis of the head. In the upper part of the fixed body of the head is mounted a sine potentiometer 5, contact with the upper surface of which is made by two brushes 6 and 7; the latter are rigidly secured by means of the brush holder 8 to the shaft of the head and rotate together with the auxiliary lug. A flat spring 9 wound by a ratchet device 10 is employed to keep the auxiliary lug constantly compressed against the master form.

In the operating position, the auxiliary lug moves ahead of the main copying lug, so to speak "exploring" the profile of the master form. During one circuit of the closed profile, the auxiliary lug, and consequently the brushes also, rotate 360°. The tracking device in this machine creates through the longitudinal and lateral feed electric drive motors the necessary force on the central copying lug to prevent the latter from withdrawing from the master form.

As may be seen from Figure 4, the voltages taken from the brushes and changing sinusoidally for one brush and cosinusoidally for the other, move to the control windings of the corresponding electromechanical amplifiers which feed the feed motors. Coincidence of the resulting feeding speed at any point of the profile with the tangential is not sufficient in this case to prevent the copying lug from breaking away from the master form. Thanks to special adjustment of the potentiometer, this speed is always directed into the profile, i.e., onto the master form. The amount of the "lead" angle of the speed reaches 15 to 20°.

In order to protect the copying lug from excessive pressures against the master form, sliding couplings are provided in the kinematic chains of the feeds, for which purpose adjustable electromagnetic couplings with a ferromagnetic filler may be successfully used.

Industrial utilization of the machines has demonstrated their high efficiency (feeding speed of up to 1800 mm/min) and satisfactory accuracy ($\pm$ 0.2 mm). A certain complexity of the electrid drive must be considered as a disadvantage of the KSF-1M machine.

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Figure 3. Overall view of the KSF-1M profile milling layout machine: 1 - longitudinal carriage; 2 - lateral carriage; 3 - electric motor of lateral carriage; 4 - copying head; 5 - milling head.

Figure 4. Skeletal diagram of the KSF-1M machine: 1 - copying lug; 2 - auxiliary lug; 3 - copying head; 4 - milling cutter; 5 - milling head; 6 - electromechanical longitudinal feed amplifier; 7 - electromechanical lateral feed amplifier; 8, 13 - sliding couplings; 9, 12 - reduction gears; 10, 11 - electric motors of lateral and longitudinal feeds; 14 - brushes of sine potentiometer; 15 - master form; 16 - article; 17 - back coupling; 18 - force of spring.

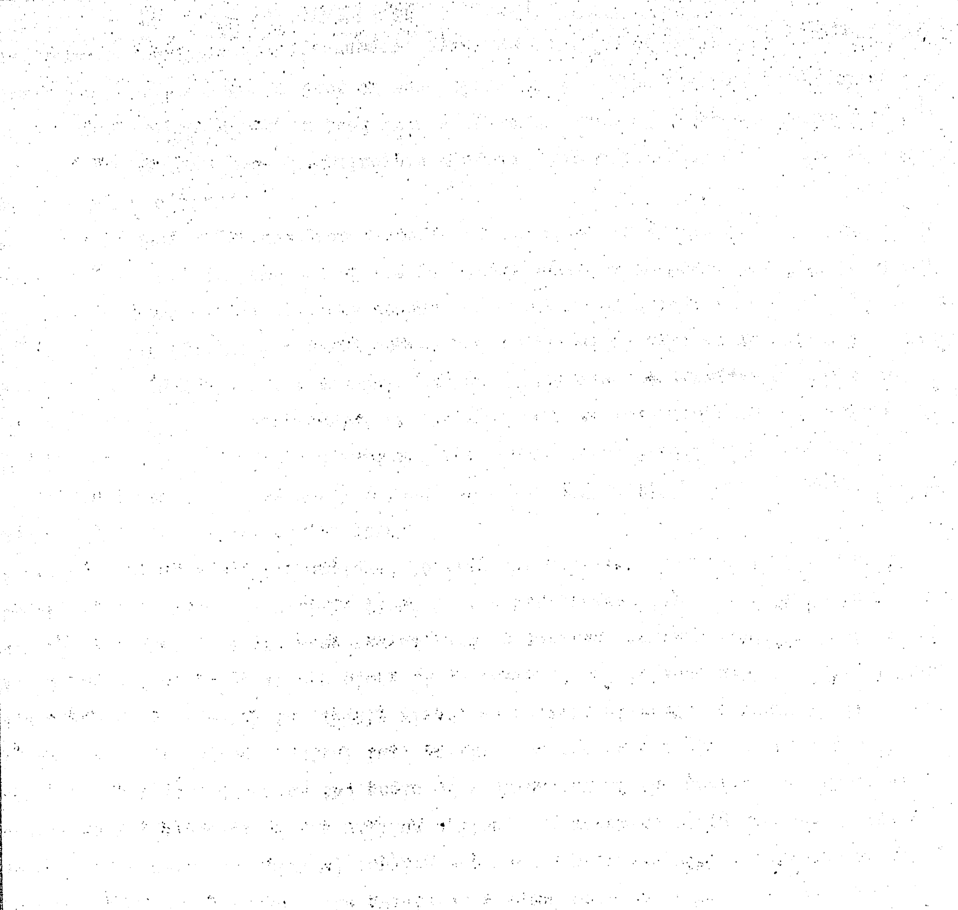

Figure 5. Copying head of the KSF-1M.

B. <u>Electromechanical tracking system with sine emitter</u>

Figures 6, 7, and 8 give a skeletal diagram of the electromechanical tracking system of a profile milling machine, a structural view of the copying instrument, and a diagram showing its principle of operation. This is the first system in which the problem of automatic copying of closed profiles has been fully solved.

As may be seen from Figure 6, the copying and milling heads 1 and 5 are fixed on the crosspiece of the machine 8; the plate bearing the master form 6 and blank 7 is moved by two guide screws which are rotated by separate direct current motors 10 and 11. The speeds of rotation of both guide screws are regulated by signals from both copying lugs along two control channels, those of the longitudinal (V_x) and lateral (V_y) feeds. Distribution of the signals along both amplification channels of the system is accomplished by a sine potentiometer 4, which in this system is the mixer of the signals of central and angular deviations. Each of the channels of the system consists of an intermediate electronic amplifier 12 and 13 and an electromechanical amplifier 14 and 15. Preamplification of signals from deviation of the central lug 2 is accomplished by single-tube electronic amplifiers 12 and 13.

A general view of the copying instrument, which is constructed on the principle described above (Figure 2), is given in Figure 7. It has an outer fixed body 8 and an inner moving body, rigidly connected with which is the auxiliary copying lug 3,

Figure 6. Skeletal diagram of the electromechanical tracking system of a profile milling machine: 1 - copying head; 2 - central lug; 3 - auxiliary lug; 4 - sine potentiometer; 5 - milling head; 6 - master form; 7 - article; 8 - crosspiece of the machine; 9 - reduction gears; 10, 11 - electric feed motors; 12, 13 - electronic amplifiers; 14, 15 - electromechanical amplifiers; 16 - additional voltage generator; 17 - drive motor.

the lower part of which ends in a cap.

An inductive meter of central deviations (errors) consisting of two magnetic coils 2 and an armature 9 mounted in a moving body serves as the basic element of the copying instrument. The position of the armature is fixed by the joint action of two regulating spiral springs. The armature is acted on by a lever, the lower part of which, the central (sensitive) copying lug 10, is provided with a calibrated cap equalling the diameter of the milling cutter. In movement along the master form, the auxiliary lug, tracking the angle of inclination of the profile of the former, is displaced around the central lug, rotating to the same angle the moving body, in the upper part of which is rigidly fixed, on a vertical shaft, a brush holder with two brushes 7 situated in one plane at an angle of 90° to each other. Both brushes, moving in a circle of the same radius, are in constant contact with the flat sine potentiometer 1, which is rigidly fixed in the stationary body. An asynchronous motor 5 of momentum operation is employed to keep the auxiliary lug constantly pressed against the edge of the master form.

Such a design of the copying head, with the sufficiently small distance between the points of contact of both lugs and the master form, practically insures coincidence of the plane of oscillation of the main lug and the normal to the profile at the point of contact of the central lug, and hence insures equality of the angle of inclination of the profile being copied and the turning angle of the brushes. A special potentiometer which provides adequate smoothness of change of the voltage and an only slightly distorting form of the potential curve is employed in order to obtain voltages on them which are proportionate to the sine and cosine of the angle of the profile being copied.

As may be seen from the principle diagram (Figure 8), each of the two sections of the sine potentiometer is feed from a separate anode of a two-electrode tube L_2. Since the grid voltages of each of the halves of this tube are rectified voltages of the arms of the inductive instrument, the currents in both the sections of the poten-

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Figure 7. General view of the rotary copying instrument: 1 - sine potentiometer; 2 - magnetic coils; 3 - auxiliary lug; 4 - selsyn transmitter; 5 - momentum motor; 6 - reduction gear; 7 - brushes; 8 - fixed body; 9 - armature of inductive emitter; 10 - central lug.

tiometers are adjusted to be equal when the armature (the copying lug) of the inductive instrument is in the zero position. When deviations (errors) of the armature exist, the currents in both the sections change in proportion to these deviations in such a way that the current increases in one half of the sine potentiometer, and decreases in the other by the same amount. This makes it possible to obtain currents on the brushes of the potentiometer in the form of a sinusoid and cosinusoid with variable amplitude linearly dependent on the error. Thus, the sine potentiometer in this system is a signal mixer.

Mixed signals, preamplified by the two-channel tube amplifier L_3, L_4, move to the control windings OU_1, OU_2 of the electromechanical amplifiers EMU_1, EMU_2 of the longitudinal and lateral feeds. Owing to the constant component of the voltage in the grid circuits of the tubes L_3 and L_4, the voltage on the brushes of the EMU will be of one sign, and this greatly reduces distortions of the characteristics of the amplifiers due to the phenomenon of hysteresis. An additional voltage generator is counter-connected in the armature circuits of the EMU in order to obtain variable sign voltages on the drive motors of the longitudinal and lateral feeds. As experience has demonstrated, such connection perceptibly reduces the zone of non-sensitivi-

Figure 8. Principle diagram of the tracking system of a profile milling machine: 1 - sine potentiometer (mixer); 2 - inductive instrument; 3 - EMU_1; 4 - OU_1; 5 - VDG; 6 - OU_2; 7 - EMU_2; 8 - L_3; 9 - L_4; 10 - L_2; 11 - L_1.

ty of the electric drive motors.

Figure 9 gives a calculation graph of the speeds of the longitudinal and lateral feeds in relation to angle α -- the inclination of the profile being copied. The thin lines limiting the shaded area determine the zones of possible change of feeding speeds in the event of deviations of the central lug, i.e., when errors are present. The curves 1 on the graphs are the main feeding sppeds V_{0x} and V_{0y} towards which the system strives upon correction.

The following conclusions, which reveal the nature of copying systems as an automatic regulating device, have been drawn as the result of analytic and experimental research of the system under discussion:

1. When an article is being machined, the dopying tracking system strives for an equlibrium position of the speeds, which position is determined by the <u>main feeding speeds</u>, which in the system under discussion stand in sinusoidal and cosinusoidal relation to the angle of inclination of the profile being copied.

2. The static accuracy of the copying system is determined exclusively by the presence of distortions of the sinusoidal form of the main feeding speeds.

An electrohydraulic profile milling machine for the production of master forms

Figure 9. Graph of feeding speeds and correction zone.

(the KFS-3A) has been developed and is being produced as a combined machine, i.e., one for copying both by a rigid master form and by drawing. It is designed for the high accuracy production and duplication of steel master forms of large dimensions (4000X1700mm). The machine has an electrohydraulic tracking drive the use of which makes it possible to utilize hydraulic cylinders of small dimensions as the main tracking drive.

The Photoelectric-hydraulic copying system operating with a drawing. As we know, for over twenty years work has been conducted, both in the Soviet Union and abroad, on the creation of an industrial copying system which would operate by drawing. However, for a long time this work produced no results, in consequence of the instability of the systems, an instability conditioned chiefly by the stability of the characteristics of the phototubes themselves. Success was achieved in the construction of a stable system only after the development at the Academy of Sciences of the USSR of a phase-impulse system of control of phototubes and of a tracking copying system capable of copying closed profiles, at the NIAT.

An industrial electrohydraulic tracking system for the profile milling machine was created on the basis of these developments. The KFS-3A machine, described above, for copying by drawing, is equipped with such a system. A special electric circuit (Figure 10) based on the phase-impulse method of control of the phototubes has been developed for control of the hydraulic adjusting system by optical means (by drawing).

A photocopying instrument, a constructional view of which is given in Figure 11, has been developed on the basis of this circuit. The lower, revolving part of the instrument has two lamps 3, a lens 2, a synchronous motor 8 with a hollow shaft, and a phototube 9. The light reflected by the surface of the drawing falls on the phototube through the hollow shaft of the electric motor 8, which has in its upper end a blind with an eccentrically situated aperture. When a black line is present on the drawing, twice in one revolution of the motor this aperture crosses its image; hence

Figure 10. System of control of a phototube by phase-impulse methods: 1 - DF; 2 - L_2; 3 - L_3; 4 - L_4; 5 - L_1; 6 - DG; 7 - Turning angle circuit; 8 - Error circuit; 9 - Amplifier; 10 - FE.

the phototube is darkened twice. The impulses of the photoelectric current which then occur are preamplified and move to a grid transformer TC (Figure 10) which has four secondary windings. The thyratrons of both the turning angle circuit and the error circuit are controlled by these voltage impulses. The error circuit is the electrohydraulic tracking system, which is designed for correction of the error. When it is absent, the currents passing through both halves of the coils of the transmitter of the hydraulic system DG are mutually equal and the slide valve connected to this coil is in the neutral position. When the phase of the impulse changes, this usually being due to an error, the currents change differentially and the slide valve is displaced in the direction of its reduction. The turning angle circuit is the circuit of the ion tracking drive, which checks the angle of inclination of the tangent to the profile being copied. In this case, when an angular error is absent, the voltage of the positive half-period of one thyratron is balanced by the voltage of the negative half-period, which moves through the selenium rectifier of the other half of the circuit. The voltage on the armature of the motor DF is then zero.

When the phase of the impulse of the photoelectric current changes, this balance is disrupted, and the electromotor LF rotates the photoelectric head 1 (Figure 11), and together with it the brushes of the sine potentiometer 5 and the rotor of the selsyn transmitter 6, in the direction of reduction of the angular error.

Further control of the feeding speeds by signals from a sine potentiometer, as well as correction of errors by a hydraulic corrector, are accomplished as in the KFS-3A machine when in operation with a rigid master form.

A machine equipped with the tracking system described above has been integrated into industry and is giving stable performance under factory conditions.

A record feeding speed of 1000 mm/min and a high copying accuracy reaching ± 0.1 mm have been established on these machines operating by drawing.

Figure 11. Photocopying instrument: 1 - photocopying head; 2 - lens; 3 - lamps for illumination of drawing; 4 - electric motor; 5 - sine potentiometer; 6 - selsyn transmitter; 7 - line of drawing; 8 - synchronous motor; 9 - phototube.

Special mention should be made of the demands made of the drawings themselves. The operation of the machine is stable with drawings made with black or brown ink on a light background (light gray, light blue, etc.). The line may be from 0.1 to 0.3mm thick. The machine copies the center (axis) of the line. As experience has shown, the drawings themselves can be made with an accuracy reaching ± 0.10mm, which is adequate for many branches of mechanical engineering.

It may then be stated, on the basis of results of operation of a machine copying from a drawing, that the photocopying tracking system is fully reliable in operation and may be recommended for industry.

G. I. KAMENETSKII

HYDRAULIC TRACKING DRIVES OF COPYING MACHINES

Hydraulic tracking drives are being extensively employed in copying machines.

Hydraulic tracking drives have a constant action, brought about by back coupling of the performing mechanism on the controlling slide valves, in contrast to the ordinary self-acting hydraulic drives, in which this action is performed only at the moments control commands are given.

Hydraulic tracking drives are built into copying machines or are produced in the form of copying attachments for general-purpose machines.

The hydraulic tracking drive is superior to other tracking drives in a number of features, the principal of which are the following:

1. Small driven mass of the hydraulic motors. The mass of the moving parts of reciprocating-motion hydraulic motors (cylinders) is small in relation to the mass of the performing member displaced, while the driven mass of the rotors of direct-current electric motors is much greater (approximately 1000 times) than the mass displaced. The driven mass of rotary-motion hydraulic motors is also much less (approximately 100 times) than the driven mass of direct-current electric motors.

The relatively small driven masses make it possible to produce tracking systems with hydraulic drive in accordance with simple diagrams and in so doing to insure more accurate copying than with other drives.

2. The possibility of gapless connection of the hydraulic motors with the machine units displaced. This is very important, since the copying process entails reversal, and gaps in the kinematic chain result in fluctuations which complicate the copying.

3. Simplicity of design, reliability in operation, and resistance to wear.

The tracking device generally consists of simple controlling slide valves and hydraulic motors. Auxiliary motions (clamping of the part, accelerated feeds, etc) and other automation of the machining cycle can be accomplished with the aid of standardized hydraulic equipment.

The units of the hydraulic drive which have oil as their operating medium practically do not wear out, with the exception of the pumps, which work chiefly at low pressures and for this reason have a long life.

The hydraulic tracking systems of copying machines belong to the category of static systems, in which the force and velocity of displacement of the performing members are determined by deviation of a clearance gage from a position with zero force and velocity of displacement of the performing member.

In the majority of the systems of copying machines, the clearance gage, which is in contact with the master form, is mechanically connected with the controlling slide valve.

However, there are systems in which the clearance gage and the slide valve are connected through a hydraulic, pneumatic, or electric control system.

Pneumatic-hydraulic and electric-hydraulic tracking systems have hardly been used at all in domestic machine building.

The following copying systems are distinguished, in accordance with the number of tracking motions:

a) single-coordinate systems with independent speed of the driving motion;

b) single-coordinate systems with dependent speed of the driving motion;

c) two-coordinate systems;

d) three-coordinate and multiple-coordinate systems.

The sphere of employment of the tracking devices listed above is determined chiefly by the possible angles of the profiles to be copied.

In the case of a flat profile with a total angle not exceeding 120°, use is made of the most reliable and simplest tracking system, the single-coordinate system with independent speed of the driving motion.

Such a device is situated on lathes perpendicularly, parallel, or at an angle of 60° or 45° to the longitudinal axis of the machine. The slanting arrangement makes it possible to machine the most frequently encountered parts -- stepped shafts -- by the copying method.

In the case of a profile with an angle reaching 180°, use is made of single-coordinate devices with dependent speed of the driving motion, and of two-coordinate devices in the case of a profile angle exceeding 180°. Three-coordinate and multiple-coordinate devices are employed in the copying of bodies limited by curvilinear surfaces forming various angles of inclination to the base plane; in the copying of double-curved surfaces with the tool set tangentially (or normally) to the surface being machined; in the copying of closed curves of surfaces sections of which are situated at differing altitudes with respect to the plane of the bedplate; etc.

THE BASIC HYDRAULIC CIRCUITS

Figure 1 shows a diagram of a copying machine with a single-coordinate tracking device with independent driving motion.

The part being machine 1 and the master form 2 are secured on the bedplate 3, which moves horizontally at the speed v_{dr}.

The milling head 4 is displaced vertically be means of a hydraulic cylinder controlled by a slide valve 6, the body of which is rigidly connected to the body of the milling head. Oil from a pump is fed into the circulation chamber 8 of the slide valve, whence it is directed into one of the cavities of the cylinder. When the slide valve is displaced upwards a certain distance from the central position, the oil moves through the circulation chambers 8 and 9 into the cavity 10 of the cylinder, lifting the milling head and forcing the oil from the cavity 5 to the overflow through circulation chambers 11 and 12 of the slide valve.

The upward movement continues until the slide valve body rising with the milling head overlaps the oil ducts in the cylinder cavity, i.e., until the slide valve in the cylinder body occupies the initial position. When the slide valve 6 is displaced a certain distance downwards, the oil moving from the pump is directed into the cavity 5 and displaces the milling head downwards, forcing the oil from cavity 10 to the overflow through the circulation chambers 9 and 7. Motion ceases when, as in upward movement, the initial relative position of the slide valve and its body is restored.

In the machining process, the master form 2, which is in contact with the slide valve 6, continuously displaces the latter, and the milling head 4 duplicates the

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Figure 1. Diagram of a single-coordinate tracking system with independent driving motion: a - overall diagram; b - relative position of the performing member and clearance gage when $p = 0$, $v = 0$; c - relative position of the performing member and clearance gage at p_{max} and v_{max}; d - x_{max}; e - $p_{overflow}$; f - p_{pump}; g - $p_{overflow}$; h - removalbe allowance; i - v_{driv}.

movement of the slide valve 6.

Figures 2 and 3 give diagrams of the copying rests of the lathes of the ENIMS and the SS Ordzhonikidze Machine Tool Factory.

In contrast to the diagram given in Figure 1, in lathes the master form and the article are stationary (except for the rotation of the article and displacement in adjustment) and the tool simultaneously participates in the driving and copying motion. The driving motion is made in the longitudinal direction of the mechanical transmission common in lathes.

The ENIMS tracking drive (Figure 2) has a differential cylinder. Oil is fed through a slide valve 2 into the cylinder cavity 1. Cylinder cavity 3 is constantly connected to the pump 4. When the tracking slide valve moves downwards, cavity 1 is connected to the overflow and the oil pressure in cavity 3, and the copying rest is displaced towards the center of the machine together with the body of the slide valve, until the initial position of the slide valve and its body is restored. When the slide valve moves upward, cavity 1 is connected to the pump and the copying rest is moved from the center of the machine by the oil pressure in cavity 1.

In the copying rest of the S. Ordzhonikidze Factory (Figure 3), the oil is fed into cavity 1 through a damping port 5.

From cavity 1 the oil can pass to the overflow through circulation chambers 6

Figure 2. Diagram of the ENIMS-designed lathe copying rest: 1 and 3 - cylinder cavities; 2 - slide valve; 4 - pump.

and 7 and the slide valve 2. When slide valve 2 is displaced, the pressure in cavity 1 changes, forcing the rest to follow the movements of the slide valve.

The hydraulic diagram of the principle of the rest of the "Red Proletariat" Factory is similar to the hydraulic diagram shown in Figure 1.

Figure 4 shows a diagram of the copying milling machine of the Odessa Milling Machine Factory. The body of the slide valve is fixed, and the table with the article to be machined and three-dimensional master form performs the driving and copying motions.

The driving motion of the table is performed mechanically. The crosspiece 4 is balanced with the table 5, which moves along it, by the pressure in the hydraulic system acting on the piston of the cylinder 2. A spring 7 compresses the tracking finger against the three-dimensional master form through a slide valve 3 and lever 6. In this process, aperture A is great enough so that the discharge of pump 1 passing through it is at the pressure necessary for balancing the vertical component cutting forces and the weight of the table and crosspiece.

In the case of the direction of driving motion indicated in the diagram, slide valve 3 is lifted by the master form, aperture A becomes smaller, the pressure in the hydraulic system increases, and the table with the crosspiece rises, lowering slide valve 3 through lever 6, until the pressure decreases to the extent that the equilibrium of the active forces is restored. The tracking drive, keeping the area of the aperture A almost constant, raises and lowers the table in conformity with the profile of the master form, in such a way that the cutting tool reproduces the con-

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Figure 3. Diagram of the copying rest of the lathe designed by the S. Ordzhonikidze Machine Tool Factory: 1 - cylinder cavity; 2 - slide valve; 3 - master form; 4 - article; 5 - damping port; 6 and 7 - circulation chambers of slide valve; 8 - copying rest release handle; 9 - limb of radial adjustment displacements of the cutting tool.

Figure 4. Diagram of the copying milling machine designed by the Odessa Milling Machine Factory: 1 - pump; 2 - cylinder; 3 - slide valve; 4 - crosspiece; 5 - table; 6 - lever; 7 - spring; 8 - tracking finger; 9 - cutting tool; 10 - v_{driv}.

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Figure 5. Diagram of the ENIMS MG933 copying device: 1 - pump; 2 - driving motion hydraulic motor; 3 - damping aperture; 4 - clearance gage; 5 - controlling slide valve; 6 - copying motion hydraulic motor; 7 - clearance gage slide valve; 8 - master form; 9 - clearance gage finger; 10 - cavity of controlling slide valve; 11 - spring; 12 - overflow valve.

figuration of the master form on the article to be machine.

In Figure 5 is shown a diagram of the ENIMS designed, type MG933 single-coordinate copying device with dependent driving motion.

The device makes it possible to accomplish duplicating machining on universal lathes.

The oil fed by the pump 1 moves simultaneously into the controlling slide valve 5, the driving motion hydraulic motor 2, and, through the damping aperture 3, into the clearance gage 4.

When the controlling slide valve 5 is in the central position, the copying motion hydraulic motor 6 does not revolve; when the slide valve is displaced, it revolves at a speed roughly proportionate to the displacement. When the direction of displacement of the slide valve from the central position changes, the direction of rotation of the hydraulic motor 6 also changes.

The speed of rotation of the driving motion hydraulic motor 2 also is determined by the position of the slide valve 5: when the latter is in the central position, the speed is the greatest; as it is displaced from the central position, the speed decreases; and when the slide valve is in the extreme positions, the hydraulic motor does not revolve.

The driving motion speed decreases as the copying motion speed increases; an

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approximately identical amount of feed is hereby achieved with different angles of the master form, as well as the possibility of machining profiles with a total angle reaching 180°.

The connection between the controlling slide valve 5 and the clearance gage 4 is hydraulic; when the clearance gage finger 9 is deflected by the master form 8, the slide valve 7 of the clearance gage moves, the pressure in the cavity 10 of the controlling slide valve changes, and the controlling slide valve 5 is displaced to the position of equilibrium between the forces of pressure in the cavity 10 and the force created by the powerful spring 11.

The hydraulic connection between the controlling slide valve and the clearance gage has been introduced in order to simplify mounting of the clearance gage on universal lathes. Motion is transmitted from the hydraulic motors of the device to the motion screws of the lathes through telescopic splined shafts and articulated couplings.

The device has been tested in various copying jobs, including the machining of blades of axial turbines and pumps.

Figure 6 shows a diagram of the single-coordinate tracking system with dependent speed of the machine, of the S. Ordzhonikidze Machine Tool Factory.

The tracking motion is effected through a cylinder 1, controlling slide valve 3, and throttle 5; the driving motion, through a cylinder 2, regulator 4, and throttle 6. The pressure in front of the throttle 5 (the leakage in the slide valve 3 being ignored) is proportionate to the square of the speed of the tracking motion, and the pressure in front of the throttle 6 is proportionate to the square of the speed of the driving motion. The pressure created by the throttles 5 and 6 is led to the cavities 7 and 8 of the springed slide valve 9 so that the sum of the forces of pressure on the slide valve when the latter is in the position of equilibrium remains almost constant, this insuring constancy of the sum of the squares of the speeds of the driving and copying motions, and, consequently, feeding stability during copying.

A diagram of the two-coordinate tracking system of the Odessa Milling Machine Factory is shown in Figure 7. The part to be machined and the master form are secured on the table of the machine, which is moved in two mutually perpendicular directions by cylinders 1 and 2. The system is fed by a twin pump: the high-pressure pump 3 feeds oil through the controlling slide valves 5 and 6 into the cylinders 1, 2; the low-pressure pump 4 feeds oil through damping apertures 13 into the slide valves 11, 12, of the clearance gage. The finger 7, which comes in contact with the master form 8, is secured in a bar 9 which has in its center a spherical bearing. Rocking of the bar 9 is limited by a ring 10. Resting against the bar 9, near its upper end, are the two slide valves 11, which are situated parallel to cylinder 1, and the two slide valves 12, which are situated parallel to cylinder 2. The connection between the clearance gage and the controlling slide valves 5 and 6 is hydraulic: when the bar 9 is deflected by the master form, the slide valves 11, 12 of the clearance gage are displaced, the pressure in the cavities 14 changes, and the controlling slide valves are displaced to the position of equilibrium between the forces of the pressure in the cavities 14 and the forces developed by the springs 15. Four plungers 16, situated at an angle of 45° to the slide valves 11, 12, are connected to the latter in such a way that they represent a hydraulic motor which strives to turn the bar 9 within the limits of the clearance in the ring 10.

Figure 6. Diagram of the single-coordinate tracking system with dependent speed of the driving motion, of the copying lathe designed by the S. Ordzhonikidze Machine Tool Factory: 1,2 - tracking and driving motion cylinders; 3 - controlling slide valve; 4 - regulator; 5,6 - throttles; 7,8 - cavities of regulator slide valve; 9 - regulator slide valve; 10 - pump; 11 - overflow valve.

The master form 8 is moved to the finger 7 and stops the bar 9 in the deflected position; the finger is compressed against the master form by the plungers 16 and forces the table to move along the master form. The direction of feed in copying is determined by the sequence of connection of the slide valves 11,12 with the plungers 16, and the amount of feed is fixed by change in the rigidity of the springs 15 by movement of the blocks 17. The valves 18 are designed to maintain a constant pressure drop in the overflow apertures of the controlling slide valves 5,6. Two principles of construction of two-coordinate tracking systems are illustrated in the diagram shown.

1. The clearance gage finger is constantly deflected along a tangent to the profile to be copied, by means of a special mechanism. Deflection of the clearance gage is effected directly by a hydraulic motor, as described above (Figure 7) which deflects the clearance gage through a supplementary tracking system, or by an electric motor, which deflects the clearamce gage by means of a magnetic roller.

2. The projections of the deflection of the clearnace gage in two mutually perpendicular directions of movement of the performing member control two corresponding single-coordinate tracking systems; the controlling slide valves may be connected mechanically or hydraulically (Figure 7) to the clearance gage, or through pneumatic and electric connections.

Three-coordinate and multiple-coordinate tracking systems represent combinations of single-coordinate tracking systems, control of which is generally accomplished through several clearance gages instead of one.

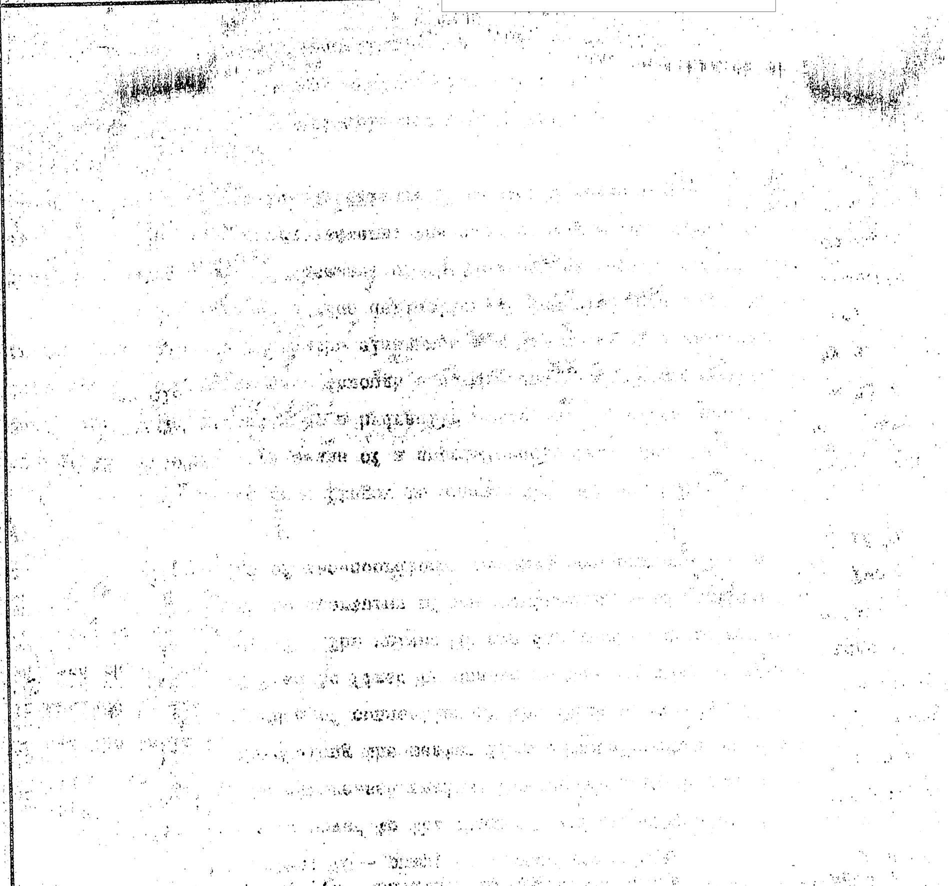

Figure 7. Diagram of the two-coordinate tracking system of the Odessa Milling Machine Factory: 1,2 - cylinders; 3,4 - pumps; 5,6 - controlling slide valves; 7 - finger; 8 - master form; 9 - bar; 10 - ring; 11,12 - slide valves; 13 - damping apertures; 14 - cavities of controlling slide valves; 15 - springs; 16 - plungers; 17 - spring blocks; 18 - valves; 19 - overflow.

PRESSURE IN THE WORKING CAVITIES OF A CYLINDER

With a differential cylinder (Figure 8a), the pressure in the rod cavity (cylinder cavity with a relatively small area F_2) is fixed as equal to the pressure maintained by the valve p_{val}.

The condition of equilibrium is used as the basis for determining the pressure in a rodless cavity. Assuming the left to right direction as positive and projecting all the forces to this direction, we obtain

$$p_1 = p_{val} \frac{F_2}{F_1} - \frac{R}{F_1},$$

where F_1, F_2 (cm^2) is the large and small area of the cylinder of the performing member;

R (kg) is the load.

The numerical value of R must allow for the magnitude and direction of the cutting forces, weight (in the case of vertical movements), and the forces of friction.

The condition $F_1 = 2\,F_2$ is generally observed to obtain identical forces in the case of movements in both directions in differential cylinders. In this case, the

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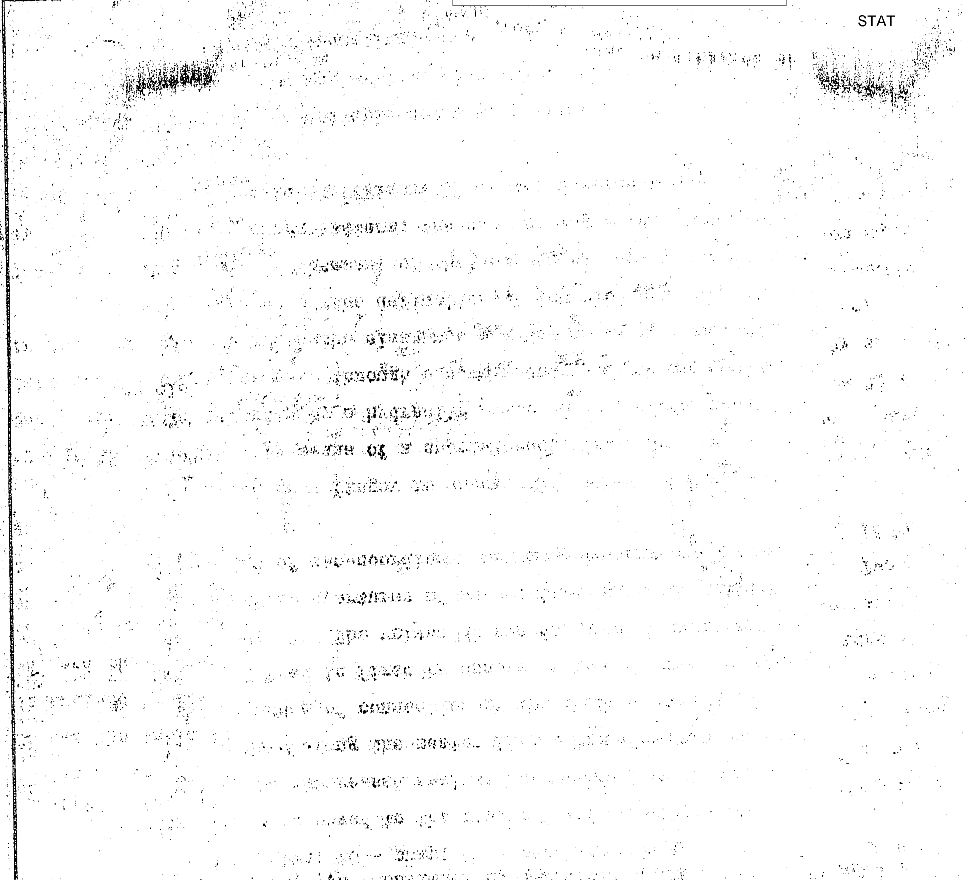

Figure 7. Diagram of the two-coordinate tracking system of the Odessa Milling Machine Factory: 1,2 - cylinders; 3,4 - pumps; 5,6 - controlling slide valves; 7 - finger; 8 - master form; 9 - bar; 10 - ring; 11,12 - slide valves; 13 - damping apertures; 14 - cavities of controlling slide valves; 15 - springs; 16 - plungers; 17 - spring blocks; 18 - valves; 19 - overflow.

PRESSURE IN THE WORKING CAVITIES OF A CYLINDER

With a differential cylinder (Figure 8a), the pressure in the rod cavity (cylinder cavity with a relatively small area F_2) is fixed as equal to the pressure maintained by the valve p_{val}.

The condition of equilibrium is used as the basis for determining the pressure in a rodless cavity. Assuming the left to right direction as positive and projecting all the forces to this direction, we obtain

$$p_1 = p_{val} \frac{F_2}{F_1} - \frac{R}{F_1},$$

where F_1, F_2 (cm^2) is the large and small area of the cylinder of the performing member;

R (kg) is the load.

The numerical value of R must allow for the magnitude and direction of the cutting forces, weight (in the case of vertical movements), and the forces of friction.

The condition $F_1 = 2 F_2$ is generally observed to obtain identical forces in the case of movements in both directions in differential cylinders. In this case, the pressure in the cavity 1 is determined by the equation

$$p_1 = 0.5p_{val} - \frac{R}{F_1}. \qquad (1)$$

The pressures in the working and overflow cavities of a simple cylinder (Figure 8b) (p_{rab}, p_{obr}) are determined from the following conditions:

a) the working areas of the cylinder F_1, F_2 from the direction of injection and overflow equal each other;

b) with the slide valve in the central position, the end play δ_x along all the edges are equal; the magnitude a may be both positive and negative;

c) the side play δ_r must be constant the entire length of the slide valve, and all the edges of the slide valve and the body are geometrically equal.

Oil from the pump is fed to circulation chamber A, whence it passes through apertures 2 and 1 into circulation chamber B and through apertures 3 and 4 into circulation chamber C. Circulation chambers B and C are connected to the overflow.

On the whole, the pressure drop Δ_p, discharge Q, and the dimensions of the apertures of the slide valves are related by the function

$$\Delta p = c\frac{Q^2}{x^2},$$

where c is the coefficient ~~of~~ the magnitude of which is determined by the discharge factor $\mu(\mu_1)$, the passage area f and the relationships between the two types of play δ_x and δ_r.

Substituting c, Q, and δ_x in the function written with the ordinal index of the aperture and expressing the pressure drop in each aperture by p_{val}, p_{rab}, p_{obr}, we obtain

$$p_{obr} = \frac{c_1(Q_c + q_2)^2}{(a + x_{zol})^2};$$

$$p_{val} - p_{obr} = \frac{c_2 q_2^2}{(a - x_{zol})^2};$$

$$p_{val} - p_{rab} = \frac{c_3(Q_c + q_4)^2}{(a + x_3)^2};$$

$$p_{rab} = \frac{c_4 q_4^2}{(a - x_3)^2},$$

where Q_c (cm^3/sec) is the discharge of the cylinder: $Q_c = F_v + q_c$;

q_2, q_4 (cm^3/sec) is the overflow through the apertures 2,4;

x_{zol} (cm) is the displacement of the slide valve from the central position;

v (cm/sec) is the speed of movement of the performing member;

q_c (cm^3/sec) is the overflow from one cylinder cavity into the other.

Preliminarily assuming the pressure drops in apertures 2,4, as well as in apertures 1,3, to be correspondingly equal, and taking into account the equality of the passage sections of the apertures 2,4 and 1,3, we have

$$c_2 = c_4; \; q_2 = q_4; \; c_1 = c_3.$$

From the condition of equilibrium and from the functions cited above follows:

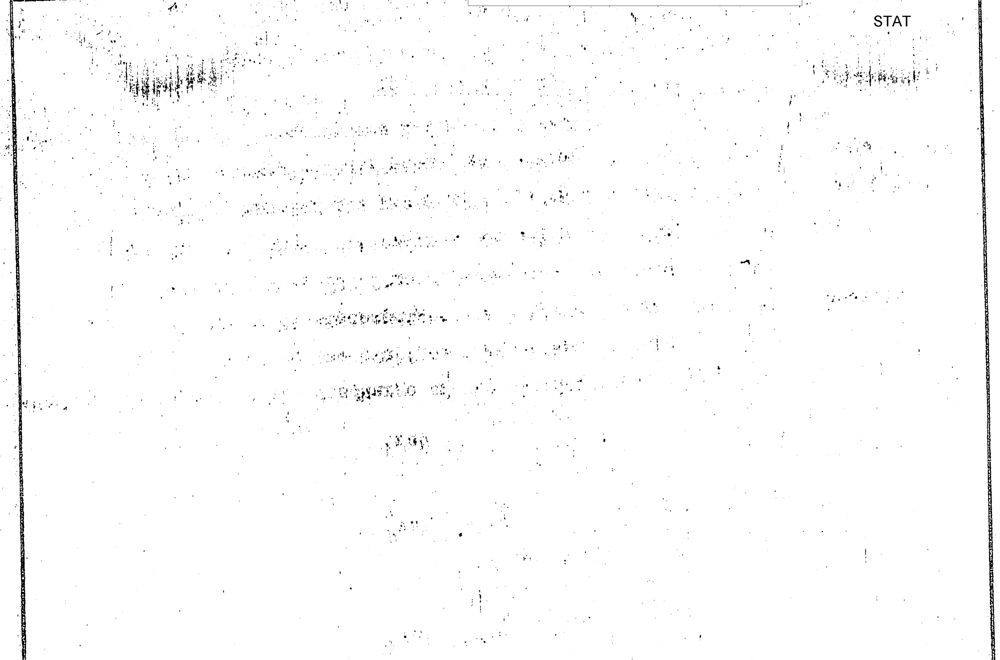

Figure 8. Diagram of oil circulation through controlling slide valves: a. in a differential cylinder; b - in a simple cylinder; c - $p_{val}F_2$; d - p_{val}; e - feed; f - $Q_c + q_1$; g - overflow; h - q_c; i - p_{rab}; j - p_{obr}; k - $x = a + x_{zol}$; l - $\delta_x = a - x_{zol}$; n - $Q_c + q_4$; o - $Q_c + q_2$; p - p_{val}; q - feed; r - overflow.

$$p_{rab} = 0.5\left(p_{val} - \frac{R}{F_1}\right); \quad p_{obr} = 0.5\left(p_{val} + \frac{R}{F_1}\right). \tag{2}$$

In apertures 1, 2, 3, and 4, the following drops take place:

$$\Delta p_1 = 0.5\left(p_{val} + \frac{R}{F}\right); \quad \Delta p_2 = 0.5\left(p_{val} - \frac{R}{F}\right);$$

$$\Delta p_3 = 0.5\left(p_{val} + \frac{R}{F}\right); \quad \Delta p_4 = 0.5\left(p_{val} - \frac{R}{F}\right).$$

Hence the preliminary condition of equality of the drops in the apertures 2, 4 and 1, 3 is maintained and formulas (2) are correct.

DISCHARGE THROUGH THE APERTURES OF THE CONTROLLING SLIDE VALVES

The apertures of the controlling slide valves are conditionally divided into open 1, 3 and closed 2, 4 (Figure 8b). The discharge Q in cm^3/sec through an open aperture may be determined by writing the Bernoulli theorem for sections I, II, III:

$$\frac{p_I}{\gamma} + \frac{v_I^2}{2g} = \frac{p_{II}}{\gamma} + \frac{v_{II}^2}{2g} + h_{I,\,II_{tr}} = \frac{p_{III}}{\gamma} + \frac{v_{III}^2}{2g} + h_{I,\,II_{tr}} + h_{II,\,III_{tr}} + h_{ud},$$

where v_I, v_{II}, v_{III}, p_I, p_{II}, p_{III} are the speeds (cm/sec) and pressures (kg/cm^2) in sections I, II, III;

$h_{I,\,II_{tr}}$ is the loss of head (cm) due to friction as the oil flows between sections I and II;

h_{ud}, h_{II}; III_{tr} is the loss due to percussion and friction (cm) as the oil flows between sections II and III;

γ is the volumetric weight of the oil; $\gamma \approx 0.0009$ kg/cm^3;

g is the acceleration of the force of gravity; $g = 981$ cm/sec^2.

The speeds in sections I and III are small, and for this reason it may be assumed that

$$v_I \approx 0;\ v_{III} \approx 0;\ h_{ud} \neq h_{II},\ III_{tr} \approx \frac{v_{II}^2}{2g}.$$

The loss due to friction is overcome by a velocity head, allowance being made for it through the last equation.

The loss h_I, II_{tr} may be expressed by the velocity and coefficient of contraction:

$$h_I, II_{tr} = \xi \frac{v_{II}^2}{2g}.$$

The equations written above give:

$$v_{II} = \frac{1}{\sqrt{1 + \xi}} \sqrt{\frac{2g}{\gamma}(p_I - p_{II})}.$$

The discharge through the open aperture of the slide valve is

$$Q = v_{II} \varepsilon f = \mu f \sqrt{\frac{2g}{\gamma}(p_I - p_{II})} = \mu f \sqrt{\frac{2g}{\gamma} p}, \qquad (3)$$

$$\mu = \sqrt{\frac{\varepsilon}{1 + \varepsilon}},$$

where ε is the coefficient of compression of the flow of oil upon passage through the aperture of the slide valve;

μ is the coefficient of discharge;

p is the pressure drop in sections I, III:

$$p = p_I = p_{III} \approx p_I - p_{II}.$$

The equality of pressure of p_{II} and p_{III} has been tested experimentally with manometers. With a discharge of up to 1000 cm^3/sec, pressure p_I of up to 30 kg/cm^2, and slide valve diameter of 16 mm, the difference in the pressures p_{II} and p_{III} did not exceed 0.5 kg/cm^2.

Finding the magnitude of the coefficient of discharge μ is a difficult matter. The passage area f of the aperture can be calculated only in approximation, since the gaps in the aperture depend on the axial displacement of the edges δ_x, side play δ_p, and obstruction of the edges and the parallelism of the latter. But exact measurement of all the magnitudes mentioned is impossible.

It is more advisable not to determine the coefficient of discharge μ on the slide valves, but on a special device, a diagram of which is shown in Figure 9a.

The play δ is small in relation to the diameter of the slide valve d_3, $\frac{\delta}{d_3} < 0.005$, hence the aperture of the controlling slide valve (up to section II,

Figure 8b) may be considered a narrow flat aperture with the profile shown in Figure 9b and having the length "b" equalling the developed length of the edge of the slide valve, an aperture difficult of measurement thus being replaced by one which permits exact measurement.

The coefficient of discharge is determined with play δ varying from 0.007 to 0.035 cm, viscosity from 0.17 to 0.75 cm^2/sec (2.6 to 10° E) and head of up to 250cm, with the Reynolds indices (Re), which vary from 7 to 145. The Reynolds index was computed from the formula $Re = \frac{2v\delta}{d}$, where v (cm/sec) is the average velocity of the oil in the aperture.

The magnitude of the coefficient of discharge μ lies between the limits 0.53 and 0.76. Within these limits, higher values of the coefficient of discharge pertain to less viscous oil and to greater head.

The average value of μ subsequently adopted was 0.6.

The passage area of the aperture of the slide valve (Figure 10) is

$$f = \delta b.$$

In determining the play δ it is necessary to allow for side play δ_p, end play δ_x, and a certain obstruction of the edges of the slide valve and body. It is simplest to find the play δ graphically on a large scale, e.g., 1000:1 (Figure 10a). Measurement under a microscope has shown that the obstructions of the edges of the tracking slide valves have the form of a cone having a length of ≈ 0.01 and an angle $\approx 20^\circ$.

Figure 9. Coefficient of discharge of open apertures of controlling slide valves: a - diagram of device for finding coefficient of discharge; b - form of apertures.

From the Bernoulli equations written in sequence for sections III, IV, V, VI (Figure 8b), assuming the coefficient of compression ε of the stream upon entry into an overlapped aperture to equal 1 (which is permissible, since the obstruction of the edges is commensurable with the play), derives the initial equation for finding the discharge through overlapped apertures

$$\frac{p_{III}}{\gamma} - \frac{p_{IV}}{\gamma} = \frac{p}{\gamma} = \frac{1}{\mu^2}\frac{Q^2}{f^2 2g} + h_{IV,Vtr}.$$

The loss due to friction h_1 and v_{tr} in the zone of flow between sections IV and V is found by use of the well-known function for laminar motion:

$$h_{IV,Vtr} = \frac{12Q\nu l}{f\,\delta^2 g} = \frac{48}{Re}\,\frac{l}{\delta}\,\frac{v^2_{IV}}{2g},$$

where l is the length of the overlap (Figure 10b).

Then the discharge through an overlapped aperture is:

$$Q = \mu_1 \sqrt{\frac{2g}{\gamma}\Delta p}, \qquad (4)$$

where

$$\mu_1 = \frac{\mu}{1 + \frac{48}{Re}\mu^2\frac{l}{\delta}}.$$

Figure 12 gives the values of μ_1 for the coefficient adopted, $\mu = 0.6$.

Figure 10. Passage area of a slide valve: a. play with open aperture; b - play with overlapped aperture.

Formulas (3) and (4) require verification because of the allowances made when they are evolved. Experimental data may be utilized for such verification.

Figure 11a shows the passage area of an aperture f (the obstruction of the edges noted above being taken into account), and Figure 11b the discharge of oil through the aperture, computed by use of formulas (3) and (4) for the conditions of the experiment; the experimental points are given there also.

The discharge of mineral oil through narrow apertures decreases with the lapse of time. However, for single-coordinate tracking systems this decrease in discharge should be ignored, since it is a comparatively slow process and the controlling slide valves are constantly displaced when in operation and restore the maximum discharges.

In the literature one encounters incorrect statements concerning finding of the discharge through the apertures of controlling slide valves.

1. It is proposed that losses of head and the discharge in the apertures of slide valves be found as in local resistance, on analogy with the rectangular gates in pipes. The coefficients of resistance in both instances are levelled when the relative (with respect to the maximum passage area) magnitudes of the passage areas are equal.

The analogy with the gate is unfounded: the maximum passage section of a gate has an approximately square shape, while the maximum passage section of the aperture of a slide valve has a length b more than 100 times greater than its width δ.

In the calculation of losses of head in slide valves, the coefficients of resistance pertain to the velocity in the aperture, while the numerical values of these coefficients are calculated in experiments with gates for the velocity which has been established for the gate. The error in these calculations, "on analogy", is many times greater than the magnitude sought.

Figure 11. Determination of discharge through the apertures of controlling slide valves: a - passage section of the apertures, $\delta_p = 00.015$ cm, $b = 2.75$ cm; b - discharge under equal pressures through the apertures, $v = 0.32 \frac{cm^2}{sec}$; c - passage section of aperture; d - axial displacement of edges; e - Q cm^3/sec; f - discharge through aperture; g - $p = 13$ kg/cm^2; h - $p = 10$ kg/cm^2; i - $p = 7$ kg/cm^2; j - axial displacement of edges.

2. The discharge through overlapped apertures is determined without allowance being made for the velocity head, and this leads to inaccurate results when the overlap is small in length. Instead of a correct calculation of the discharge being

made in accordance with the laws of hydraulics, empirical formulas are introduced solely for the conditions approximating the conditions of the experiment.

3. It is stated that in the open apertures of slide valves, depending on the opening (axial displacement of the edges), either the Reynolds indices or the turbulent flow may be the "disrupted laminar flow". The critical magnitudes characterizing the transition to a turbulent flow are an opening of approximately 0.15 mm or a Reynolds index number of 260.

The pressure losses with a disrupted laminar flow are proportionate to the discharge to a degree greater than one and smaller than 2, and with a turbulent flow, they are proportionate approximately to the second degree of the discharge.

Figure 12. Coefficient of discharge of overlapped apertures. The numerals on the curves correspond to the ratio of the length of the aperture (along the flow) to the side play: 1 - Coefficient of discharge; 2 - Reynolds number.

It must be remembered that in a contracting flow, such as is found at the entrance to the open apertures of slide valves, the losses due to friction are small. On the whole, static pressure is consumed not in losses due to friction, but is transformed into velocity head, which is subsequently lost in "percussion" as the velocity of the flow falls to the velocity of the oil in the return pipes. Hence the pressure drop in the apertures of the slide valves, independently of the Reynolds number (at least within certain oil viscosity limits), must be proportionate to the discharge squared; this has been confirmed through experiments (Figure 9b).

STATIC CHARACTERISTICS

The deflections of the clearance gage with the different velocities and loads of the performing mechanism at the moments of equilibrium of the active forces constitute the static characteristics of a tracking system.

The clearance gage deflects the slide valve through leverage (Figure 2).

The deflection of the clearance gage (x) and the slide valve (x_{zol}) are connected by the function

$$x_{zol} = ix,$$

where i is the coefficient of transmission from the clearance gage to the slide valve.

The deflection of the slide valve, and consequently of the clearance gage also, with predetermined velocity and load of the performing mechanism, are found as follows.

1. First of all, a schematic drawing is made of the slide valve in the displaced position and of the cylinder, and the directions of movement of the STAT performing member and active forces are indicated (Figure 8).

The discharges through the apertures of the slide valve, with different end plays δ_x and different pressure drops in the apertures, are computed by use of formulas (3) and (4).

2. The pressure in the cylinder cavities and the pressure drops in the apertures of the slide valve are computed by use of formula (1) in the case of a differential cylinder, or by use of formula (2) in the case of a simple cylinder, in accordance with the assigned load R, cylinder area F, and the pressure in the system p_{val}.

Figure 13. Static characteristics of a tracking system with a simple cylinder and four-edged slide valve; a - graph for finding the displacement of the slide valve, with predetermined velocity and motive force of the performing member; b - discharge through the apertures of the slide valve; curve 1 corresponds to a pressure drop of 16 kg/cm^2; curve 2 -- 14; curve 3 -- 12; curve 4 -- 10; curve 5 -- 8; curve 6 -- 6; curve 7 -- 4 kg/cm^2; c - discharge; d - (discharge in the cylinder); e - x_{zol}; f - displacement of body of controlling slide valve; g - x_{zol} displacement; h - discharge; i - Qcm^3/sec; j - end play δ_x, cm.

3. A curve $Q_c + q_1 x_{zol}$ is constructed on the basis of the graph (Figure 13b) for the aperture through which the oil is fed into the cylinder (Figure 13a, curve A).

Q_c is the discharge of the cylinder, q - the leakage of oil through the aperture adjacent to the feed aperture, x_{zol} - the deflection of the body of the controlling slide valve with respect to the slide valve.

The discharge of the cylinder is entered on the graph (line B), and the leakages q (curve C) are found from the graph (Figure 13b) and plotted above the discharge line.

The magnitude of the deflection of the body of the controlling slide valve which insures displacement of the performing member at the predetermined velocity, under the action of the predetermined forces of resistance, is determined by the point of intersection M of the curve of discharge through the feed aperture and the curve of the sum of the discharge of the cylinder and the leakages.

In figure 14 are given the static characteristics of a tracking system in which the diameter of the slide valve $d_{zol} = 3.0$ cm; side play $\delta_p = 0.0015$ cm; overlap in the central position of the slide valve $a = 0$; area of the cylinder $F_1 = F_2 = 200$ cm^2; oil viscosity $\nu = 0.4$ cm^2/sec; load R, up to 1200 kg.

Figures 17a and 18a show the curves v, x_{zol}, R = const for a system with a differential cylinder ($F_1 = 200$ cm^2, $F_2 = 100$ cm^2) with double-edged and single-edged slide valves having a diameter of $d_{zol} = 3.0$ cm at a pressure in the system of $p_{val} = 20$ kg/cm^2.

In order to maintain identical motive forces in the case of equal external diameters of the cylinders, the pressure adopted in the system with a differential cylinder is twice as great as with a simple cylinder.

Figure 14. Velocity of the performing member in relation to displacement of the slide valve, with motive forces as follows:
curve 1 corresponds to 0; curve 2, to 400 kg; curve 3, to 80 kg; and curve 4, to 1200 kg.
a - v, cm/sec; b - $-x_{zol}$.

Figure 15. Motive force in relation to displacement of the slide valve, at the following velocities:
curve 1 corresponds to 0; curve 2, to 0.08 cm/sec; curve 3, to 0.166 cm/sec; curve 4, to 0.33 cm/sec; curve 5, to 0.5 cm/sec; and curve 6, to 0.66 cm/sec; a - P, kg.

In constructing the static characteristics, the pressure losses Δp_1, Δp_2 in the pipes feeding oil into the cylinder and carrying oil from it are allowed for in the form of an additional load R_{hydr}, computed by use of the formula

$$R_{hydr} = p_1F_1 \pm p_2F_2.$$

A deflection of the clearance gage is usually identified with an error in copying, for which reason an effort is made to have the deflections of the clearance gage as small as possible in tracking systems. However, in machining by the copying method, the deviations of the clearance gage are partially compensated for by means of the mutual displacement of the calibration instrument and the part being machined (Figure 19), when the error does not exceed one-third of the deviation of the clearance gage.

Figure 16. Motive force in relation to velocity, upon displacement of the slide valve as follows:
curve 1 corresponds to 0.001 cm; curve 2, to 0.002; curve 3, to 0.003; curve 4, to 0.004; curve 5, to 0.005; curve 6, to 0.006; and curve 7, to 0.007 cm.
a - P, kg; b - v, cm/sec.

The copying error in existing machine tools is not less than 0.01 mm, in planishing passes, for which reason the deflection of the clearance gage at the maximum velocities and forces must be less than 0.06 ÷ 0.08 mm.

Table 1

Parameters of comparison	Simple cylinder with four-edged slide valve (Fig. 1a)	Differential cylinder with two-edged slide valve (Fig. 2)	Differential cylinder with single-edged slide valve (Fig. 3)
Relationships $\frac{p_{val}F}{P_{max}}$	1.8	1.8	1.8
Maximum deflection of clearance gage x, with P_{max}, v_{max}, in mm	0.124	0.094	0.115
Maximum power of tracking system, $P_{max} \cdot v_{max}$, in kg . cm/sec	800	800	800
Pump discharge Q, in cm^3/sec	167	103	385
Required power Qp_{val}, in kg . cm/sec	1670	2060	7710

Table 1 gives comparative data (without allowance for oil resilience) of the different systems of tracking devices, found by constructing the static characteristics.

It will be seen from Table 1 that, the conditions being equal, the tracking systems with a differential cylinder and double-edged slide valve and the non-tracking systems with a simple cylinder and four-edged slide valve have the least deflection of the clearance gage.

It must be pointed out that comparison of tracking systems from the aspect of deflection of the clearance gage is purely arbitrary, since it is possible to change the amount of deflection of the clearance gage within the limits of one system by selection of the area of the cylinder and the pressure. Comparison indicates that in the case of the system with a differential cylinder and single-edged slide valve, the required power greatly exceeds the power required for the other two systems.

STABILITY OF THE TRACKING SYSTEM

Aperiodically stable, oscillatorily stable, and non-stable tracking systems are distinguished according to the behavior of the systems when brought out of a condition of equilibrium (upon cessation of action of the disturbing forces). The aperiodically stable systems return to a position of equilibrium, after completing one or several oscillations; the oscillatorily stable systems, after completing an infinite number of attenuating oscillations; the non-stable systems do not return to a position of equilibrium.

We shall explain the conditions of aperiodic stability of hydraulic tracking systems by using Figure 1, in which the clearance gage and slide valve are made as a unit, this simplifying somewhat solution of the problem. The movement of the performing member of the tracking system may be represented in the form of the sum of the transfer movement, together with the clearance gage 6, which is in contact with the master form 2, and of the movement of the performing member 4 with respect to the clearance gage 6 (Figure 1).

We shall designate the velocity of displacement of the performing member 4 by v, the velocity of displacement of the clearance gage by v_{cop}, and the amount of displacement of the performing member in relative movement by x. The coordinate axis x is connected to the clearance gage. The displacement x is counted off from the position occupied by the performing member at the moment of equilibrium of the system at the assigned velocity v and motive force P.

The basic equation of movement of the performing member is:

$$-mv' + P + R = -mx'' - mv'_{cop} + P - R = 0, \qquad (5)$$

where m (kg $\cdot$ sec^2/cm) is the mass of the performing member;

P (kg) is the motive force;

R (kg) is the load -- $R = R_{cut} + R_{fr}$;

R_{cut} (kg) is the cutting load;

R_{fr} (kg) is the force of friction;

x'' is the acceleration in relative movement;

v'_{cop} is the acceleration in transfer movement;

v' is the acceleration of the performing member.

We shall designate by P_0, R_0 the numerical values of the motive force and load of the performing member at the moment of equilibrium of the system (when $x'' = 0$; $v' = 0$).

From equation (5) and the condition of equilibrium of the forces acting on the performing member, we have

$$-mx'' - mv''_{cop} + (P - P_0) + (R' - R_0) = 0. \qquad (6)$$

In the oscillation process around the position of equilibrium

$$P - P_0 = \frac{gP}{dx} + \frac{gP}{dv}\Delta v; \quad R - R_0 = \frac{dR}{dx}x + \frac{dP}{dv}\Delta v, \qquad (7)$$

where Δv is the velocity increment of the performing member with respect to the velocity at the initial moment in time.

It is obvious that $\Delta v = \Delta v_{cop} + x'$,

where x' is the velocity in relative movement,

Δv_{cop} is the copying velocity increment, which is related to the profile of the master form, the shape of the clearance gage, and the driving motion velocity.

Considering the process in the copying of smooth curves to be a brief interval of time, it may be assumed that

$$\Delta v_{cop} \approx 0; \quad v'_{cop} \approx 0.$$

We shall introduce the dimensionless coordinate σ:

$$\sigma = \frac{x}{x_{max}}. \qquad (8)$$

Figure 17. Static characteristics of a tracking system with a differential cylinder and double-edged slide valve, with $p_{val} = 20 \frac{kg}{cm^2}$; $F_1 = 200cm^2$; $F_2 = 100cm^2$; $d_{zol} = 3.0$ cm:

a - velocity of performing member in relation to displacement of the slide valve with these motive forces: curve 1 corresponds to 0; curve 2, to 400; curve 3, to 800; and curve 4, to 1200 kg; b - motive force in relation to displacement of the slide valve at these velocities: curve 1 corresponds to 0; curve 2, to 0.008; curve 3, to 0.166; curve 4, to 0.33; curve 5, to 0.5; and curve 6, to 0.66 cm/sec; c - motive force in relation to the velocity, with displacement of the slide valve as follows: curve 1 corresponds to 0.001 cm; curve 2, to 0.002; curve 3, to 0.003; curve 4, to 0.004; curve 5, to 0.005; and curve 6, to 0.006 cm.

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d - P, in kg; e - v, in cm/sec; f - $-x_{zol}$; g - P, in kg; h - $-x_{zol}$; i - v, in cm/sec.

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x_{max} is the deflection of the performing member in relation to the clearance gage (Figure 1b) at the maximum velocity v_{max} and motive force P_{max}

$$x' = \sigma' x_{max}; \quad x'' = \sigma'' x_{max}.$$

From equations (6), (7), (8), we have:

$$-m\sigma'' + \left(\frac{dP}{dv} + \frac{dR}{dv}\right)\sigma' + \left(\frac{dP}{dx} + \frac{dR}{dx}\right)\sigma = 0. \qquad (9)$$

The derivative $\frac{dP}{dx_{zol}}$ is found from the static characteristics of the tracking system (Figures 15d; 17b; 18b) or is computed analytically.

Figure 18. Static characteristics of a tracking system with a differential cylinder and single-edged slide valve: $p_{val} = 20\ \frac{kg}{cm^2}$; $F_1 = 200 cm^2$; $F_2 = 100 cm^2$; $d_{zol} = 3.0 cm$:

a - velocity of performing member in relation to displacement of the slide valve with motive forces as follows: curve 1 corresponds to 0; curve 2, to 400; curve 3, to 800; curve 4, to 1200; curve 5, to minus 400; curve 6, to minus 800; and curve 7, to minus 1200 kg;

b - motive force in relation to displacement of the slide valve at these velocities: curve 1 corresponds to 0; curve 2, to 0.2; curve 3, to 0.4; curve 4, to 0.6; curve 5, to minus 0.2; curve 6, to minus 0.4; and curve 7, to minus 0.8 cm/sec;

c - motive force in relation to the velocities with displacement of the slide valve as follows: curve 1 corresponds to minus 0.002; curve 2, to minus 0.004; curve 3, to minus 0.006; curve 4, to plus 0.002; curve 5, to plus 0.004; curve 6, to plus 0.006; and curve 7, to plus 0.008cm.
d - v, in cm/sec; e - -x, displacement of edges of slide valve with the performing member not loaded (at rest); f - x, in cm; g - P, in kg; h - $-x_{zol}$; i - x_{zol}, in cm; j - P, in kg; k - v, in cm/sec.

In a tracking system with a simple cylinder (Figure 8b), we shall consider an instance in which the slide valve is displaced to the extent that its edges overlap apertures 2 and 4.

The discharges q, $Q_c + q$ through apertures 3 and 4 of the slide valve are found by use of formulas (1), (2), (3), and (4):

$$q = \mu_1 f_4 \sqrt{\frac{2g}{\gamma}} \sqrt{p_{rab}} = \mu_1 f_4 \sqrt{\frac{2g}{\gamma}} \sqrt{0.5(p_{val} + \frac{P}{F})};$$

$$Q_c + q = \mu f_3 \sqrt{\frac{2g}{\gamma}} \sqrt{0.5(p_{val} - \frac{P}{F_1})}.$$

Figure 19. Copying error in the case of a spherical support of the clearance gage lever:
1 - calibration instrument; 2 - article; a - displacement of calibration instrument with respect to the article, with the cutting tool and clearance gage on a common perpendicular to the longitudinal axis; b - error.

Ignoring the leakages in the cylinder, we have:

$$Q_c = F_1 v; \quad F v + \mu_1 f_4 \sqrt{\frac{2g}{\gamma}} \sqrt{0.5(p_{val} \frac{P}{F_1} -)} =$$

$$= \mu f_3 \sqrt{\frac{2g}{\gamma}} \sqrt{0.5(p_{val} - \frac{P}{F_1})}; \qquad (10)$$

$$\frac{dP}{dx} = 4F_1 \frac{\sqrt{0.5(p_{val} - \frac{P}{F_1})}\,\frac{d\mu f_3}{dx} - \sqrt{0.5(p_{val} + \frac{P}{F_1})}\,\frac{d\mu_1 f_4}{dx}}{\frac{\mu_1 f_4}{\sqrt{0.5(p_{val} + \frac{P}{F_1})}} + \frac{\mu f_3}{\sqrt{0.5(p_{val} - \frac{P}{F_1})}}}.$$

Ignoring the relatively small magnitudes, after algebraic conversions we obtain

$$\frac{dR}{dx} \approx -\frac{P}{x\varphi},$$

where

$$\varphi = \frac{\left(1 + \frac{a}{x_{zol}}\right)}{2(\alpha - 1)\left[1 + \beta\frac{x_{zol} + a}{2(x_{zol} - a)}\right]};$$

the minus sign allows for the opposite directions of the increments dP and dx;

$\alpha = \frac{p_{val}F}{P}$ characterizes the load of the system;

$\beta = \frac{q}{Q_c}$ characterizes the leakages in the slide valve.

With an overlap of $a = 0$; $\varphi = \frac{1}{2(\alpha - 1)(1 + 0.5\beta)}$.

The leakage factor β is found from the equation

$$= \frac{q}{Q_c} = \frac{q}{(Q_c + q) - q} = \frac{1}{\frac{\mu f_3}{\mu_1 f_4}\sqrt{\frac{\alpha - 1}{\alpha + 1}} - 1}. \qquad (11)$$

Figure 20 (a and b) gives the values of β'' and φ computed for displacement of the slide valve from 0.004 to 0.010 cm, when the overlap $a = 0$, oil viscosity $\nu = 0.2$ cm^2/sec, and pressure $p_{val} = 10, 15, 20$ kg/cm^2.

The change in the load R in the case of oscillations of the performing member may be ignored, assuming that $\frac{dR}{dx} = 0$.

We shall introduce into equation (9) the coefficients of self-regulation θ_1 and θ_2:

$$\theta_1 = -\frac{v_{max}}{P_{max}}\frac{dP}{dv}; \quad \theta_2 = -\frac{v_{max}}{P_{max}}\frac{dR}{dv}.$$

The derivative $\frac{dP}{dv}$ may be found from the static characteristics (Figures 16; 17c; 18c):

$$\sigma'' + \frac{P_{max}}{mv_{max}}(\theta_1 + \theta_2)\sigma' + \frac{P_{max}}{x_{max}m}\sigma = 0. \qquad (12)$$

The solution to equation (12) has this form:

$$\sigma = c_1 e^{k_1 t} + c_2 e^{k_2 t},$$

where k_1, k_2 are found by use of equation (13);

t (sec) is the time from the beginning of the process;

c_1, c_2 are the integration constants in accordance with the initial conditions;

$$k_{1,2} = \frac{P_{max}(\theta_1 + \theta_2)}{2mv_{max}} + \sqrt{\left(\frac{P_{max}(\theta_1 + \theta_2)}{2mv_{max}}\right) - \frac{P_{max}}{\varphi x_{max}m}}. \qquad (13)$$

Aperiodic stability is provided for by the inequality

$$x_{max} \gg \frac{m\frac{v_{max}^2}{2}}{P_{max}}\frac{8}{\varphi(\theta_1 + \theta_2)}; \qquad (14)$$

$$\frac{m\frac{v_{max}}{2}}{p_{val}F} \ll \frac{x_{max}\varphi(\theta_1 + \theta_2)^2}{8\alpha}.$$

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Figure 20. The coefficients β, φ, θ_1 of a tracking system with a simple cylinder and four-edged slide valve having an overlap of $a = 0$ and side play of $\delta_p = 0.0015$ cm, with an oil viscosity of $\nu = 0.2 cm^2$/sec and pressures in the hydraulic system of 10, 15, and 20 kg/cm^2 in relation to the displacement of the slide valve:

Curves 1 correspond to a motive force of $P = \frac{p_{val}F}{1.5}$; curves 1, to $P = \frac{p_{val}F}{2}$; curves 3, to $P = \frac{p_{val}F}{3}$; and curves 4, to $P = \frac{p_{val}F}{4}$.

$d - = \frac{q}{Q_c}$; $e - p_{val} = 10 kg/cm^2$; $f - p_{val} = 15 kg/cm^2$; $g - p_{val} = 20 kg/cm^2$; $h - x_{zol}$; $i - p_{val} = 10 kg/cm^2$; $k - p_{val} = 15 kg/cm^2$; $l - p_{val} = 20 kg/cm^2$; $m - \theta_1 = - \frac{V}{P} \frac{P}{V}$; $n - p_{val} = 10 kg/cm^2$; $o - p_{val} = 15 kg/cm^2$; $p - p_{val} = 20 kg/cm^2$

We shall explain analytically the value of the coefficients θ_1, θ_2. STAT

Differentiating expression (10) by v, we obtain the equation which determines the derivative $\frac{P}{v}$ in relation to the load P, velocity v, and the parameters of the system:

$$\frac{dP}{dv} = -\frac{(p_{val}F)^2 - P^2}{q(p_{val}F - P) + (Q + q)(p_{val}F + P)} \cdot 2F = \frac{-2(\alpha^2 - 1)}{\alpha(2\beta + 1) + 1} \cdot \frac{P}{v}.$$

When finding the stability at each point, it may be assumed that $P_{max} = P$ and that $v_{max} = v$. Hence we have

$$\theta_1 = \frac{2(\alpha^2 - 1)}{\alpha(2\beta + 1) + 1}.$$

The values of θ_1 are given in Figure 20c.

$$\frac{dR}{dv} = \frac{dR_{cut}}{dv} + \frac{dR_{fr}}{dv}; \quad \frac{dR_{cut}}{dv} = 0;$$

$$\frac{dR_{fr}}{dv} = -\frac{R_{fr}}{\tau}\frac{\tau}{dv}, \qquad (15)$$

where R_{fr} τ is the force of friction and the cited coefficient of friction with load and velocity of displacement of the performing member; the minus sign makes allowance for the opposite direction of the force R_{fr} and velocity v.

Figure 21. Relationship of the coefficient of friction in guides to velocity: a - v, in cm/sec.

The coefficient of friction τ depends on the specific pressure on the bearing sliding surface, grade of the lubricating oil and its temperature, the design and dimensions of the glands, and, in particular, on the accuracy of adjustment of the guides and setting of the cylinder.

Figure 21 gives the coefficients of friction in relation to the velocity v measured with a specific load on the guides of the order of 0.8 $\div$ 1.18kg/cm^2 and copious lubrication. The numerical values of the coefficient θ_2 are computed on the basis of Figure 21, the pressure losses in the pipelines being ignored (Table 2).

The pressure losses in the pipelines of the system must be allowed for in numerical computations of the derivative $\frac{dR}{dv}$ in addition to the value of $\frac{dR}{dv}$ by use of formula (15).

Table 2

Numerical values of the coefficient $\theta_2 = \frac{dv_{max}}{P_{max}} \frac{dR_{fr}}{dv}$

$\frac{R_{tr}}{R}$	v, in cm/sec							
	0.1	0.2	0.3	0.4	0.6	0.8	1.0	1.8
0.15	−0.033	−0.0435	−0.052	−0.0615	−0.07	−0.08	−0.78	−0.78
0.2	−0.044	−01058	−0.07	−0.082	−0.1	−0.108	−0.104	−0.104
0.3	−0.066	−0.087	−0.105	−0.123	−0.147	−0.162	−0.156	−0.156
0.4	−0.088	−0.116	−0.14	−0.164	−0.196	−0.216	−0.208	−0.208
0.5	−0.11	−0.145	−0.176	−0.205	−0.245	−0.27	−0.26	−0.26

After substitution in equation (14) of the values of the coefficients φ, θ_1, θ_2, and α in relation to displacement x of the clearance gage, the maximum ratio of the kinetic energy of the performing member $m\frac{v^2}{2}$ to the motive force $p_{val}F$, to which the aperiodic stability of the system is provided for, is found.

In calculation of the admissible ratio $\frac{m\frac{v^2}{2}}{p_{val}F}$, it is necessary to allow for the coefficient i of transmission from the clearance gage to the slide valve, since the coefficients φ and θ are determined in relation to the displacement of the slide valve x_{zol}, and displacement of the clearance gage x enters into formula (14).

Figure 22 gives the maximum ratios $\frac{Gv^2}{p_{val}F} = \frac{2gm\frac{v^2}{2}}{p_{val}F}$ in relation to displacement x of the clearance gage when $i = 1$, $p_{val} = 20 kg/cm^2$ and when the ratio $\frac{R_{fr}}{R} = 0.5$, without allowance for the pressure losses in the pipelines of the system (G_{kg} is the weight of the performing member).

With respect to stability, the system with a simple cylinder and four-edged slide valve is equivalent to the system with a differential cylinder and double-edged slide valve, since when the degree of the load (ratio of the maximum motive force of the system to the load) is identical, the values of the coefficients φ and θ are equal respectively for both systems.

The system with the differential cylinder and single-edged slide valve is the least stable (permitting a relatively smaller ratio $\frac{Gv^2}{p_{val}F}$, other conditions being equal), since the coefficients φ and θ_1 are smaller in magnitude in relation to the other systems, due to the greater value of the leakage factor β.

Let us consider the influence of elasticity of the oil. The minimum factor of rigidity T_1 in kg/cm of a system with a simple cylinder is $T_1 \approx \frac{4EF^2}{W}$. The minimum factor of rigidity T_2 in kg/cm of a system with a differential cylinder is

$$T_2 = \frac{EF^2}{W},$$

where E is the modulus of elasticity of the oil; $E = 17{,}000\ kg/cm^2$;

W (cm^3) is the volume of the oil in the cylinder and in the feed pipes.

When a load is momentarily thrown off or applied, attenuating free oscillations of the performing member begin which add up to the main movement of the latter. Other conditions being equal, the maximum range of free oscillations in a system with a simple cylinder is smaller (up to 4 times) than in a system with a STATer-ential cylinder.

Figure 22. Minimum displacement of the clearance gage insuring aperiodic stability of a system with double-edged and four-edged slide valves, with $l = 1$ and $p_{val} = 20$ kg/cm^2:

$$a = \frac{Gv^2}{p_{val}F};\quad b = \frac{p_{val}F}{p} = 4;\quad c = \frac{p_{val}F}{p} = 3;\quad d = \frac{p_{val}F}{p} = 1.5$$

The analysis made makes it possible to draw the following conclusions:

1. The minimum deflection of the clearance gage suitable for the load and velocity of the performing member selected must be fixed in accordance with function (14). Decrease in the deflection of the clearance gage against what has been stated must result in increase in the error in copying in consequence of transition of the tracking system from the sphere of aperiodic stability to the sphere of oscillating stability.

2. With short cylinders, when the elasticity of the oil may be ignored, the system with the simple cylinder and four-edged slide valve and the system with a differential cylinder and double-edged slide valve are equivalent. When it is necessary to allow for the elasticity of the oil (with long cylinders), preference must be given to the system with a simple cylinder.

3. The simplest system, with the differential cylinder and single-edged slide valve may be employed when:

a) increased deflection of the clearance gage is permitted due to the lesser stability of the system in comparison to the other systems;

b) excessive power loss, in comparison to the other systems, is permitted.

V. A. LESHCHENKO

THE CHARACTERISTICS OF THE TWO-COORDINATE TRACKING DRIVES OF THE KFS-3 COPYING MACHINES

Parts of large dimensions and complex form are employed at the present time in mechanical engineering. Such parts increase the durability of articles, improve their aerodynamic and acoustic qualities, reduce weight, and save metal.

The main obstacle in the path of extensive employment of large-dimension parts is their "non-technologicalness", i.e., the excessively great expenditure of labor needed for their production. These parts are usually made by hand or by marking or by master forms. For this reason, the machine tool engineers were assigned the task of creating new equipment which by mechanization and automation of the machining process would raise large-dimension parts of complex form from the non-technological to the technological category.

The tasks assigned made necessary the development of new tracking devices for copying machines. In particular, the solution of these problems was undertaken at the NIAT, where new tracking devices for copying milling machines were developed and tested.

THE REQUIREMENTS SET FOR THE TRACKING DRIVES OF MACHINES

As we know, for ideal functioning of the tracking feed drive of metal-cutting equipment, it is necessary that:

(a) the feed velocity be constant;

(b) the feed velocity vector be directed strictly at a tangent to the profile of the master form at the point to be copied;

(c) the relative positions in space of the pick-up member (copying finger) and performing member (cutting tool) be fixed;

(d) the requirements of the process (?) be observed; they must be independent of the profile of the article and not require the interference of the operator in the copying process when the tracking drive is in stable operation.

The tracking drives of existing copying machines for outlining a closed profile use as a control signal the loss of linear adjustment between the positions of the pick-up and performing members and diverge greatly from the conditions listed, which are necessary for the ideal functioning of a drive. For example, in the Model 6441A semiautomatic electric copying and milling machine, every 90° during the outlining of a profile the operator must manually shift the master and tracking feed speeds. The maximum feeding speed in copying is 300 mm/min, and the machining accuracy is $\pm$ 0.05 mm.

In the Cincinatti firm's Model 28" hydraulic copying machine, the quality of reaction of the system changes in relation to the location of the profile with respect to the direction of the feeds; in the reversing zones of the feed drives (longitudinal and traverse), the system is only slightly sensitive and produces a large copying error. In addition, copying accuracy depends on the irregularity of the load on the hydraulic cylinders and the accuracy of production of the distributing slide valves. This is a defect also inherent in the 1S-70 machines.

The tracking drives must usually have minimum clearances in the kinematic chains so that a high accuracy will be achieved. In electric drives with a screw pair this is achieved by high quality of their production; and in hydraulic drives, by the use of hydraulic cylinders of limited length. The difficulty of meeting these requirements in a machine with feed drives the movement along which amounts to several meters and the imperfection noted of the existing two-coordinate tracking drives have made it necessary to develop a new drive which would make it possible to obtain a high copying accuracy on the machine, on the condition of:

(a) long kinematic chains of the feed drives with large errors, clearances, and inertia masses;

(b) maintenance of a constant level of the feeding speed;

(c) constant copying accuracy, independently of the configuration and location of the profile on the machine;

(d) complete automation of the working cycle;

(e) high feeding speed.

PRINCIPLE OF CONSTRUCTION OF THE KFS-3 TWO-COORDINATE ELECTRIC-HYDRAULIC TRACKING DRIVE

A combination of two drives forms the basis of the two-coordinate electric-hydraulic tracking drive which has been developed: a coarse transport drive, which assigns the pick-up and performing members of the machine a feeding speed of a constant level, the direction of which is near the tangential on the section being copied as the profile is being outlined; and an accurate adjusting tracking drive which corrects the direction of the feeding speed mentioned until it coincides exactly with the tangential.

Figure 1 gives an overall view of the KFS-3 copying milling machine and Figure 2 a structural diagram of it.

The machine has been given a portal form so that articles of large dimensions may be machined on it. The bedplates (2) with the master form 3 and blank 4 secured on them are fixed, while the portal device, consisting of two pairs of cross carriages 5, 6, 7, 8, move along the bedplates. The cross carriages communicate automatic movement (respectively) along the profile of the master form and blank, to the copying instrument 17 and the milling head 15, which are secured on the upper carriage.

The lower pair of carriages 5 and 6 is set in movement by electric motors 9 and 10, and the upper pair 7 and 8, by hydraulic cylinders 13 and 14.

Control of the movement of all the carriages in the copying process is accomplished automatically from the three command units of the copying instrument. The main finger of the copying instrument (Figure 3), the diameter of which corresponds to the diameter of the cutter, during the copying process moves automatically along the profile of the master form and is constantly in contact with it, oscillating on the tapered centers about its neutral position within a range of hundredths of a millimeter.

The cutter above the bedplate of the articles repeats the movement of the axis of the copying instrument and reproduces either a copy of the master form or the article equidistant to the master form, depending on the ratio of the diameters of the main finger of the copying instrument and the cutter. The copying instrument has a turning rotor, the turning angle of the rotor setting, by means of the supplementary finger and the electric motor of the copying instrument, the angle of inclination of the tangent to the profile of the master form at the point of contact of the main finger.

Figure 1. Overall view of the KFS-3 machine.

The automatic movement of the copying instrument along the profile takes place as follows. An electric position regulator (sine potentiometer) 18 located in the copying instrument controls electric motors 9 and 10 of the lower longitudinal and cross carriages by means of a two-channel electronic amplifier 24 and a power control unit. The latter consists of two electromechanical amplifiers 23 and 24, exciter 25, and electric drive motor 26 with a velocity return coupling after machining, which is accomplished by two tachogenerators 27 and 28. Along this control channel, the heavy lower carriages, which move over long distances, are assigned the velocities v_{xe} and v_{ye} (Figure 4) in accordance with the laws of the cos and sin of the angle of inclination of the tangent to the profile of the master form at the point of contact of the main finger, the resulting velocity of which, v_{0e}, in the outlining of the profile is constant in magnitude and in direction is near the tangential.

Creation of a resulting feed velocity v_0 directed strictly at a tangent (which is necessary for accurate tracking of the profile of the master form) is insured by the automatic action of the electro-hydraulic adjustment tracking drive. The performing hydraulic cylinders of this drive are automatically assigned such velocities, v_{xg} and v_{yg}, that their resulting velocity, v_{0g}, being added geometrically to the velocity v_{0e}, gives the velocity v_0 at any point of the profile of the master form.

Control of the electro-hydraulic adjustment drive is accomplished along two channels from the copying instrument: along the control channel, by the magnitude and sign of the resulting velocity from the hydraulic cylinders, and along the velocity distribution channel between the two hydraulic cylinders.

Control of the command slide valve of the hydraulic system 39 is exercised along the first channel, from the main finger of the copying instrument, by means of an inductive emitter 20, an electronic amplifier 37, and a transmitter of commands to the hydraulic system 38 to which is communicated the displacement from the neutral position corresponding to the deflection of the main finger of the copying instrument from the neutral. The feeding of oil into the performing hydraulic cyl-

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Figure 2. Structural diagram of the tracking drive of the KFS-3 machine:

1 - bed; 2 - bedplates; 3 - master forms; 4 - blank; 5, 6, 7, and 8 - cross carriages; 9 and 10 - electric feed motors; 11-cross feed screw; 12 - longitudinal feed screw; 13 and 14 - hydraulic cylinders; 15 - milling head; 16 - milling cutter; 17 - copying instrument; 18 - electric position regulator; 19 - selsyn transmitter; 20 - inductive transmitter; 21 - main copying finger; 22 - electronic amplifier; 23 and 24 - electro-mechanical amplifiers; 25 - exciter; 26 - electric motor; 27 and 28 - tachogenerators; 29 - selsyn receiver; 34 - cam of hydraulic position regulator; 35 and 36 - distributing slide valves of the hydraulic position regulator; 37 - electronic amplifier; 38 - emitter of commands to the hydraulic system; 39 - command slide valve of the hydraulic system; 40 - electric motor of the copying instrument; 41 - path return coupling; 42 - velocity return coupling; 43 - hydraulic pump.

inders of the upper pair of carriages changes in amount and direction in relation to the amount and direction of displacement of the command slide valve 39. This determines the magnitude and direction (within the profile or outside along the perpendicular to the profile) of the resulting velocity v_{0g}.

Figure 3. Situation of fingers of copying instrument and milling cutter of the KFS-3 machine with respect to the master form and article:

1 - direction of feed; 2 - master form; 3 - tangent at point 0; 4 - main copying finger; 5 - auxiliary finger; 6 - rigid connection; 7 - blank; 8 - allowance; 9 - milling cutter.

Figure 4. Diagram of the velocities of the transport and adjustment feeds: 1 - tangent at point 0; 2 - v_{ye} ($v_{adj}2$); 3 - v_{0g} (v_{adj}); 4 - v_{xg}; (v_{adj}); 5 - v_{0e} (v_{feed}); 6 - v_{xe} (v_{feed1}).

Along the second channel control is exercised by the hydraulic position regulator 33 from an auxiliary finger of the copying instrument by means of a selsyn-synchronous tracking system composed of a selsyn-transmitter 19, a selsyn receiver 29, an electronic amplifier 20, an electromechanical amplifier 31, and an electric drive motor 32. The turning angle of the cam 34 of the hydraulic position regulator is coordinated with the turning angle of the rotor of the copying instrument. As a result of this, the two distributing slide valves 35 and 36 of the hydraulic position regulator are communicated the deflections from the neutral position, which are proportionate to the sin and cos of the angle of inclination of the tangent to the profile of the master form in the section being copied. In approximately the same manner is the flow of oil coming from the command slide valve distributed among the hydraulic cylinders of the upper longitudinal and cross carriages, the velocities v_{xg} and v_{yg} being thus created. On the whole, the direction of the resulting velocity v_{0g} of the upper pair of carriages in automatic movement along the profile coincides approximately with the direction of the perpendicular to the profile of the master form on the section being copied.

Since the closed profile sections on which the direction of feed v_{0e} from the electromotors deviates from the direction of the tangent, within the profile or towards the exterior, differ in length, the magnitude of the deviations themselves also vary; in the process of automatic copying, constant displacement of the upper pair of carriages from the fixed neutral position may take place. In subsequent outlinings of the profile, accumulation of these displacements is possible, in consequence of which the pistons of the hydraulic cylinders may come to a stop and the adjusting action cease. A system of automatic levelling of the travel of the upper pair of carriages consting of two potentiometers (of the upper cross and longitudinal carriages) fed by direct current from an exciter 25 (Figure 2) is employed to prevent this from occurring.

The moving slides of the potentiometers are secured on the upper cross and upper longitudinal carriages, while the potentiometers themselves are secured on the upper longitudinal and lower cross carriages, with respect to which the carriages bearing the slides are displaced from the hydraulic cylinders.

The neutral positions of the potentiometers correspond to the neutral positions of the upper carriages. The amount of voltage taken from each potentiometer is proportionate to the amount of displacement of their slides from the neutral position, and the sign depends on the direction of this displacement.

When the carriages are displaced from the neutral position (as a result of the action of the velocities v_{xg} and v_{yg} from the hydraulic cylinders), supplementary voltages are sent through the electromechanical amplifiers 24 and 23 to the electromotors of the lower carriages. The velocities v_{xe} and v_{ye} change in consequence of this in such a way that their resultant velocity approaches the direction of the tangent (in the diagram in Figure 4, v_{xe} decreases, v_{ye} increases), and the magnitude of the velocity v_{0g} strives toward zero.

It follows from the principle of operation of the machine given above that the tracking drive proper is a very simple single-circuit tracking system with return completion coupling (Figure 5). This system consists of a controlling throttle reversible slide valve 1 which controls the hydraulic system, two distribution slide valves 2 which regulate the position of the hydraulic system, and two hydraulic power cylinders 3 situated at an angle of 90° to each other.

The constructional execution of the units mentioned may be seen from the hydraulic diagram showing the operation of the adjustment system, which is given in Figure 6.

The command slide valve has as the main operating unit a reversible throttle slide valve of the circulating type with oil circulating constantly through all four operating apertures, with a 90° angle of inclination of the edges. The slide valve is rigidly connected to the rod of the hydraulic system command transmitter, which reproduces the deflections of the main finger of the copying instrument and is in a state of constant oscillation of small amplitude with a frequency of 50 cycles per second. Thanks to this, the influence of dry friction is eliminated and stability of spouting of oil through the operating apertures of the slide valves is achieved.

The position regulator of the hydraulic system has as the main operating elements two reversible throttle slide valves with an angle of inclination of the working edges of 6°. Control of the movements of the slide valves is exercised from eccentrically mounted ball bearings with adjustable accentricity. The bearing shaft turns in strict conformity with the turning angle of the rotor of the copying instrument by means of a selsyn synchronous tracking system. Thanks to such a design, accurate and smooth distribution is achieved of the flow of oil between the two hydraulic cylinders in proportion to the sin and cos of the angle of the tangent to the profile of the master form at the point of contact between it and the main finger of the copying instrument, and the possibility is created of adjusting the amount of flow (by changing the eccentricity).

Figure 5. Structural diagram of the tracking (adjustment) drive the the KFS-3 machine: 1 - command slide valve; 2 - hydraulic system position regulator (sine distributor); 3 - hydraulic power cylinders; 4 - x_{ex}; 5 - master form; 6 - x_{en}.

SOME DATA GAINED FROM EXPERIENCE IN OPERATION OF THE KFS-3 TWO-COORDINATE ELECTRIC-HYDRAULIC TRACKING DRIVE

The two-coordinate combined electric-hydraulic tracking drive under discussion was tested on an experimantal model, and on the basis of the latter, the KFS-3 copying milling machine, which has given a good performance in operation, was developed, completed, and introduced into production.

On the KFS-3 copying milling machine, the machining is performed of the profiles of steel forms in one sheet or in piles, of a thickness of up to 12mm, a length of up to 3800mm, with an accuracy of $\pm$0.1 mm from the profile of the master form, with a cleanness of the machined surface conforming to the fifth and seventh classes of GOST-2789-51, depending on the condition of the cutting tool and the milling rate. The feeding may be regulated from the control desk during the machining process, within a range from 20 to 400 mm/min. The machine makes it possible to machine automatically profiles with any external radius, external acute angles of more than 45°, internal radiuses of more than 15mm, and internal angles of more than 135° without curvature at the vertex.

The operator secures and removes the article and sets the copying instrument and cutting tool for automatic operation. The entire process of outlining of a closed contour is performed automatically, without participation of the operator, at a constant feeding speed when any desired profile is being outlined. On steep sections of the latter (with a radius of less than 30mm), it is recommended that

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Figure 6. Hydraulic diagram of the principle of the adjustment system of the KFS-3 machine: 1 - command slide valve; 2 - slide valve of position regulator; 3 - upper longitudinal carriage; 4 - position regulator; 5 - transmitter of commands to hydraulic system; 6 - upper cross carriage.

the feeding speed be reduced to 60 to 80 mm/min for the purpose of retaining the high copying accuracy. The machining accuracy, i.e., the deviation of the dimensions of the article from the dimensions of the master form, is made up of the accuracy of the mechanical system of the machine and the accuracy of the tracking system.

The accuracy of the mechanical system of the machine, which is determined chiefly by the play in the guides of the cross carriages and the linearity of the guides, reaches ± 0.03mm for the play in the guides and 0.05mm for the linearity along the 6000mm length of the guides.

The accuracy of the tracking system of the machine is determined by the accuracy created by the adjustment system only insofar as hydraulic power cylinders are the performing members of the tracking drive of the machine. Consequently, the play in the long kinematic chains of the feed drives from the electric motors, which reached 1mm in the screw-nut articulation (in tests); the considerable inertia of the electric feed drive motors, each having a power of 2.75 kilowatts; and the dead angle of the motors upon reversal, which reaches as high as 10° (computed by the turn of the rotor of the copying instrument) do not markedly affect the copying accuracy.

As has already been noted above, the adjustment system is a constant control tracking system, the controlling signal of which is proportionate only to the magnitude of the loss of adjustment between the positions of the pickup and performing members. The copying action is performed as a result of deflection of the command slide valve from the neutral position (Figures 5 and 6). The changes in the pressure and flow of the fluid necessary for displacement (response) of the hydrSTATic cylinders take place as a result of this deflection in the command slide valve -- hydraulic cylinders hydraulic chain.

The accuracy of the hydraulic part of the adjustment system is made up basically of three components: the zone of insensitivity, which depends on the coefficient of dry friction in the system; the zone of inaccuracy determined by the velocity of displacement of the hydraulic cylinder; and the zone of inaccuracy of the function of the load on the hydraulic cylinders. In the KFS-3 machine a ratio of 1:1 between the copying (main) finger and the command slide valve has been adopted, i.e., the linear deflections of the copying finger from the neutral position, measured at the point of contact with the master form, and those of the command slide valve along its longitudinal axis are practically equal.

Special studies made of the discharge of mineral oil through the operating apertures of the throttle valves have demonstrated the possibility of achieving a high copying accuracy with the system in question.

In the command slide valve, the opening (width) of the operating apertures in the neutral position amounts to 0.015mm, and in the slide valves of the position regulator, approximately zero. The discharge -- function of deviation of the slide valve from the neutral position characteristics for all the apertures of the slide valve are evened out in such a way by the hydraulic testing method that the difference in the width of all the apertures in the neutral position does not exceed 5 microns. Diagrams showing the conduct of the hydraulic tests and the characteristics of the command slide valve and those of the position regulator are given in Figures 7 and 8.

Figure 9 shows the characteristics of the command slide valve from which it is possible to determine the zone of insensitivity of the system in relation to the amount of dry friction. Dry friction in the drive system of the upper pair of carriages of the KFS-3 machine corresponds to a pressure drop in the power cylinders on the scale of 8 to 10 kg/cm^2 (diameter of the cylinders, 120mm); the zone of insensitivity must consequently lie within the limits of $\pm$ 0.007 mm (Figure 9). In fact, a condition of forced oscillations having an amplitude of 0.12 to 0.15 mm applied to the command slide valve greatly moderates the characteristic of the slide valve, and the zone of insensitivity amounts to $\pm$ 0.02 mm.

The deviation of the direction of the resulting feed velocity v_{0e} (Figure 4) of the lower pair of carriages from the electric motors amounts to $\pm$ 15° from the tangential in the process of outlining of a closed profile. This is the result of the error of the characteristics of the flat potentiometer of the copying instrument, the electronic and electromechanical amplifiers, and of the various loads on the electric motors (the weight of the moving units of the drive of the lower longitudinal carriage is 8.5 tons, that of the lower cross carriage, 5 tons). Hence when the feeding speed is 400 mm/min, the tracking speed can reach only 104 mm/min. It is actually much lower than this, as a result of the system of automatic levelling of the travel of the upper pair of carriages, and increases only on the sharp sections. For this reason, the zone of inaccuracy -- function of speed of displacement of the hydraulic cylinders usually amounts to 0.01 to 0.02 mm.

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Figure 7. Relation of discharge Q of oil through the operating apertures of the command slide valve to the deviation of the latter Δh from the neutral position: a - test curve; b - test diagram: 1 - rotary oscillation drive; 2 - micron indicator; 3 - command slide valve; 4 - discharge of oil Q, g/min; 5 - neck III; 6 - neck I; 7 - neck II; 8 - neck IV; 9 - deviation of slide valve from neutral position Δh, in microns; 10 - P_p.

Figure 8. Relation of discharge of oil Q through the operating apertures of the slide valves of the position regulator to their displacement Δh from the neutral position: a - test curve; b - test diagram: 1 - rotary oscillation drive; 2 - micron indicator; 3 - slide valve of position regulator; 4 - discharge of oil Q, g/min; 5 - neck II ; 6 - neck IV; 7 - neck III; 8 - neck I; 9 - deviation of slide valve from neutral position Δh, in microns; 10 - P_p.

Figure 9. Relation of the pressure drop ΔP in the internal circuit of the command slide valve to the displacement Δh from the neutral position: a - test curve; b - test diagram: 1 - rotary oscillation drive; 2 - micron indicator; 3 - command slide valve; 4 - ΔP, kg/cm^2; 5 - deviation of slide valve from neutral position, in microns; 6 - P_p.

As for the inaccuracy zone -- load on hydraulic cylinders function, it is an insignificant amount, because of the fact that the milling forces are much smaller than dry friction in the drive system of the carriages. Thus, the accuracy of the tracking system of the KFS-3 machine in smooth outlinings is $\pm$ 0.03 -- 0.04 mm. A visual representation of the accuracy of the tracking system is achieved with a voltmeter installed on the control desk, connected in the circuit of the inductive transmitter of the copying instrument, which permits the operator to judge the copying accuracy while the machine is running.

The deviation of the direction of the resulting feed velocity v_{0g} (Figure 4) of the upper pair of carriages from the hydraulic cylinders is $\pm$ 10° from the perpendicular to the profile in the process of outlining of a closed profile. For this reason, the copying accuracy practically does not depend on the configuration of the profile and its location with respect to the longitudinal and cross feeds and is identical both on inclined sections and at points of reversal of the feeds. The travel of the hydraulic cylinders in the process of outlining of closed profiles lies between the limits of $\pm$ 30mm from the neutral position.

Work was subsequently conducted towards perfection of the two-coordinate electric-hydraulic tracking drive under discussion, in the direction of increasing the feeding speed during copying, simplifying the tracking drive, and increasing the operational capabilities of the machinery equipped with the tracking drive.

The simplified KFS-3B tracking drive, a structural diagram of which is given in Figure 10, has been developed and produced for this purpose. The design of this tracking drive has retained intact the copying principle discussed, but it has been substantially simplified; in particular:

(a) the inductive transmitter 20 (Figure 2), the electronic amplifier 37, and the transmitter of commands to the hydraulic system 38 have been eliminated, and the command slide valve of the hydraulic system has been mounted in the copying instrument and connected to the copying finger by a rigid kinematic connection;

(b) the selsyn transmitter 19 (Figure 2), selsyn receiver 29, electronic amplifier 30, electromechanical amplifier 31, motor of the electromechanical amplifier, and drive motor 32 have been eliminated, and the position regulator of the hydraulic system has been mounted in the copying instrument and the cam connected to the rotor of the copying instrument by a rigid kinematic connection.

Figure 10. Structural diagram of the KFS-3B machine tracking drive.

The new simplified KFS-3B tracking drive provides for expansion of the operating characteristics of the machines equipped with it. With this aim, the auxiliary finger in the copying instrument has been eliminated; the turning of the rotor of the copying instrument in conformity with the angle of inclination of the profile being copied is accomplished from the hydraulic motor, which is controlled from the command slide valve of the hydraulic system; the danger has been eliminated of breakage of the copying finger in the outlining of internal angles of less than 135°, by the introduction of a spherical support for the copying finger; units have been introduced which, controlled from the command slide valve, automatically reduce the feeding speed on sections with small radiuses of curvature, which should contribute towards increase in copying accuracy; and the possibility of both counterclockwise and clockwise outlining of a profile has been introduced.

DIAGRAM OF PRINCIPLE OF THE KFG-1 TWO-COORDINATE HYDRAULIC TRACKING DRIVE

In the following stage of simplification of the tracking drive, a reduction of the number of the cross carriages from four to two and of the number of power servo drives from four (two electric and two hydraulic) to two (hydraulic) was achieved, while the operating capabilities of the equipment were expanded.

A diagram of the principle of the two-coordinate hydraulic tracking drive developed for these purposes is given in Figure 11.

Figure 11. Diagram of principle of the KFG-1 two-coordinate hydraulic tracking drive: 29 - master form; 30 - from pump; 31 - direction of eccentricity of feed speed along profile; 32 - overflow; 33 - Δ P; 34 - direction of adjustment eccentricity.

The servomotors 1 and 2, which set the performing member in movement along the two axes of the coordinates, are controlled by two reversible throttle slide valves 3 and 4 with tapered necks, which, together with the double cam 5, form a sine transmitter. The double cam consists of two pairs of sloping bushings, inner and outer. If the sloping bushings are in the initial position, the amount of eccentricity of the outer ring of the double cam 5, against which the slide valves 3 and 4 rest, equals zero with respect to the axis of rotation, in which case oil does not move into the servomotors 1 and 2. The outer pair of sloping bushings are moved manually or mechanically from the initial position. When this is done, the outer ring of the double cam is displaced with respect to the axis of rotation, i.e., acquires the eccentricity ΔP, the amount of which is proportionate to the displacement of the outer pair of sloping bushings from the initial position, and the direction, automatically controlled, in the process of outlining of a profile coincides approximately with the direction of the tangent to the profile of the master form 6 at the point of contact with the copying finger 7. As a result of this, deviations from the neutral position which are proportionate to the sin and cos of the angle of inclination of the tangent to the profile of the master form on the section being copied are communicated to the slide valves 3 and 4. Approximately the same relationship is created between the streams of oil moving from the slide valves into the servomotors, creating the speeds v_{feed1} and v_{feed2} (Figure 4), the resulting velocity of which -- the feed speed along the profile -- v_{feed} in the outlining of a profile is constant in magnitude (determined by the amount of eccentricity ΔP) and is near the tangent in direction.

Automatic control of the direction of the eccentricity ΔP is accomplished from the copying finger 7 (Figure 11) by means of a command slide valve 8 and hydraulic motor 9.

The command slide valve occupies the neutral position (the apertures between the edges of the slide valve and the grooves in the body are equal and the pressures in the pipelines 10 and 11 are identical) when the copying finger is deflected from the vertical position, in which case the ball 12, rolling out in the cones, lifts the command slide valve from the extreme lower position, overcoming the force of the spring 13.

When the copying finger moves along a profile at the speed v_{feed} directed toward the exterior of the profile, to the right of the tangent, as shown in Figure 4, the finger begins to move away from the profile of the master form, and the command slide valve is lowered from the neutral position. The pressure in pipeline 10 is hereupon increased and that in pipeline 11 lowered. As a result, the hydraulic motor 9 begins to rotate the double cam counterclockwise, changing the direction of the feeding speed v_{feed} so that it is towards the interior of the profile (to the left of the tangent), thus moving the copying finger to the profile of the master form. The command slide valve now moves up. When the command slide valve has gone above the neutral position, the pressure in the pipeline 11 becomes greater than the pressure in the pipeline 10, as a result of which the rotation of the hydraulic motor 9 is reversed and the double cam begins to rotate clockwise, again drawing the direction of the feeding speed v_{feed} nearer the direction of the tangent to the profile of the master form on the section being copied.

In the process of outlining of a closed profile, the direction of the feeding speed v_{feed} thus varies automatically around the direction of the tangent to the profile of the master form on the section being copied.

In order to eliminate the fluctuating nature of regulation of the speed v_{feed} and to achieve a high copying accuracy, parallel automatic control by the parameter of the amount and first derivative of the linear loss of adjustment of the location of the copying finger with respect to its neutral position has been introduced into the tracking drive; this control will subsequently be designated adjustment control (on the analogy of the terminology adopted above for the electric-hydraulic drive).

Control is exercised along the adjustment channel by the double cam 5 from the copying finger 7, by means of the command slide valve 8, the hydraulic amplifier 14, and the inner pair of sloping bushings, to which latter is communicated the displacement from the neutral position, corresponding in magnitude and sign (into the profile or away from the profile of the master form) to the deviation of the copying finger from the neutral position.

The outer ring of the double cam receives the additional eccentricity $\pm \Delta k$, which is always situated at an angle of 90° to the feed eccentricity ΔP and is proportionate to the amount and direction of the deviation of the inner pair of sloping bushings from the neutral position. As a result, the servomotors 1 and 2 are automatically assigned such speeds, v_{adj1} and v_{adj2} (Figure 4), that their resulting speed v_{adj}, adding up geometrically to the speed v_{feed}, gives the speed v_0 at any point of the profile of the profile of the master form coinciding with the direction of the tangent.

Control in accordance with the first derivative is accomplished by the inner pair of sloping bushings of the double cam from the pair of plungers 16 and 17.

In the case of slow deflection of the copying finger from the neutral position, the plunger 15 of the hydraulic amplifier 14 is slowly displaced from the neutral position, for example, upwards (Figure 11), compressing the spring 18, which always strives to return the plunger 15 of the hydraulic amplifier to the neutral position. By means of a lever 19, which is rigidly connected to plungers 15 and 16, and through the spring 20, which always strives to keep unchanged the position of the plunger 17 with respect to the plunger 16, plunger 17 displaces the inner sloping bushing of the double cam by the same amount and at the same speed as plunger 15 of the hydraulic amplifier. Oil hereupon flows from cavities 21 and 22 of the damped cylinder and moves into cavity 23. Due to the fact that the diameters of all the rods of plungers 16 and 17 are equal, the amount of oil in volume of the damper cylinder does not change during displacement of the plungers. Oil flows from cavity 22 (when the plungers are displaced upward) or into it (when the plungers are displaced downward), overcoming the resistance of the throttle 24.

In consequence of the fact that the resistance of the throttle is proportionate to the speed of the oil running through it, and hence also to the speed of displacement of plunger 16, as the speed of deflection of the copying finger from the neutral position increases, the displacement of plunger 17 begins to outstrip the displacement of plungers 15 and 16 (which are rigidly connected to each other) the more, the greater is this speed. As a result, the inner sloping bushing acquires additional displacement with respect to plunger 15 of the amplifier, which displacement is the greater, the greater is the speec of deflection of the finger from the neutral position.

The overflow from the servomotors 1 and 2 is directed along pipeline 25 through the command slide valve in such a way that when the command slide valve is deflected to a slight extent up or down from the neutral position, passage of the fluid is hampered to the point of complete stoppage. This makes it possible to improve the operation of the tracking system in the outlining of sharp sections of the profile of the master form, for example, inner right angles, by automatic reduction of speed when there is considerable loss of adjustment between its direction and the direction of the tangent.

The throttles 26 and 27 included in the system serve to regulate the performances of the hydraulic amplifier and hydraulic motor. Forced oscillations are forced on the command slide valve unit by means of rotary oscillations of the command slide valve having a frequency of 50 cycles per second to an angle of $\sim 5^{\circ}$ or axial oscillations of the bushing of the command slide valve having a frequency of 100 cycles per second and an amplitude on the scale of 0.05 mm.

CERTAIN RESULTS OF TESTING OF THE KFG-1

TWO-COORDINATE HYDRAULIC TRACKING DRIVE

The simplified two-coordinate hydraulic tracking drive under discussion was tested on the 6N12 machine (Figure 12), which was especially remodelled for hydraulic feeds in longitudinal and lateral directions and for the mounting of a copying instrument. The hydraulic feed drive consists of a pair of cross carriages 1 (Figure 12) moved by hydraulic cylinders 2 120mm in diameter; it is secured on the bedplate 3 of the machine. The copying instrument 4 of the KFS-3B tracking drive was used as the base for mounting the copying instrument. The instrument is secured to the forward wall of the body of the milling head 5.

Figure 12. The 6N12 machine, equipped with the KFG-1 two-coordinate hydraulic tracking drive (experimental model).

Tests have shown that a tracking drive produced in accordance with the diagram shown in Figure 11 is fully efficient and insures copying at speeds above 1000mm/min (in the tests, the copying speeds varied from 100 to 1200mm/min).

The gear ratio of the reduction gear of the hydraulic motor to the shaft of the double cam adopted was 1:7. Reduction of the gear ratio to 1:30 produced worse results with respect to rapid operation of the drive.

The hydraulic system of control of the double cam from the command slide valve requires a specific feed pressure, which depends on the area of the plunger and the rigidity of the spring of the hydraulic amplifier. In the drive tested, this pressure was 25 to 30 kg/cm^2. The hydraulic power system of the feed can operate within a wide range of pressures in the delivery pipeline (the system was tested for pressures from 10 to 50 kg/cm^2).

The greatest stability of the tracking drive takes place when axial oscillations are applied to the bushing of the command slide valve; worse results are obtained when axial or rotary oscillations are absent.

The tracking drive makes it possible to outline any desired closed profile having outer and inner right angles, without curvature; the passage of the oil overflow from the cylinders through the command slide valve exercises substantial influence in the outlining of inner right angles. A certain amount of overflow into the command slide valve circuit should be provided. Play in the transmission chain from the command slide valve to the double cam has an adverse effect and it should be eliminated.

An increased pull of the hydraulic motor is observed at the points of reversal of the feeds, for which reason a certain increase in the copying error may take place here at speeds above 500 mm/min. In the outlining of a closed, smooth profile, the copying accuracy of the tracking drive at speeds of 600 mm/min is 0.03 mm, when an axial vibrator is present.

The copying accuracy is little affected by the copying speed and the exactness of coincidence of the direction of the copying speed vector and the eccentricity of the outer pair of sloping bushings of the double cam.

In order to insure reliable operation of all the elements of the hydraulic amplifier, it is necessary to insure copious delivery of lubricant to all its friction parts.

On the basis of the positive results of the testing of the tracking drive under discussion, there has been developed a special copying-milling machine of the KFG-1 type with a length of travel in the longitudinal direction of 1500 mm; in the lateral, of 1000 mm; and in the vertical, of 800 mm.

The machine is intended, in addition to end milling of the closed profiles of complex form of parts of steel and light alloys, by automatic copying, also for end milling of three-dimensional surfaces of parts of the coining die type, by automatic copying with periodic feed onto the line in the lateral direction. The same tracking drive is used in both instances; in the first case, it is connected to the hydraulic longitudinal and lateral movement cylinders, and in the second case, to those of longitudinal and vertical movement. It has proved possible to produce the hydraulic power cylinders with a single-ended rod (which is simpler in design than hydraulic cylinders with a double-ended rod and makes it possible to have a longer length of travel with the same overall dimensions), since asymmetry of the working surface of the hydraulic cylinders does not affect the copying accuracy.

CONCLUSIONS

The two-coordinate electric-hydraulic tracking drives discussed determine the following features of the equipment:

(1) Automatic outlining of a closed profile without any intervention of the operator;

(2) Constant feeding speed in the outlining of any desired profile;

(3) Tracking at speeds comprising a small part of the copying speed;

(4) High copying speed with a high accuracy;

(5) Only slight influence of play in the kinematic chains of the feed drive on copying accuracy;

(6) Uniform copying accuracy, independently of the situation of the profile with respect to the longitudinal and cross feeds, as well as of the points of reversal of the feeds;

The KFG-1 two-coordinate hydraulic tracking drive retains the positive features of the KFS-3 electric-hydraulic tracking drive, noted above, but requires the elimination of play in the kinematic chains of the feed drives. Together with this, it makes it possible to expand greatly the operating capabilities of the machinery equipped with it and to simplify the design of the latter.

The tracking drives discussed make it possible to create highly productive automatic metal-cutting equipment for the high-accuracy machining of parts of large dimensions and complex form both with respect to the profile and to the size.

B. T. KOLOMIETS

PHOTOELECTRIC RESISTORS IN PHOTOELECTRIC AUTOMATION[1]

Especially prominent among the many modern systems and methods of automation and control of production processes is so-called photoelectric automation.

At the present time our country's engineering has available high quality phototubes which meet all the rigid requirements. Antimony-cesium phototubes are characterized by their high sensitivity and stability, qualities which, in combination with the availability of diversified types of phototubes (the STSV, FEU-1, FEU-19) and structural designs (STSV-3, STSV-4), create extensive possibilities for the development of photoelectric automation.

New phototubes, which have been designated photoelectric resistors, have recently come into use in engineering.

It may be pointed out that the very first phototubes were photoelectric resistors of selenium, obtained as early as 1872. However, they did not become established in practice, as was true also of those of thallium sulfide (the thalofides), discovered in 1917. This is explained by their serious shortcomings: instability in operation, great inertia, temperature dependence, and non-linearity between the photoelectric current and the luminous flux. The shortcomings of the first resistors and the absence of any new ones, until 1945, resulted in a negative attitude towards the question of the possibility of utilizing photoelectric resistors for the solution of technical problems.

The recently conducted detailed study of the photoelectric properties of semiconductors has shown that the shortcomings of the first photoelectric resistors of selenium and thallium do not constitute an organic characteristic of all the semiconductor materials from which the former are made. It is possible to obtain photoelectric resistors in which the shortcomings listed above are manifested to the minimum and cannot act as an obstacle to their employment in engineering. Among these are, for example, photoelectric resistors of lead sulfide, bismuth sulfide, and cadmium sulfide.

At the present time industry has developed and is producing photoelectric resistors of lead sulfide, bismuth sulfide, and cadmium sulfide in several constructional versions of each type, namely, the FS-A1, ... FS-A5; FS-B0, ... FS-B2; and the FS-K1 and FS-K2.

In their design, the photoelectric resistors are very simple. They represent ordinary ohm resistors of a thin, homogeneous layer of a semiconductor the resistance of which when illuminated decreases hundreds of times.

Given below are the basic data on the industrial types of photoelectric resistors mentioned.

CHARACTERISTICS OF INDUSTRIAL TYPES OF PHOTOELECTRIC RESISTORS

Overall sizes and ohmic resistance

The main photoelectric resistors of the mass type are characterized by their small dimensions and electrodes designed for switching into a radio tube panel.

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[1] Printed in abridged form. -- Ed.

Figure 1 shows sketches of the new photoelectric resistors of the FS-K1 and FS-K2 types. It must be emphasized that the dimensions of the resistors of all the types are not the minimum ones, but depend on the system adopted of making contact with the radio tube panel. If necessary, the resistors may be produced in practically any size. Thus, there are photoelectric resistors with reception areas of 4 cm^2 and 3 mm^2.

The dimensions of the photosensitive surface of the standard designs of the photoelectric resistors, as well as their ohmic resistance, are given in Table 1.

Figure 1. Overall dimensions of photoelectric resistors: 1 - of the FS-K1 type; 2 - of the FS-K2 type

Table 1

Type of photoelectric resistor	Dimensions of photosensitive surface, in cm	area, in cm^2	dark resistance, in ohms
FS-A1	0.3 X 0.7	0.21	$1 \cdot 10^4$ -- $2 \cdot 10^5$
FS-B2	1.1 X 1.1	1.21	$2 \cdot 10^5$ -- $1 \cdot 10^7$
FS-K1	0.35 X 0.72	0.25	10^7 and above
FS-K2	0.35 X 0.72	0.25	10^6 and above

Sensitivity

Figure 2 gives the functional voltampere characteristics of the photoelectric resistors, recorded in the dark and with illumination. Figure 3 illustrates the main distinguishing feature of the photoelectric resistors, which is manifested in the circumstance that their current sensitivity is proportionate to the voltage and has no saturation, in contrast to phototubes with external photoelectric effect.

"Specific sensitivity", i.e., the amount of photoelectric current per volt of applied voltage with a certain specified amount of luminous flux or intensity of illumination, has been introduced for the purpose of mutual comparison of the various types of photoelectric resistors. This latter must be borne in mind because of another characteristic of photoelectric resistors, the lack of proportion between the photoelectric current and the luminous flux. The approximate nature of this relationship, which differs for the various types of resistors, is shown in Figure 3.

The maximum sensitivity of photoelectric resistors is determined by the maximum permissible operating voltage. Data concerning the amount of maximum voltage are given in Table 2.

It is sometimes convenient to characterize the sensitivity of photoelectric resistors by the magnitude of relative resistance change and by the multiplicity. These data also are given in Table 2. The amounts of specific sensitivity in this table are the maximum, while the relative changes and multiplicity are the average. For the FS-A1 and FS-B2, the data pertain to an intensity of illumination on the scale of 200 luces, and for the FS-K1 and FS-K2, 100 luces.

In consequence of a certain inertia of the FS-B2, FS-K1, and FS-K2 resistors, the amounts of all the types of sensitivity were determined from the amounts of current or resistance 15 seconds after cessation of the illumination. The amount of maximum sensitivity of the photoelectric resistors was found as the product of the amounts of maximum operating voltage and specific sensitivity.

Figure 2. General view of the voltampere characteristics of photoelectric resistors: 1 - current under illumination; 2 - current in darkness; 3 - current; 4 - photoelectric current; 5 - voltage.

Figure 3. General view of relationship of photoelectric current to intensity of illumination: 1 - photoelectric current; 2 - luminous flux.

Table 2

Type of photoelectric resistor	Maximum operating voltage, v	Specific sensitivity, mka/lm · v	Relative change of resistance $\frac{\Delta R}{R} \cdot 100\%$	Multiplicity of change
FS-A1	15	500	17	1.2
FS-B2	50	1000	80	4
FS-K1	400	3000	99.28	140
FS-K2	300	2500	97.14	35

Inertia

The photoelectric resistors of the FS-A1 type are characterized by low inertia, this making it possible to ignore sensitivity losses upon modulation of the luminous flux, up to 1000 per/sec. The FS-B2, FS-K1, and FS-K2 photoelectric resistors have worse properties of inertia, as may be seen from Figure 4. An idea of the nature of the inertia may be gained in addition from Figure 5, in which is shown the progress of growth and diminution of the photoelectric current with a rectangular impulse of the luminous flux illuminating the photoelectric resistor. As investigations have

Figure 4. Nature of inertia in various types of photoelectric resistors: 1 - FS-A1; 2 - FS-B2; 3 - FS-K1; 4 - FS-K2; 5 - vacuum phototubes; 6 - useful signal; 7 - frequency; 8 - per/sec.

Figure 5. Nature of inertia of photoelectric resistors of the FS-K1 type with a single rectangular light impulse: 1 - photoelectric current; 2 - light.

shown, the regularities of growth and diminution of the current differ, and their constants change in various sectors of current change. It follows from Figure 5 that the growth of the current occurs more rapidly than the diminution. By determining the inertia in accordance with the diminution and accepting the exponential principle of this diminution, it is possible to determine with adequate accuracy the amount of sensitivity or current in a certain interval of frequencies of modulation of the luminous flux, in accordance with the formula

$$I_f = I_0 \frac{1}{\sqrt{1 + (2\pi f \tau)^2}}, \qquad (1)$$

where I_f is the sensitivity or photoelectric current with a frequency of modulation of f;

I_0 is the sensitivity under constant illumination;

f is the frequency of modulation in per/sec;

τ is the time constant.

The constant τ represents the time during which the photoelectric current changes e times. The models of the photoelectric resistors had the values of τ given in Table 3.

The data were obtained with a sinusoidal modulation of the light and small amounts of luminous flux, on the scale of 10^{-5} lumina. When the intensity of illumination is increased, the time constant is decreased.

Table 3

Type of photoelectric resistor	τ, in sec	Interval of frequencies, in per/sec	Type of photoelectric resistor	τ, in sec	Interval of frequencies, in per/sec
FS-A1	$4 \cdot 10^{-5}$	1000 - 6000	FS-K1	$25 \cdot 10^{-3}$	10 - 100
FS-B2	$1 \cdot 10^{-3}$	100 - 400	FS-K2	$31 \cdot 10^{-3}$	10 - 100

Temperature dependence

The considerable dependence of the photoelectric current on temperature is a serious obstacle to the utilization of photoelectric resistors under conditions of great temperature diversity.

A great achievement of recent times was the achievement of photoelectric resistors with a very small temperature dependence (FS-K2). The photoelectric resistors made from cadmium sulfide, but of another type (the FS-K1), have a temperature dependence scarcely differing from those of the FS-A1 and FS-B2 photoelectric resistors. The dependence of the photoelectric current on temperature is given in Table 4 for the photoelectric resistors of the FS-K1 and FS-K2 types.

Table 4

Temperature, in °C	Amounts of photoelectric current, in mka		Temperature, in °C	Amounts of photoelectric current, in mka	
	FS-K1	FS-K2		FS-K1	FS-K2
-100	720	475	20	245	285
- 80	600	425	40	200	280
- 60	500	390	60	160	276
- 40	415	350	80	120	272
- 20	345	320	100	90	270
0	300	300			

Stability

In contrast to the first types of photoelectric resistors, all the new industrial types are characterized by a high stability. For example, during 2000 hours of continuous operation after a period during which the properties were established, no irreversible changes in the characteristics were discovered in them.

Spectral sensitivity

The FS-K1 and FS-K2 photoelectric resistors have sensitivity only in the visible region of the spectrum. The sensitivity of the FS-B2 embraces the near infrared region. The photoelectric resistors of the FS-A1 type on the whole have sensitivity only in the infrared region of the spectrum. Figures 6 and 7 illustrate the spectral characteristics of all the photoelectric resistors listed. Mention must be made of the photoelectric resistors of the FSK-M1 type, developed by the Institute of Physics of the Ukrainian Academy of Sciences. They are remarkable for the fact that, in addition to the visible region, they are sensitive to ultraviolet and roentgen rays. The specific sensitivity of the FSK-M1 is 300 mka/lm $\cdot$ v. They are characterized by a smaller inertia as compared with the FSK-1 and FSK-2.

It may be seen from the characteristics of the new photoelectric resistors cited that a number of the defects formerly present are absent from the present-dSTATindustrial types of photoelectric resistors. There are now photoelectric resistors with slight inertia (the FS-A1) and slight temperature dependence (the FS-K2). It has recently proved possible to obtain photoelectric resistors in which the photoelectric current is proportionate to the luminous flux. Instability in operation, one of the chief defects inherent in photoelectric resistors, has been eliminated from all the industrial types.

Photoelectric resistors should now be considered reliable instruments deserving of a place of honor in engineering practice.

Figure 6. Nature of spectral sensitivity: 1 - the FS-A1 photoelectric resistor; 2 - the FS-B2 photoelectric resistor; 3 - photoelectric current per unit of falling energy, in relative units; 4 - wavelength, in mk.

Figure 7. Spectral sensitivity of photoelectric resistors: 1 - the FS-K1; 2 - the FS-K2; 3 - photoelectric current per unit of falling energy, in relative units; 4 - wavelength, in mmk.

PHOTOELECTRIC RESISTORS IN INDUSTRIAL AUTOMATION

The sphere of application of photoelectric resistors in photoelectric automation at the present time is limited to the simplest tasks, in which inertia cannot play a substantial part. Experience indicates, however, that there is a fairly large number of such simple tasks, for which reason the question of utilization of photoelectric resistors is of great technical significance.

The first step is the replacement of the vacuum phototube by a photoelectric resistor in existing systems with electronic lamps. The much higher integral sensitivity of the latter may serve as the occasion for this. The amount of useful signal may be determined simply by the current sensitivity:

$$\Delta v = k \cdot v \cdot R \cdot \Phi \qquad (2)$$

or by the relative change of resistance

$$\Delta v = v\left(\frac{1}{2} - \frac{1 - \frac{\Delta R}{R}}{2 - \frac{\Delta R}{R}}\right), \qquad (3)$$

where k is the specific sensitivity;

v is the voltage applied to the photoelectric resistor (in formula 2) and the full operating voltage (in formula 3);

R_n is the load resistance;

Φ is the luminous flux;

$\frac{\Delta R}{R}$ is the relative change of resistance. Expression (2) is correct for R_n $R_{\Phi s}$; expression (3), for $R'_n = R'_{\Phi s}$, where $R'_{\Phi s}$ is the light resistance, determined in turn by the formula

$$R'_{\Phi s} = R_{dark}(1 - \frac{\Delta R}{R}). \qquad (4)$$

It follows from (3) that with large values of $\frac{\Delta R}{R}$ (Table 2), the useful signal may equal half the total supply voltage. This circumstance is graphically illustrated by Figure 8, in which are shown the load characteristics of the FS-B2, which give an idea of the amount of useful signal at various load resistances and with various intensities of illumination. The data were obtained with a supply voltage of 30 volts.

Figure 8. Load characteristics of photoelectric resistors of the FS-B2 type with a supply voltage of 30 volts: 1 - intensity of illumination of 20 luces; 2 - intensity of illumination of 200 luces; 3 - useful signal; 4 - load resistance; 5 - 10^6 ohms.

The great potential differences for control of the operation of an electronic lamp greatly simplify the creation of reliable photoelectric equipment. Photoelectric resistors have another merit, the possibility of removing the light receiver to long distances from the performing members and of situating it in places difficult of access.

The first attempt to replace a vacuum phototube in a photorelay with a semiconductor photoelectric resistor was made with an industrial design, which has been given the model designation of FR-4. A functional electric diagram of such a photorelay is given in Figure 9, and an overall view in Figure 10. In view of the dependence of the photoelectric current on voltage and intensity of light, and hence on fluctuation of voltage in the circuit, supply is effected by the introduction of ferroresonance stabilization. Also provided for in the circuit are supply by direct current of an electromagnetic counter and lowered voltage for the illuminating lamp.

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Photorelays such as these have been successfully used in emergency circuits, counting articles in assembly-line production, and in the automation of certain production processes.

Figure 9. Electrical circuit of a photorelay with photoelectric resistors: 1 - 127v circuit; 2 - FS; 3 - relay contacts.

Extensive possibilities for the development of photoelectric automation are created as a result of the appearance of the photoelectric resistors of the FS-K1 and FS-K2 types, the new quality of which consists in the possibility of obtaining from them photoelectric currents measured in milliamperes. Protracted loading with a photoelectric current of 2 ma, with a direct current supply voltage of 100v does not cause any irreversible changes in the photoelectric resistors. A photoelectric current of 10 to 15 ma may be obtained from the photoelectric resistors with brief impulse loads.

Figure 10. Overall view of a photorelay with photoelectric resistors: 1 - source of light; 2 - photoelectric resistor.

With such a current power and great amount of operating voltage, and with the sensitive electromagnetic relays being produced by industry, it has proved possible to execute an extremely simple design of the photorelay. The electrical circuit of the latter is shown in Figure 11.

The principle of operation of such a photorelay is very simple. The ohmic resistance of the photoelectric resistor in the dark is great, hence the current in the circuit is slight and inadequate for the response of the electromagnetic relay. When the photoelectric resistor is illuminated, its resistance decreases, in consequence of which the current increases, and when it reaches the required amount, the

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relay responds. The required amount of curren is determined by the selection of the voltage applied and the intensity of illumination.

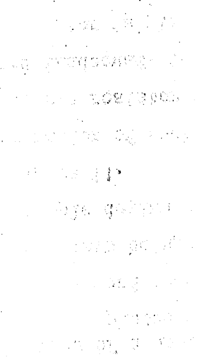

Figure 11. New circuit of a photorelay with photoelectric resistors of the FS-K1 and FS-K2 types: 1 - FS-K1 photoelectric resistor; 2 - ABC-6-270 selenium rectifier; 3 - RKN.U.171.77.52 relay of the telephone type; 4 - 0.25 mkf, 400v condenser.

The new design of the photorelay has yet another interesting feature: it is made up entirely of parts produced by industry. Precisely this circumstance has made possible photoelectric resistors even at the present time in industrial equipment, for example, the FS-K1; the photorelay forms a part of the equipment of the automatic control and sorting machines of the 5AK type produced for the ball-bearing industry.

There has also been produced an automatic machine of the 07B24 type for the line production of nuts, in which photoelectric resistors verify the presence of a hole in the blank prior to the threading.

Experience has shown that one type of electromagnetic relay is inadequate for the solution of a number of specific problems. In view of this fact, the electric industry has developed and produced two new relays of the RKN type. The first of them (rating plate y. 171.78.39) has one group of switching contacts, a response current of 1.7 ma, and a resistance of 18 kom. The second (rating plate y. 171.78.40) is characterized by a smaller response current (1.3 ma), and one group of closing contacts. Its ohmic resistance also is 18 kom.

The availability of three types of electromagnetic relays is a great convenience in the selection of variants of an automatic or control device. In individual instances it is nevertheless necessary to resort to selection of a relay meeting the specific requirements of utilization of a photoelectric resistor.

To illustrate certain interesting instances of employment of the new photorelay design with photoelectric resistors in automatic operation, we may cite two examples from the practice of the "Pechatnyy dvor" /"Hall of Printing"/. A number of rotary presses are equipped with an automatic photoelectric device (Figure 12) which follows the breaking of the strip of paper. Normal operation of the machines is verified by three photoelectric resistors located at different points in the machines and connected in parallel to one relay of the RKN type (rating plate y. 171.78.40). When the paper breaks at any one of the three points, light falls on the photoelectric resistor, its resistance decreases, the relay responds, and the machine stops. A supplementary alternating current relay of the MKU-8 type (rating plate sh. 171.90.

Figure 12. Electric circuit of an automatic device for rotary presses: 1 - FS-K1 photoelectric resistor; 2 and 3 - relays; 4 - selenium rectifier; 5 - condenser; 6 - electric lamps; 7 - contacts to performing member; 8 - paper.

69), the contacts of which permit a load of 1000 voltamperes, has been introduced into the circuit. Under the given conditions, voltage fluctuations in the network are negligible, hence stabilization of the voltage is not required.

The second automatic device with photoelectric resistors (Figure 13), which is installed on DPI platen printing presses, has as its purpose the performance of two functions: that of not permitting two sheets of paper to enter the print and that of not permitting idling of the type without paper. The first task is performed by means of "translumination" of the paper. The circuit is adjusted in such a way that the light passing through one sheet of paper is adequate for holding back the armature of the electromagnetic relay. Upon entry of a second sheet the amount of light on the photoelectric resistor decreases, in consequence of which the current in the circuit of the photoelectric resistor decreases: the relay and the automatic device respond, halting the printing process. The solution to this problem is related to the necessity of stabilizing the voltage feeding the photoelectric resistor and the illuminator lamp. In the instance under discussion, this is accomplished by the ferroresonance method. Although the photoelectric resistor operates under severe conditions, being under a current of 1.3 ma round the clock, there were no perceptible changes in the functioning of the automatic device during three months of operation.

The second task of the automatic device for the platen printing press, not to permit operation when paper is absent, is performed in accordance with the system described above for rotary presses.

There are other automatic devices also executed on the basis of photoelectric resistors and in use in industry: a device for the silvering of radio parts, for checking the filling of bunkers with liquids, friable substances, and blanks, and for checking the absorption of gas.

Of interest is the "rhythmometer" device developed at one of the Leningrad mills. This device, which is related to the operation of two conveyors equipped with photoelectric counters, makes it possible to gain an idea of the execution of the program of production of articles, both for a monthly period and during any interval within the current shift.

The examples cited convincingly point up the existence of important prospects in the matter of utilization of semiconductor photoelectric resistors in automatic photoelectric devices -- an element of automation new to engineering.

Figure 13. Electric circuit of an automatic device for platen printing presses: 1 - FS-K2 photoelectric resistor; 2 - FS-K1 photoelectric resistor; 3, 4, 5 - relays: 6 - selenium rectifier; 7,8 - electric lamps.

In conclusion, it is necessary to sum up the advantages and disadvantages of photoelectric resistors as the principal element of photoelectric automation.

The greatest disadvantage of photoelectric resistors during the present stage of development is the impossibility of utilizing them in precision engineering. The lack of proportion between the photoelectric current and the luminous flux, as well as the temperature dependence, limits the sphere of utilization of photoelectric resistors to tasks related to abrupt transitions from light to darkness and vice versa. The second limitation in the utilization of photoelectric resistors is related to their inertia. In this instance, however, the slight inertia in the photoelectric resistors of the FS-A1 type must be borne in mind.

Among the advantages of photoelectric resistors must be counted: high sensitivity, stability in operation, simplicity of maintenance and feeding, small dimensions and the possibility of situation in places difficult of access, and in individual instances, small inertia and small ttemperature dependence, and the possibility of locating the light receiver at considerable distances from the performing mechanism.

The great advantage of photoelectric resistors and the new type of photorelays for photoelectric automation is the possibility of designing automatic devices without the use of lamp amplifiers and of supplementary development of the production of any of the elements of these automatic devices by supplying them from parts produced by industry.

N. T. ZHAROV

THE QUESTION OF AUTOMATIC CONTROL IN FOUNDRY PRODUCTION

A study carried on by the author over many years of the sequence of operations of foundry and other production processes has led him to the conclusion that in the majority of cases controlled sequences of operations have common structural elements not depending on the nature of the sequence. There have been developed on this basis unified structural systems and equipment common to any sequence of operations, independently of their nature and method of accomplishment. These methods and equipment make it possible to draw up simply, and, what is most important, by following a standard procedure, the control systems of different sequences of operations and select the proper control equipment. We shall discuss the essentials of the proposed method by use of the example of the controlled sequence of operations shown in Figure 1.

It is necessary to pour from the bunker B into the trough VB a specified portion of fine material. The filled trough must be displaced by means of the trolley VT towards the intermediate bunker PB and pour into the latter the contents of the trough by striking of the hook KR against the stop U[1]; the empty trough must be displaced to the initial position (the bottom of the trough itself is closed during the process by the roller RL). The loss of material in the bunker B must be replaced to a specified height H by switching on of the conveyor TR, which carries this material on its belt. Given the process and the equipment by means of which it is carried out, it is necessary to render the process automatic in such a way that it will proceed as follows:

A1 - manually open the slide bar (switch on current in the electromagnet EM, which will turn the handle of the valve KL of the pneumatic cylinder C1);

A2 - automatically close the slide bar (switch off the electromagnet EM).

After this it is necessary to carry out simultaneously two processes:

<u>Process a</u>

B1 - automatically switch on the mechanism for displacement of the trolley VT to the left (switch on D_1);

B2 - automatically switch off the mechanism for displacement of the trolley (switch off D_1);

C1 - automatically switch on the mechanism for counting the time of pouring of the material[2];

C2 - automatically switch off the time counting mechanism;

D1 - automatically switch on the mechanism for displacement of the trolley VT to the right (switch on D_1 in the opposite direction);

D2 - Automatically switch off the mechanism for displacement of the trolley (switch off D_1).

1. The bottom of the trough VB hereupon opens as a result of striking of the hook KR against the stop U, but no action whatever on the part of the control elements are required for this purpose.

2. The necessity for such a mechanism derives from the fact that if the reverse of the trolley VT is switched on immediately after the latter stops, the material will not have time to pour out.

Figure 1. Diagram of automatic proportioning of loose material.

Process b

E1 - automatically switch on delivery of the material from the conveyor TR into the bunker B (switch on D_2);

E2 - manually stop delivery of material into the bunker B after the initial level H has been reached (switch off D_2).

As may be seen, the entire process breaks down into very simple individual actions. We shall call them the <u>elements</u> of the process (elements A1, A2, B1, B2, etc). It is obvious that any controlled process similar to the one under consideration may also be broken down into elements. It will be demonstrated later that specific groups of elements of any controlled process share common rules requiring in the majority of cases identical methods of automation and instruments. We shall arbitrarily designate the manual elements by squares and the automatic elements by circles.

Figure 2 shows a diagram of automation of the process of proportioning of loose material. Here A1 is the beginning of the setting, switching on of the EM; A2 is the end of the setting, switching off of the EM; B1 is the beginning of displacement of the trolley to the left, switching on of D_1; B2 is the end of the displacement of the trolley, switching off of D_1; C1 is the beginning of the pouring; C2 is the end of the pouring; D1 is the beginning of displacement of the trolley to the right, switching on of D_1; D2 is the end of displacement of the trolley, switching off of D_1; E1 is the beginning of the refilling, switching on of D_2; E2 is the end of the refilling, switching off of D_2.

Upon examination of the interconnection among the various elements of any automatic process, it may be seen that it is always specific pairs of elements which condition the progress of any elemental process. For example, the elements A1, A2 (Figure 2), pouring out of portions of the material; B1, B2, displacement of STAT trolley and trough towards the intermediate bunker, etc.

Figure 2. Diagram of the automatic process shown in Figure 1.

We shall designate the actions carried out by individual pairs of elements as the operations of the process. Such operations in the example under consideration are operation A (with elements A1 and A2); operation B (with elements B1 and B2); operations C, D, and E.

The most characteristic factors which determine the system of the process are the connection between the elements of a given operation and the connection between the operations themselves.

Let us discuss the connection between the elements of a given operation. There are three instances differing in principle which are possible in this case.

1. Arbitrary adjustment in manual control, determined by the worker himself. We shall designate it manual connection and arbitrarily designate it in the diagrams by dashes (as between elements E1 and E2 in Figure 2). Manual connection halts the automatic flow of the process; one or more manual elements are required for the resumption of the latter.

2. The form of connection most frequently encountered in practice is one in which accomplishment of the following element is determined by the technological readiness of the operation in question. The indications of readiness may be very diverse. For example, reaching of the required temperature or specified displacements (as between elements B1 and B2, D1 and D2); reaching of a specified weight (as between A1 and A2), color, force of current, etc. The general feature here is that the following element must be carried out after the one in progress only when the process is characterized by a certain fully specified parameter. In automatic devices we shall term such a connection reflex and arbitrarily designate it in diagrams by an unbroken line (Figure 2).

3. Whenever an operation must be carried out within a specific period of time, such a form of connection is used in which execution of the following element of the

given operation is determined only by time (for example, between elements C1 and C2). We shall term such a connection in automatic devices temporal and arbitrarily designate it in diagrams by a zigzag line (Figure 2).

In manual control of a process, the individual operations are connected to one another by arbitrary adjustment, i.e., manual connection takes place in this instance. In automatic control, the various operations must be connected to one another in such a way that the final element of the operation in progress automatically causes the start of the initial element of the following operation (for example, the connection between operations B and C accomplished through elements B2 and C2). The characteristic feature here is the fact that the connection between the elements of the connected operations is not reflex or temporal, but is carried out practically instantaneously or after a brief interval determined by the design of the connecting apparatus.

The time of transition from one operation to another thus does not form part of the working time of the process and is only a transitional, intermediate element. In the diagrams of automatic control, it is expedient in the majority of cases to consider such a connection as instantaneous. We shall term this automatic connection transitional and arbitrarily designate it in the diagrams by a dot between the adjacent elements of connected operations. For example, in Figure 2, the connection between operations A and B, accomplished through elements A2 and B1; the connection between operations C and D, accomplished through elements C2 and D1, etc.

Very often in automatic devices the necessity arises of commencing a certain operation B while another operation A with reflex connection is still being completed. In this instance the commencement of operation B must not be accomplished by the reflex connection mechanism controlling the operation, but by the intermediate condition of operation A itself. We shall term such a connection a transit connection and arbitrarily designate it by a cross as shown in Figure 3.

Figure 3. Designation of a transit connection.

There is drawn up, on the basis of what has been stated above, a so-called diagram of process automation, shown in Figure 2 as applied to the sequence of operations shown in Figure 1.

Generalizing what has been stated, we may draw the following conclusions.

1. Any controlled sequence of operations may be represented graphically in the form of an automation diagram.

2. In order to draw up an automation diagram, it is necessary:

a. to know the process itself, including the sequence of operati STAT

b. to know the equipment and mechanisms by means of which the process is carried out;

c. to decide which elements of the process will be manual and which automatic;

d. establish the forms of connection within the operations and between them.

If these conditions are observed, there need be only one diagram of automation of any process.

In the practical development of diagrams and equipment for an automatic process, the principal difficulties arise in the selection of instruments for automatic connection between the individual operations; for this reason, the development and unification of diagrams and instruments for the accomplishment of temporal and reflex connections are of the greatest moment.

Upon examination of any operation with a temporal connection it may be seen that they are all construced in accordance with an identical principle: the first action is switching on of the temporal connection mechanism itself, with simultaneous initiation of the operation to be executed (beginning of the operation). After this the temporal connection mechanism must keep the operation in progress for the period of time for which the mechanism has been set. The mechanism must subsequently switch itself off and terminate the operation being carried out (end of the operation). Since one operation is automatically followed by another (or others), the functions of the temporal connection mechanism do not end here; it is necessary for the mechanism of the operation in question to switch on the mechanism controlling the following mechanism. Upon completion of the actions indicated, it is also necessary for the temporal connection mechanism to be charged automatically and be ready for action upon repetition of the process.

The functions of the temporal connection mechanism indicated are universal and absolutely identical for any operations with a temporal connection, regardless of the nature of the sequence of operations. The only difference may be that one sequence of operations is initiated by closing the contacts of an electric circuit, another by turning a handle, a third by depression of a pedal, etc.

The creation of a universal temporal connection mechanism which would be capable of carrying out sequences of operations differing in nature is extremely difficult, since in this case a separate mechanism would be required for each operation which would allow for the characteristic features of the latter. However, in any of these instances the functions of the mechanism would remain the same as described above; only its constructional execution would vary. Consequently, if we succeed by a uniform procedure in commencing and terminating any sequence of operations with a temporal connection, it becomes possible in principle to create such a universal temporal connection mechanism. The present-day development of electric drive makes it possible almost always to meet this requirement.

In fact, upon examination of the overwhelming majority of elemental sequences of operations, it is not difficult to convince oneself that almost any of them may be commenced by the simple closing of electric contacts and terminated by opening of the contacts.

Time relays are utilized in engineering as the instruments which accomplish a specific time lag; in accordance with this, we shall call the universal time connect- ion mechanism the standard time relay and arbitrarily designate it by the letters RV.

A conventional diagram of the standard time relay is shown in Figure 4. The coil of the RV is the time indicator; the contacts 3RV are block contacts; P are the starting contacts; 1RV, the operating contacts of the relay; and 2RV, the contacts for starting the following relay. The standard time relay, as follows from what has been stated, must perform the following six functions (Figure 5): 1 - starting of the relay RV by brief closing of the contacts P, manually or by the preceding relay; 2 - commencement of operation A by closing of the contacts 1RV; 3 - counting off of the time of duration of operation A, keeping the contacts 1RV closed; 4 - termination of operation A, opening the contacts 1RV; 5 - switching on of the relay of the fol- lowing operation, B, by brief closing of the contacts 2RV for 0.5 to 0.6 second; 6 - automatic charging of the relay, preferably instantaneous.

Let us examine the overall diagram of an automatic process with a temporal con- nection effected by means of the relay described. Let us say that it is necessary to accomplish an automatic cycle consisting of two operations, A and B, with a time relay.

The automation diagram of this process is shown in Figure 6, where RV1 is the time relay controlling operation A and RV2 is the time relay controlling operation B.

The electric circuit for carrying out the automatic cycle indicated is shown in Figure 7.

Figure 4. Conventional electric circuit of a standard time relay.

Figure 5. Diagram of required functions of the standard time relay.

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Figure 5. Diagram of required functions of the standard time relay.

Figure 6. Diagram of automation of a standard cycle of two operations with temporal connection.

Figure 7. Electric circuit of a standard cycle of two operations with temporal connection.

The cycle begins with depression of the starter button BP; the coil of RV1 is hereupon excited (the time relay RV1 is switched on), and at the same time, the contacts 3RV1 and 1RV1 are closed. The contacts 3RV1 block the circuit of the relay coil, and the contacts 1RV1 initiate operation A (element A1).

The duration of operation A is the time τ_1 for which the time relay RV1 is set. Upon the lapse of this interval, the contacts 3RV1 and 1RV1 open and the contacts 2RV1 close for 0.1 to 0.5 second. When the contacts 3RV1 open, the coil of RV1 is deenergized and the relay charges automatically; when the contacts 1RV1 are opened, operation A (element A2) is terminated, and after the contacts 2RV1 have been closed briefly, relay RV2, which controls operation B, is switched on. The contacts 3RV2 and 1RV2 hereupon close. The contacts 3RV2 block the coil of RV2 (the contacts 2RV1 open after this), and the contacts 1RV2 initiate operation B (element B1).

If the duration of operation B equals the time τ_2 (the relay RV2 is set for this time), upon the lapse of this interval, the contacts 3RV2 and 1RV2 open and the contacts 2RV2 close for 0.1 to 0.5 second. When the contacts 3RV2 open, the coil of RV2 is deenergized, and the relay charges automatically; when the contacts 1RV2 open, operation B (element B2) is terminated, after the contacts 2RV2 have been closed briefly, the relay RV1 is switched on, and the entire cycle begins to repeat itself.

Let us examine process with reflex connection. It was established above that a reflex connection between elements is characterized by the technological readiness of the automatic operation being performed. Generally speaking, the indicatiSTAT of readiness may be very diverse (reaching of a specific weight, color, length, temperature, power of current, speed, etc), and the reflex connection mechanisms may be different in all these instances. However, in the majority of processes with automatic control, the technological readiness of the process can be reduced to a specific position in space of the moving system or object. For this reason, the creation of a single procedure and uniform equipment for the accomplishment of reflex connection of this type in any processes is of particular interest. We shall term this type of reflex connection <u>mechanical reflex connection</u>, in contrast to those of temperature, electrical connections, and others.

We shall attempt to develop a single procedure and uniform equipment for the accomplishment of mechanical reflex connection. On the analogy of the time relay which effects the temporal connection, we shall term this mechanism the standard electromechanical reflex connection and conditionally designate it by the letters RR. It was demonstrated above that in the majority of cases any operation can be accomplished by the closing and opening of electric contacts. The reflex relay also is constructed on this basis.

Let us examine the various operations with a mechanical reflec connection. It may be seen from comparison of the operations with a mechanical reflex connection (Figure 2), that they are all constructed on one principle and in all cases the reflex relay (Figure 8), just as the standard time relay, must carry out the following six functions (Figure 9): 1 - starting of the RR relay by brief closing of the contacts P, manually or by the preceding relay; 2 - initiation of operation A by closing of the 1RR contacts; 3 - keeping the 1RR contacts closed until the 4RR contacts are set in action by the mobile system controlling the process; 4 - termination of operation A by opening of the 1RR contacts; 5 - switching on of the relay of the following operation, B, by brief closing (for 0.5 -- 0.6 second) of the 2RR contacts; 6 - automatic charging of the relay, preferably instantaneous. The only difference here is that with the reflex connection, the duration of the operation is not determined by time adjustment but by the action of a certain mobile system on the appropriate contacts of the reflex relay (contacts 4RR), which thereby terminate operation A and switch on the relay of the following operation, B.

The relay functions as follows (Figure 8). It is switched on by brief depression of the starter button P for $\tau = 0.1 - 0.5$ second, this being accomplished in manual operation by depressing the starter button and in automatic operation by brief closing of the contacts of the relay of the preceding operation. After an interval of time $\tau_1 < \tau$, the block contacts 3RR must respond, and the circuit of the contacts P may be opened. At the same time the relay is switched on, the contacts 1RR are closed, and the circuit of operation A is switched on, during which latter the contacts 1RR remain closed and the contacts 2RR open. The contacts 4RR are separated in space from the relay, being located at appropriate points of automatic adjustment.

When operation A is completed, a certain mobile system mechanically closes the contacts 4RR for a period of time τ_2 greater than the time τ_1 of response of the block contacts 3RR. The latter hereupon open and upon subsequent closing of the 4RR contacts, the relay is not switched on. At the same time, the contacts 2RR are closed for the period of time τ and switch on another reflex relay or time relay. After this the relay must be charged automatically.

A diagram of automation of a standard cycle of two operations with a reflex connection is shown in Figure 10, and the electric circuit in Figure 11. The functioning of the cycle is accomplished as follows:

When the starter button P is depressed, the coil of the RR1 relay is energized; the 3RR contacts, having blocked the coil circuit, as well as the 1RR1 contacts, having switched on the circuit of operation A (element A1), hereupon close. When operation A is completed, a certain moving system presses against the head of the 4RR1 contacts (element A2), and the latter open. The RR1 coil is hereupon deenergized, the 3RR1 contacts open, and the RR1 relay is switched off (the 4RR1 contacts may be released); at the same time, the 2RR1 contacts are closed for 0.1 to 0.5 second, switching on the RR2 relay. The RR2 coil is hereupon energized, the 3RR2 blocking the coil circuit are closed, as well as the 1RR2 contacts, which switch on the circuit of operation B (element B1); the RR1 relay must be automatically charged. When operation B is completed, a certain moving system presses against the head of the 4RR2 contacts (element B2), and the latter open. The RR2 coil is hereupon deenergized, the 3RR2 contacts open, and the RR2 relay is switched off (the 4RR2 contacts may close); at the same time, the 2RR2 contacts close, the RR1 coil is energized, the 3RR1 contacts blocking the coil circuit close, as well as the 1RR1 contacts, switching on the circuit of operation A (element A1); the RR2 relay must be automatically charged, and the cycle will be repeated.

Figure 8. Conventional electric circuit of the standard mechanical reflex relay.

Figure 9. Diagram of the required functions of the standard mechanical reflex relay.

In practice, any controlled sequence of operations contains operations with both the reflex and the temporal forms of connection (composite connection). Upon investigation of the function of the standard time relay and the reflex relay, it may easily be seen that they are capable of being combined with one another. If the system of automation of a process with composite connection is known, the electric system of its control circuits is drawn up in accordance with the standard procedure, following the automation diagram, as is the case with processes with only temporal or only reflex connection.

Figure 12 gives a diagram of automation of a process with composite connection, and Figure 13 the electric diagram of the control circuits of the process.

It may be seen from inspection of the principles of operation of the standard reflex relay and time relay that their functions differ only in the fact that the operation of the former is initiated by external switching contacts, and that of the latter by internal switchings. For this reason it has proved possible to create a single standard reflex-time relay which performs the functions of a standard reflex relay when an external terminal switch is attached to it, and the functions of a standard time relay when a time indicator mechanism is connected to it.

Figure 10. Diagram of automation of a standard cycle of two operations, with a reflex connection.

Figure 11. Electrical diagram of a standard cycle of two operations, with a mechanical reflex connection.

Let us assume that RR(V) is the device which performs the functions of both relays (Figure 14). Then, on the basis of the diagrams shown in Figures 4 and 8, it should operate as follows.

1. Upon brief closing of the starting contacts P (manually or by the contacts of the preceding relay), the relay is set in operation, and the circuit of the contacts P is blocked by the internal contacts of this relay. The circuit of the contacts P may be opened after this.

2. At the same time, the external working contacts of this relay, i.e., 1RR(V), close, and the operation, A, begins.

The first phase of operation of the relay concludes with this, and the contacts 1RR(V) are closed until the contacts 4RR close or open for some reason, depending on the design of the RR(V) device. This may be accomplished in two ways. If the relay is utilized as a reflex relay, the 4RR contacts may be closed and opened by an external terminal switch VK in accordance with the reflex index discussed above. In this case the system, which consists of the RR(V) device and the external switch VK, represents a standard reflex relay.

If the 4RR contacts are closed (or opened) after a strictly specified interval of time as a result of connection of a special time indicator mechanism MOV to them, the system, which consists of the RR(V) device and the MOV mechanism, is transformed into a standard time relay.

Figure 12. Diagram of automation of a process with composite connection.

Figure 13. Electric diagram of control circuits of the process shown in Figure 12.

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Figure 14. Principle of operation of the standard relay: 1 - network; 2 - start; 3 - to operation control circuit; 4 - to following relay.

The following functions of the relay may be identical in both instances:

a. opening of the 1RR(V) contacts, i.e., termination of operation A;

b. simultaneous closing, for 0.1 to 0.5 second, of the 2RR(V) contacts, which close the starting contacts P of the following relay or any other starting circuit;

c. automatic, and preferably instantaneous, charging of the relay, after which it will again be ready for repetition of the cycle.

Consequently, one type of relay, specifically, the standard reflex relay, is necessary for accomplishment of automatic control in accordance with the proposed procedure. The standard time relay may be obtained by attachment of a special time indicator mechanism to the reflex relay. This fact is of great practical significance.

Despite the extensive employment of time relays in engineering and the diversity of their designs, not one type of relay performs the six functions of the standard time relay indicated above (Figure 4). The creation of a special standard time relay, although possible, requires special production, and this is advisable only under conditions of mass industrial production and is extremely difficult when such relays are produced individually, even by well mechanized plants and shops. Moreover, the standard time relay obtained is rather complex in design.

There are many types of time relays which may be utilized without any modifications as the time indicator mechanism (MOV in Figure 14) connected to the reflex relay.

Let us examine several examples of the design of standard reflex relays and time indicator mechanisms.

Figure 15 shows a diagram of one of the versions of a reflex relay. It includes: a contactor coil L1 with contacts 1L1, 2L1, 3L1; a contactor coil L2 with contacts 1L2, 2L2; a selenium rectifier VP; and two condensers C1 and C2. The L1 and L2 coils may be of any make (for example, the MKU-48), depending on the nature of the supply current and the desired power of the contacts. This holds true of the other parts also. When the relay is supplied with direct current, the necessity for the rectifier VP and the condenser C1 is eliminated.

The relay operates as follows. When the starting contacts P have been closed

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Figure 15. Diagram of a standard reflex relay -- version 1.

briefly, the coil L1, which is fed rectified current, is energized. The condenser C1 with a capacity of 30 mkf serves as a filter for smoothing out current pulsations. When the current enters the coil L1, the contacts 1L1, 2L1, 3L1 close. The contacts 1L1 block the circuit of the coil L1 (the circuit of the contacts may now be opened); the operation to be carried out is initiated by contacts 2L1. The relay remains in this position until closing of the circuit of the 4RR contacts. If they are closed by means of an external switch, the mechanism described performs the function of a reflex relay; if they are closed after a specified interval of time by a time indicator mechanism, the system is transformed into a standard time relay. In both instances, when the circuit of the contacts 4RR is closed, the coil L2 is energized, and the condenser C2 is charged practically instantaneously. However, as soon as current flows through the coil L2, the 1L2 contacts open and the 2L2 contacts close. When the 1L2 contacts open, the coil L1 is deenergized, and its contacts 1L2, 2L1, 3L1 open; the operation is hereupon terminated by the contacts 2L1, and the coil L1 is deenergized by the contacts 1L1. The starting circuit of the following relay (or another control circuit) is switched on by closing of the 2L2 contacts. The coil L2 is under current only for an instant, since the contacts 1L2 open and the coil L1 is switched off immediately, for which reason the contacts L1 open and switch off the coil L2. Hence, if it were not for the condenser C2, the contacts 2L2 would hereupon close for an instant and the following relay would not have time to switch on; but since at the moment of switching on of the coil L2 the condenser C2 is charged, after the former is switched off (upon closing of the contacts 1L1), the latter will be discharged through the same coil, L2, and its magnetic field will not disappear immediately; consequently, the contacts 2L2 will be closed for a certain period of time (which is the longer, the greater is the capacity of the condenser C2) adequate for switching on the following starting circuit. After this, all the relays occupy the initial position (as in Figure 15), and upon closing of the starting contacts P, the process described is repeated. A resistor R which reduces the initial amount of charge current is connected into the circuit of the condenser C2 in order to reduce sparking of the contacts 4RR as they close. The relay functions accurately and reliably, and may be recommended for industrial production.

As for the time indicator mechanism (in the event the relay is utilized as a standard time relay), its design may vary, being determined chiefly by the amount of time lag desired. An electronic time relay of the ERV-99 type, a diagram of which is shown in Figure 16, may be used for some time lags (on the scale of one minute). The terminals a_1, a_2, A and B are connected to the corresponding terminals of the reflex relay (Figure 15).

Figure 16. The ERV-99 time relay.

Figure 17. The MRV-26 and MRV-27 time relays.

For larger amounts of time lag, on the scale of several minutes, motor time relays, for example, those of the MRV-26 and MRV-27 types, a diagram of which is given in Figure 17, may be used as the time indicator mechanism. The terminals a_1, a_2, A and B are connected to the corresponding terminals of the reflex relay.

On the basis of the procedure discussed above of forming automatic control circuits and equipment, systems have been developed of automatic plants for the basic sequences of operations of foundry production (loading cupolas, preparation of moulding mixtures, knocking out of castings from mould boxes of various overall sizes with crosspieces, etc).

Given below as an example is a brief description of the installation and functioning of an automatic plant for the preparation of a moulding mixture on the FB-2 edge-runner mill, produced and operating at the department of foundry production of the Urals Polytechnic Institute.

The plant (Figure 18) consists of three separately mounted assembly blocks:

1. block of mechanisms (edge-runner mill, batchers, and the drives for them);

2. block of control equipment (standard relays, coils, buttons);

3. block of supply and electric equipment of performing circuits (rectifiers, magnetic starters, etc). The block of mechanisms is mounted on a welded metal

frame 1, to which is secured by screws at a distance of 500mm from the floor, the STAT edge-runner mill together with its drive. Secured immediately beside the edge-runner

Figure 18. Overall view of the automatic plant for the preparation of moulding mixtures on the FB-2 edge-runner mill: I - block of mechanisms; II - block of control equipment; III - supply block.

mill is the mechanism 3 for unloading the finished mixture, which consists of an electromotor with reduction gear, a shaft, and a door opening downwards. Two bunkers with a common outer jacket 4 having an inner partition are mouhted on the frame above the edge-runner mill. The left bunker is used for the burnt mixture, the right one for sand. The bunkers are provided with screw batchers 5 and 6 which are set in operation by electromotors with reduction gears 7 and 8. The motors and reduction gears are mounted in one case. Water with a small quantity of clay (clayey suspension) is fed from a tank 9 by means of an electromagnetic valve 10. The finished mixture is unloaded by downward opening of the unloading door 11 by an electromotor 12 through a two-stage worm reduction gear.

The diagram of automation of the plant is given in Figure 19a, the electric diagram of the control circuits in Figure 19b, the electric diagram of the performing circuits in Figure 19c, and the diagram of the electric equipment of the edge-runner mill in Figure 19d.

For verification of the functioning of the plant, 8 signal lamps have been mounted in the case of the block of performing circuits, in accordance with the number of operations; the appropriate sign has been placed above each of them: "sand",

Figure 19. Diagram of operation of the FB-2 automatic laboratory edge-runner mill: a - diagram of automation; b - electrid diagram of control circuits; c - electric diagram of performing circuits; d - diagram of electric equipment of edge-runner mill; 1 - (switching on of E1); 2 - sand; 3 - (switching off of E1); 4 - switching on of E2; 5 - burnt mixture; 6 - (switching off of E2); 7 - (zero element); 8 - dry mixing; 9 - (zero element); 10 - (switching off of EM); 11 - water; 12 - (switching off of EM); 13 - (zero element); 14 - wet mixing; 15 - (zero element); 16 - (switching on of batcher to the right); 17 - opening of unloading window; 18 - (switching off of batcher); 19 - (zero element); 20 - unloading; 21 - (zero element); 22 - (switching on of batcher to the left); 23 - closing of unloading window; 24 - (switching off of batcher); 25 - feeding of sand; 26 - feeding

of burnt mixture; 27 - feeding of water; 28 - unloading; 29 - motoISTAT edge-runner mill.

"burnt mixture", "dry mixing", "water", "wet mixing", "opening of door", "unloading", and "closing of door". When an operation is initiated, the lamp corresponding to it burns. This type of verification has proved very useful, since the burning of the lamp attests to the fact that the relay of the operation in question has functioned normally. In production installation, it is the most advisable to locate the signal lamps at the control point of the foreman of the mixture preparation section.

In addition to automatic control of all the operations of the process, provision is made for the manual control of any operation by the installation of eight starting buttons correspondingly connected to the starting contacts of each relay.

If for any reason (malfunction or temporary switching off of the electric network) the process has been interrupted, it may be resumed by depressing the corresponding button, starting with any operation desired; this cannot be done when timers are used. Switches with signs indicating each operation are connected in parallel to the terminals 1RV1 (feeding of sand), 1RV2 (feeding of burnt mixture), 1RV4 (feeding of water), 1RR1 (opening of unloading door), and 1RR2 (closing of unloading door). By closing these switches it is possible to carry out any operation manually, if a relay is out of order or in case of other malfunction in the automatic control circuit. In these cases the equipment is not idle during adjustment of the automatic system, and the sequence of operations may be carried out by manual control.

An electromagnetic pulse counter (not shown in the diagram) the coil of which is connected into a direct current dircuit of 24 volts through contacts 2RV1 is mounted in the plant. These contacts are not used in the automatic control system.

The plant functions accurately and reliably and with an adequately wide adjustment range for different rates of operation.