Deceleration system for kinematic linkages of positioning

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Abstract. Flexible automation is used more and more in various production processes, so that both machining itself on CNC machine tools and workpiece handling means are performed through programming the needed working cycle. In order to obtain a successful precise positioning, each motion degree needs a certain deceleration before stopping at a programmed point. The increase of motion speed of moving elements within the manipulators structure depends directly on deceleration duty quality before the programmed stop. Proportional valves as well as servo-valves that can perform hydraulic decelerations are well known, but they feature several disadvantages, such as: high price, severe conditions for oil filtering and low reliability under industrial conditions. This work presents a new deceleration system that allows adjustment of deceleration slope according to actual conditions: inertial mass, speed etc. The new solution of hydraulic decelerator allows its integration to a position loop or its usage in case of positioning large elements that only perform fixed cycles. The results being obtained on the positioning accuracy of a linear axis using the new solution of the hydraulic decelerator are presented, too. The price of the new deceleration system is much lower compared to the price of proportional valves or servo-valves.

1. Introduction

The dynamic running duty of a kinematical linkage is a complex process resulting from mechanical and hydraulic phenomena, interdependent and non-stationary. Practically, the transient duty will result from the variation of the mechanical factors such as torque (force), resultant inertia moment and hydraulic factors such as pressure, flow, hydraulic resistance [1]. The variation of the mechanical factors is determined by the functioning of the kinematical linkage itself and the variation of the hydraulic factors results from the deliberate intervention on the hydraulic control: start-up, stop, reversal, speed variation of the moving element. Transient phenomena are subject to interest through the fact that there are few situations when a permanent functioning may be considered. In case of the kinematical linkages used for manipulators, robotized cars, pallet mechanisms, as well as of tool changing on machine tools, conditions are imposed in terms of the quality of the transient duty [2]. The consequences of the transient phenomena will be reflected not only on the functioning quality, but on the sizing of several elements too. Transient phenomena are also subject to interest from the productivity point of view where, through the quality of the transient duty the speed rates of the moving elements might be increased, thus shortening the time of the manipulating cycles. At the same time, the start-up and stop durations are also affecting the duration of the manipulating cycles. The theoretical and experimental analysis of the transient phenomena allows the choice of the optimal automation procedure for start-up and braking as well [3]. The general solution to the dynamic problems is very complex if it is to consider every aspect that influences the running of the hydraulic circuit (thermal variations, oil viscosity variation, fluid compressibility, pipe elasticity etc.). With respect to the direct variation of the running conditions of an element belonging to the hydraulic circuit, such as the sudden variation of the pump flow or of the external load, this will lead to a rapid variation of the fluid pressure and speed [4].
2. Deceleration Transient Process of the Moving Elements

In order to avoid the over oscillations while stopping the kinematical linkages, a deceleration transient duty is used that allows the gradual decrease of the speed of the moving element. Practically, the transient duty of the kinematical linkage deceleration needs to comply with two major conditions [5]:

- To shorten the transition time up to achieving the positioning;
- To eliminate or decrease the oscillations caused by the transient duty.

Since, in case of the kinematical linkages of positioning used on work-piece and tool manipulating mechanisms, it’s only the positioning accuracy, free of over oscillations, that is required, the transition time (i.e. the duration of the deceleration process) becomes a secondary condition. The hydraulic drives currently use two types of deceleration, as presented below: In case of the first type, the initial speed \( V_1 \) of the moving element is first decreased to \( V_2 \) then to \( V_3 \) that is a “creeping” speed and provides the positioning to the buffer, Figure 1. This deceleration at fixed speed steps is achieved, in constructive terms, through the structure of the hydraulic motor and the distances (time rates) when the passage to a speed rate to another is done, so that oscillations should not come up during the speed switching. In case of the second type of deceleration, Figure 2, the deceleration is performed through the continuous passage from the first speed \( V_1 \) to the last speed \( V_6 \) that is the creeping speed that provides the stop at the buffer. The profile of the deceleration diagram may exponential, linear, hyperbolic etc.

![Figure 1. Transient duty of deceleration at fixed speed steps.](image)

![Figure 2. Transient duty of deceleration at continuous speed.](image)

This type of deceleration eliminates the possibility of speed reversal at the final stage of positioning. At the same time, this theoretically settled profile of the deceleration diagram should allow, on its turn, the adjustment of the deceleration slope through experimental trials. If the kinematical linkage is designed to perform positioning in closed loop (canned cycles), the position taking speed (creeping speed) should be limited so that the inertia of the kinematical linkage will not exceed the final point of the needed position [1]. In case of the kinematical linkages designed to manipulate light and medium weights, the linear deceleration will be used, because it has performances superior to the exponential deceleration, but imposes higher requirements regarding the idle stroke of the kinematical linkage [2].

3. Settling the Parameters of the Transient duty of Deceleration of the Moving Elements

It is necessary to establish the values of deceleration, space and time of the exponential transient duty, in order to determine the maximal deceleration that, on its turn, imposes the maximum value of the pressure into the circuit and takes part to sizing the components of the hydraulic circuit as well. By considering that the motion of a moving element is performed at the speed \( V_1 \) and at a moment the speed modification into \( V_t \) (that is the creeping speed) is necessary, Figure 3, the speed variation in the exponential transient duty will be described by the following equation:
\[ V = A + B e^{-\beta t} \]  \hspace{1cm} (1)

The symbols mean: \( A, B \) - constants; \( \beta \) - constant named passage range. The constants \( A \) and \( B \) are used to locate the beginning and the end points of exponential process, which inclination is assigned by constant \( \beta \). The constants \( A \) and \( B \) can be calculated from the limit conditions:

\[
\begin{align*}
    t &= 0; \quad V = V_i \quad \text{respectively}, \quad \lim_{t \to \infty} V = V_t
\end{align*}
\]  \hspace{1cm} (2)

It will result:

\[ A = V_t; \quad B = V_i - V_t \]  \hspace{1cm} (3)

\[ V = V_t + (V_i - V_t)e^{-\beta t} \]  \hspace{1cm} (4)

In this way the time needed for the moving element to go through the transient duty may be determined:

\[ e^{-\beta t} = \frac{V - V_t}{V_i - V_t} \rightarrow t = \frac{1}{\beta} \ln \frac{V_i - V_t}{V - V_t} \]  \hspace{1cm} (5)

The space \( S \) described by the moving element during the transient duty will be:

\[ S = \int_0^t V dt = \int_0^t [V_t + (V_i - V_t)e^{-\beta t}] dt \]  \hspace{1cm} (6)

\[ S = V_t t - \frac{1}{\beta} (V_i - V_t)(e^{-\beta t} - 1) \]  \hspace{1cm} (7)

Based on the relations above, the time and space of the moving element described during the deceleration process may be calculated and, by deriving the speed, the deceleration of the moving element as well as the identification of the deceleration maximum value necessary for the sizing process, may be determined.

4. Design wise and Functional Presentation of the Hydraulic Decelerator

The hydraulic decelerator is composed of the body 1, Figure 4, inside of which the plunger 2 is located that can be moved to the right through the left side piston 4 by means of the bolt 4, guided into the flange 5. The body of the left side piston 6 supports the adjustable mechanical buffer 7 that is sealed through the seals 8 and locked through the nuts 9 and 10. The body of the left side piston 6 is fixed to the body 1 and inside it the supply system is located. The supply system is composed of the hydraulic resistor 12, the bypass valve composed of the ball 13, the helical spring 14 and the adjustable stopper 15. The motion to the left of the plunger 2 is performed by the right side piston 16, whose force is transmitted through the bolt 4.

The integration diagram of the hydraulic decelerator is shown at Figure 5, where the hydraulic decelerator \( DH \) performs the deceleration of the hydraulic motor \( MR \) by means of the distributor \( D_2 \).
The hydraulic motor $MR$ rotates to a direction or another through the distributor $D_1$ and the stopping accompanied by deceleration is performed upon the receipt of the control signal that changes the direction of rotation.

**Figure 4.** The hydraulic decelerator: (a) - side view; (b) - section by the plane B-B; (c) - longitudinal section by the plane A-A. 1 - body; 2 - plunger; 3 - piston; 4 - bolt; 5 - flange; 6 - left side; 7 - mechanical buffer; 8 - seals; 9, 10 - nuts; 11 - screw; 12 - hydraulic resistor; 13 - ball; 14 - helical spring; 15 - adjustable stopper; 16 - right side piston; 17 - right side cylinder; 18 - screw.
position of the distributor $D_2$ so that the right side cylinder 17 (Figure 4) is supplied and the oil removal from the cylinder 6 is performed by the hydraulic resistor created by the thread pin 12 that allows the adjustment of the speed of the plunger 2 and, by default, the progressive shut-off of the passage section of the main flow. While performing a positioning, as shown in the speed – space diagram, Figure 5, the moving element moves at the speed $v$ and at the distance $l_2$ it begins to decelerate up to the creeping speed $V_t$ that is adjusted through the adjustable buffer 7, Figure 4. Afterwards the control signal for stopping sets the distributor $D_1$ at the median position.

The deceleration slope has a quasi-exponential shape due to the inclination of the supply hole at the angle $\alpha$ and the inclination of the deceleration slope may be done through the thread pin 12 (Figure 4) within the limits marked by the interrupted line on the speed – space diagram, Figure 6. The bypass valve composed of the ball 13 and the helical spring 14 (Figure 4), assures the fluid to bypass the hydraulic resistor composed of the thread pin 12, in order to increase the speed of the plunger 2 during its motion in reverse direction, needed to obtain a high acceleration of the moving element.

5. Experimental Results
Trials have been carried out on a pallet mechanism whose rotation motion is generated by a hydraulic motor. Its hydraulic control diagram is shown at Figure 4. The decelerator described above has been included to the hydraulic circuit as well. For trials, the positioning of the pallet mechanism with unidirectional rotation mechanism has been used, by approaching the same deceleration slope. On the pallet mechanism one or two pallets of size 800 x 800 mm, loaded with various weights, have been located. The rotation speed of the pallet mechanism has had different rates and the creeping speed has had two rates: 0.5 and 0.3 rpm, respectively.

The related results are shown at Table 1, where a good repeatability and a behavior free of deceleration shocks may be noticed. The deceleration time has light variations in relation to the load weight on the pallet that confirms the controlled deceleration effect imposed to the pallet mechanism.
Table 1. Experimental data of the trials.

| Pallet size (mm) | Number of pallets on the mechanism | Load on the pallet (kgf) | Rotation pallet (rpm) | Creeping (rpm) | Positioning repeatability (°) | Deceleration time (s) |
|-----------------|----------------------------------|--------------------------|-----------------------|----------------|-------------------------------|----------------------|
| 800 x 800       | 1                                | 200                      | 8                     | 0.5            | 0.021                         | 1.2                  |
| 800 x 800       | 2                                | 500                      | 7                     | 0.5            | 0.023                         | 1.24                 |
| 800 x 800       | 1                                | 700                      | 5                     | 0.3            | 0.021                         | 1.27                 |
| 800 x 800       | 2                                | 200                      | 8                     | 0.5            | 0.0271                        | 1.32                 |
| 800 x 800       | 2                                | 500                      | 7                     | 0.5            | 0.0276                        | 1.35                 |
| 800 x 800       | 2                                | 700                      | 5                     | 0.3            | 0.0279                        | 1.36                 |

6. Conclusions
It may be noticed that the kinetic energy of the moving element and of the fluid is completely accumulated under the form of potential energy into the elastic system composed of the piping and fluid, reified in the pressure of the compressed fluid of inside the pipes between the hydraulic motor and decelerator. The generation of a decompression wave will follow, that will move towards the distributor. Such waves continue to move to one or other direction, being accompanied by the interchanges of the kinetic and potential energies until the friction dissipates the energy involved into the transient duty. It results that the most efficient way to reduce the generation of overpressure in this transient process is to design the pipe network so that the fluid speed is low, i.e. using pipes of large cross section. If the maximum admitted speed of the fluid into the pipes is settled at 5 m/s, the overpressure that occurs upon the sudden shut-off of a valve will be approx. 50 bar. This is generally deemed an acceptable design value and is a criterion for sizing the piping. The presence of the decelerator into the hydraulic circuit assures a decrease of the overpressure level and, by default, a motion free of shocks in the deceleration process. The overpressure level is directly influenced by the value of the deceleration time, in other words by the value of the deceleration slope. The decelerator presented in this work features the possibility for adjusting the deceleration slope in function of the load value and the speed of the moving element at stationary duty, as well. The same solution concerning the deceleration transient duty may be applied in case of the linear hydraulic motors as well, in order to perform intermediary positioning inside the piston stroke.

7. References
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