Heat and Technology
Vol. 32, n. 1-2, 2014

ANALYTICAL MODEL OF A MACHINING CENTRE VERTICAL AXIS WITH DYNAMIC WEIGHT COUNTERBALANCING BY A PNEUMATIC PROPORTIONAL VALVE

Giorgio Olmi*

* Department of Industrial Engineering (DIN), University of Bologna, Viale del Risorgimento 2, 40136 Bologna, Italy

ABSTRACT

High Speed Machining is getting more and more important in order to fulfill the goal of a reduction of manufacturing times and costs, together with an increase of quality. The design of modern Machining Centres is quite a difficult task, as high forces of inertia arise from high speeds. Thus, the development of simulative mathematical models can be of a great support for the designer. This paper deals with the development of a Simulink model of the vertical axis of a machining centre. Full details are provided on the mechanical and pneumatic devices it is composed of, in particular regarding the architecture and the operative behaviour of a pneumatic proportional valve. This is important device has the basic role of dynamically counterbalancing the spindle head weight, maintaining the resulting force at a constant value, independently of the applied motion. Full details on the algebraic and differential equations to be implemented are provided throughout the paper. The results proved the efficiency of the pneumatic device and its short response time. Processing the simulative outcomes led to the identification of an equivalent second-order dynamic system. A simplified Simulink model could therefore be developed accordingly and used to simulate the response to whatever motion law applied to the head.

KEYWORDS

Machining Centre, Pneumatic proportional valve, Pneumatic cylinders, Weight counterbalancing, Dynamic response

1. INTRODUCTION

High Speed Machining (HSM) is becoming one of the most promising advanced manufacturing technologies, as it enables a reduction of manufacturing times and of product costs, and, at the same time, an increase in quality. Design techniques have evolved over the last few years to deal with the dynamic aspects of design: the forces of inertia, due to the moving masses, are nowadays the main issues to be considered in the structural analysis. Therefore, the development of virtual models for the simulation of the dynamic behaviour of a part or of the entire Machining Centre (MC) is getting more and more important [1-6]. The development of numerical models simulating the dynamic behaviour can be framed within the increasing demand for numerical and analytical models, supporting the study of machine performance. Nowadays, these models can be regarded as powerful tools that can significantly support designers. Their main feature consists in the possibility of predicting the mechanical response, from many points of view. For instance, the design of pneumatic circuits for automatic machines is highly supported by models simulating the entities of flows and pressures, and the duration of transient times [7]. In the field of structural mechanics, Finite Element and analytical models can provide the stress/strain distribution [8-9]. Integrated experimental and numerical models have a great importance at predicting the fatigue response [10-15]. However, a critical issue in the development of virtual models stands in the determination of the numerical parameters to be introduced and processed. For instance, the determination of stiffness or of the coefficients of friction and viscosity is really crucial in the development of models for the simulation of the dynamic response after an impulsive load. The estimation of pneumatic conductances is the first step to start the simulation of a pneumatic circuit [7]. An accurate determination of the correct numerical parameters is the basic requirement to obtain reliable result from the quantitative point of view. The most suitable strategy for the determination of the aforementioned parameters usually stands in the execution of experimental tests and measurements. Regarding this issue, many examples can be found in literature, dealing with experiment-based models and with the integration between experimentation and simulation [10-25].

This paper is focused on the development of a numerical model that simulates the dynamic response of a portion of a MC, where pneumatic and mechanical devices are integrated. The conducted analysis regards the dynamic behaviour of the vertical axis (z-axis) of a MC, where a pneumatic system is specifically designed to counterbalance the weight of the spindle moving head. The most difficult challenge consisted in the modeling of a compensation valve: it has the important function of maintaining constant the value of the air pressure that sustains the moving head, even when it shifts at high speed. The implemented mathematical model accounts for the physical laws describing both air flow and the conversion of pressure into the reacting force that balances weight. The mechanical devices are modeled, considering their actual stiffness and damping properties.

The goals of the present paper can be summarized in the following points:

- Determining the dynamic response of the z-axis system to motion laws applied to the head, e.g. upward or downward step displacements, and ramp motions.
- Discussing the efficiency of the pneumatic valve, in particular, if the head weight can be dynamically counterbalanced with a sufficiently short response time.
- A final aim consisted in the identification of the parameters (stiffness and damping) of an equivalent second-order system, having the same dynamic properties of the studied device.

2. THE PNEUMATIC-MECHANICAL SYSTEM AND THE PROPORTIONAL VALVE

The here considered MC is a high speed milling machine with five controlled feed drives. The mechanical system along the z-axis consists of the vertical head with electromagnetic mandrel and of the related feed drive system by recirculating ball screw. The pneumatic force is transmitted by a couple of pneumatic cylinders. The cylinders are designed so that the weight of the moving head and of the spindle can be sustained by the pressure acting on their pistons. A scheme of the general layout is visible in Fig. 1: the pneumatic circuit is completed by air supply devices and pipes.

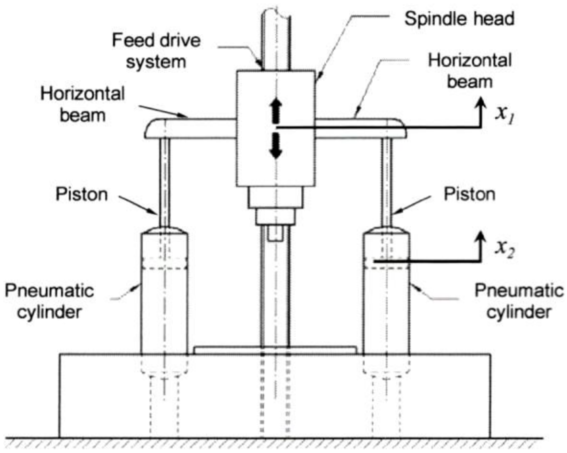

Figure 1: General layout of the pneumatic-mechanical system: the moving head sustained by pneumatic cylinders.

The most important component of the pneumatic circuit is a proportional valve connected to the machine frame, whose layout is sketched in Fig. 2. This is specifically designed to dynamically counterbalance the weight of the aforementioned parts. The important role of the valve can be clarified with reference to the typical operation tasks of the z-axis of the MC. The pressure value in the cylinder downward chambers is initially set to 6.2 bar (absolute pressure), when the head displacement (indicated as x_1 in Fig. 1) along the vertical axis is zero.

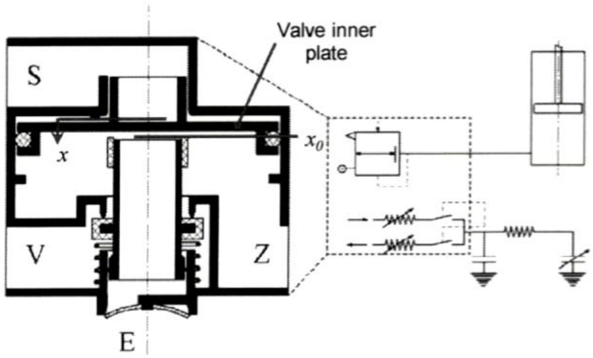

Figure 2: The compensation proportional valve, sketched together with the pneumatic circuit layout.

As the spindle head is raised up, the volume in the chamber increases and the fluids experiences an expansion with consequent pressure drop. On the other hand, when a downward displacement is applied, the volume in the chamber decreases and a pressure increment follows air compression. It is clear that, as pressure decreases or increases, the resulting force that reacts to weight varies as well. Since the weight value is fixed, this force variation would imply an unexpected extra force, which could prevent the regular head motion. It is therefore evident that a compensation valve must be introduced, with the important function of maintaining pressure at the fixed value of 6.2 bar, independently of the upward or downward motion of the moving head. When an abrupt vertical displacement and consequently an impulsive pressure change take place, the steady-state value must be restored after a sufficiently brief transitory. The valve consists of an inlet (*V*), an outlet towards the actuators (*Z*), an outlet for exhausting (*E*), and another inlet for regulated pressure (*S*). Exhausting to atmosphere is performed through a short pipe, which is able to shift in the axial direction, partially contrasted by the elastic force of a high stiffness spring. The layout of the valve is completed by a plate which is also able to shift in the same direction. When it is adherent or slightly far off the upper edge of the flange, the chamber *Z* at the low side is connected to atmosphere. Otherwise, when the plate moves down, the O-ring is able to prevent any connection to the external environment, while compressed air (6.2 bar pressure) may be supplied by the inlet *V*. In the initial conditions, pressure in the chamber *Z* is equal to its nominal value of 6.2 bar, meanwhile pressure in *S* is regulated at a slightly lower entity and then maintained constant. As a consequence, the resulting forces are balanced and the plate stands in its initial position (indicated by x_0 in Fig. 2), closing the connection with the exhausting pipe. It can be observed that when the plate is in contact with the gasket around the upper part of the exhausting pipe, the entire inferior surface is no longer available for pressure. The reason is that the area of the tube cross section, including also the gasket, must be detracted from the total area.

The behaviour of this device, which acts as a compensation valve, able to maintain the pressure constancy, may be explained as follows. Let us start from the hypothesis that pressure in *Z* experiences an abrupt increase, as the head is lowered. The plate moves upward, so that too high pressure can be exhausted to atmosphere. It must be remarked that the pneumatic conductance (considering its definition in [26-28]) of the connection to the external environment is directly proportional to the distance between the plate and the outlet and therefore strictly related to the plate upward displacement. Thus, pressure decreases, coming back to its nominal value after a transitory. It is easy to understand that if pressure decreases below 6.2 bar, when the head is raised up, the plate moves downward, closing connection to atmosphere and loading the spring. When the force generated by the pressure in S overcomes the elastic force of the spring, the inlet V is opened and compressed air can be supplied, thus re-establishing the initial pressure condition in Z.

3. DEVELOPMENT OF A SIMULATIVE MODEL

A mathematical model was developed for the numerical simulation of the performance of the entire system described in the previous section, considering the valve, the pneumatic circuit and its cylinders and the moving head. The plate translation was modelled, using Eq. (1), typically applied to second-order mechanical systems. The inertial forces, the spring elastic action and friction viscous effects are here considered. It must be remarked the differential equation in Eq. (1) has a different formulation, depending on the value of the plate displacement x (see Fig. 2). When the floating plate is not in contact with to the O-ring, i.e. for $x < x_0$, the spring stiffness must be set to zero. Otherwise, when $x \geq x_0$, the model must account for the spring stiffness (k_s), moreover the mass of the tube for exhausting (m_t) must be considered along with that of the plate (m_p) in the computation of the inertial force.

If $x < x_0$:

$$m_p \ddot{x} + c_p \dot{x} = p_s A_s - p_z A_z \tag{1a}$$

If $x \geq x_0$:

$$(m_p + m_t)\ddot{x} + c_p \dot{x} + k_s (x - x_0) = = p_s A_s - p_z A_z \tag{1b}$$

c_p is the damping coefficient, accounting for the viscous friction between the plate and the valve box, p_S is the regulated pressure (constant value), p_Z is the current pressure at the valve outlet, A_S and A_Z are the surfaces of the upper and of the lower sides of the plate. Experimental lab tests and suitable measurements were arranged to estimate the actual values of the aforementioned parameters.

The pneumatic conductances are estimated as follows:

If $x < x_0$:

$$C_{in} = 0 \tag{2a}$$

If $x \geq x_0$:

$$C_{in} = \alpha (x - x_0) \tag{2b}$$

If $x = 0$:

$$C_{out} = C \tag{3a}$$

If $0 < x < x_0$:

$$C_{out} = C - \frac{C}{x_0} \cdot x \tag{3b}$$

If $x \geq x_0$:

$$C_{out} = 0 \tag{3c}$$

C_{in} in Eq. (2) is the inlet conductance (from V towards Z), whereas C_{out} in Eq. (3) is the outlet conductance (from Z towards E), α is a suitable coefficient and C is the conductance of the completely opened exhausting pipe.

Considering then the pneumatic cylinders, it was necessary to model the pressure variations, considering both the entering or exiting flows and the compressions or expansions due to the head and cylinder piston displacements.

$$p_i = \left[\int \frac{G}{Ca} \right]_{i-1}^{i} + p_{i-1} \cdot \left(\frac{V_{i-1}}{V_i} \right)^n \tag{4}$$

The pressure at the *(i-th)* step of expansion/compression is computed in Eq. (4), where G represents flows entering or exiting the cylinders, p, V and Ca are the current values for pressure, internal chamber volumes and pneumatic capacity (according to its definition in [7, 29]). This term also depends on the current values of pressure and volume and on the polytrophic coefficient n (a value of 1.2 was assumed, as in [7]). The subscripts *(i-1)* and i stand for the calculation steps.

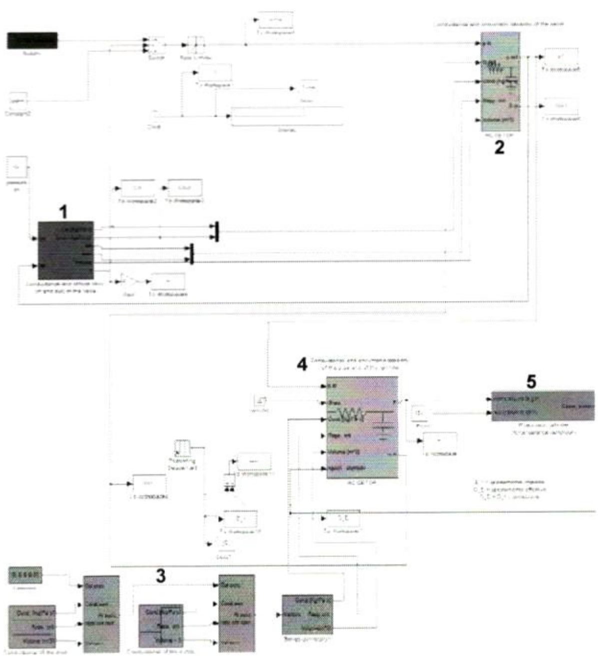

Figure 3: Layout of the Simulink model. Blocks 1 and 2: Proportional valve. Blocks 3 and 4: Pipes and cylinder. Block 5: Piston displacement.

As mentioned in Section 1, the head displacement is performed by ball screw. The following equations are able to model how the head position is followed by pistons, and how the pressure and the force counterbalancing weight vary after perturbations. The displacement of the pistons can be retrieved by the integration of Eq. (5), modelling the vibration of a second-order mechanical system with two degrees of freedom. The viscous term c_c accounts for friction between the pistons and the cylindrical surfaces, whereas the symbol m_c stands for the mass of cylinder pistons. The symbol k_c

finally stands for the stiffness of the horizontal beams connecting the moving head to each piston.

$$m_c\ddot{x}_2 + c_c\dot{x}_2 + k_c(x_2 - x_1) = F - P \qquad (5)$$

F is the total pneumatic force generated by the two cylinders and P is the head weight. For the sake of clarity, subscript 2 is added to refer to displacement (x_2), velocity ($\dot{x}_2$) and acceleration ($\ddot{x}_2$) at the piston location, whereas subscript 1 is appended to indicate the head controlled displacement (x_1). The related reference system is depicted in Fig. 1.

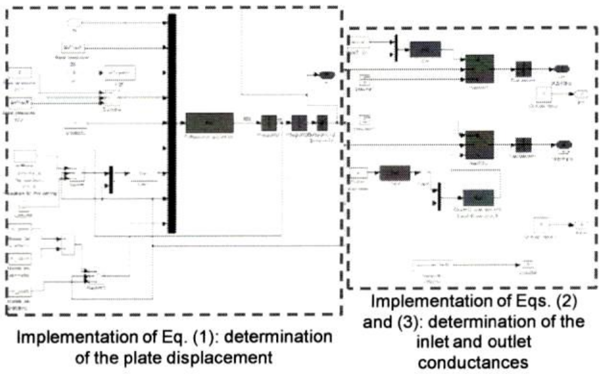

Figure 4: Internal layout of Block 1: implementation of Eqs. (1), (2) and (3).

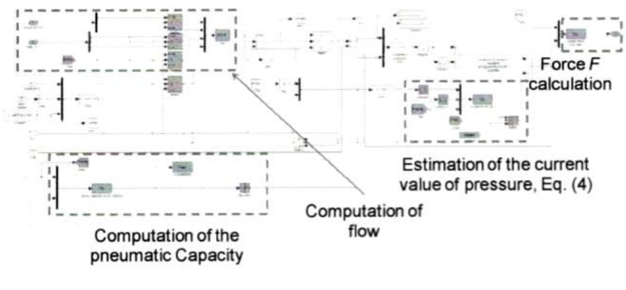

Figure 5: Internal layout of Block 4: implementation of Eq. (4) and calculation of the force F.

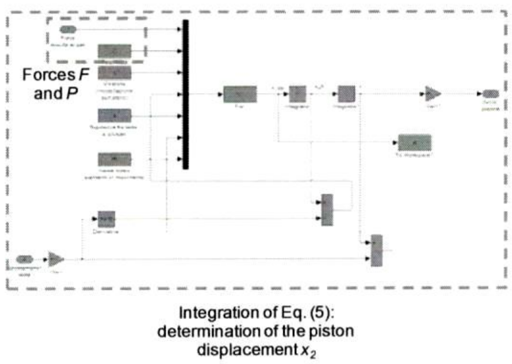

Figure 6: Internal layout of Block 5: integration of Eq. (5) for the determination of the piston displacement (x_2).

The entire mathematical model was implemented, using the Simulink package in the Matlab environment. This environment proved to be highly efficient in the implementation of differential and algebraic equation systems, as confirmed by previous contributions available in literature [7, 15]. A general view of the Simulink model is sketched in Fig. 3, some details are shown in Figs. 4 to 6.

4. SIMULATION RESULTS

The implementation of the math formulas, describing the pneumatic and mechanical behaviour can be summarized as follows. The input is given by the controlled displacement of the head. When a displacement is applied, the new position of the cylinder pistons is computed, integrating Eq. (5). Afterwards, a new value of the pressure in the cylinder is calculated by Eq. 4: two phenomena are taken into account: air supply or exhausting from the proportional valve and air compression or expansion. Then, the pressure p_Z (acting at the valve output), related to the pressure in the cylinder chamber, can be estimated. The implementation of Eq. (1) leads finally to the determination of the new plate position and of the updated inlet and outlet conductances (Eqs. 2-3). The developed model is therefore able to simulate the response of the proportional valve that maintains pressure p_Z at an approximately constant value. In particular, compressed air supply, as pressure decreases, and air exhausting, following a pressure increase, can be efficiently modelled.

The first aim consisted in the determination of the system behaviour, following abrupt displacements applied to the moving head. For this purpose, the response to step inputs was studied. It was therefore supposed that the head experienced step upward or downward displacements: the Simulink model made it possible to simulate the displacement of the pistons and the trend of pressure generating the counterbalancing force. The entities of the step displacements were chosen so that they were consistent with the technological processes performed by the MC. The trends of the head and of the piston displacements are compared in Fig. 7, considering an upward 3 mm step, generating an expansion in the cylinder chambers.

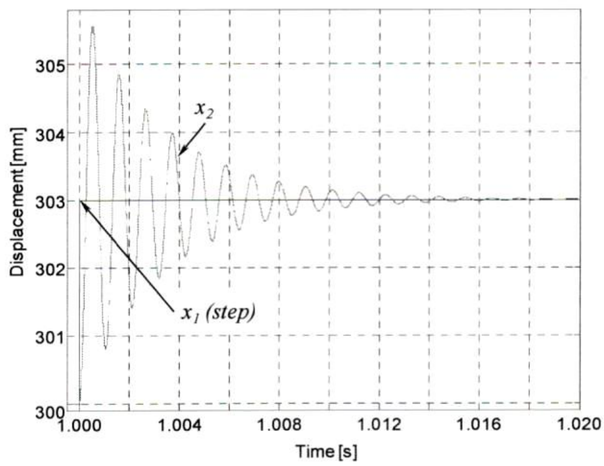

Figure 7: Controlled displacement (upward step, x_1) of the head, compared to that of the cylinder pistons (x_2).

It can be observed that the pistons experience damped vibrations which can be assimilated to those of the second-order mechanical systems. It is interesting to remark that the system proves to be very efficient, as the discrepancies between the two displacements drop to negligible values after just twenty milliseconds. Two pressure trends in the cylinder lower chambers are plotted in Figs. 8-9. The first one (Fig. 8) refers to the response to the aforementioned upward step. Whereas, the latter (Fig. 9) refers to the response to a downward 5 mm step generating compression. For the sake of brevity, the displacement trends are not plotted in this last case, as they are qualitatively similar to those in Fig. 7.

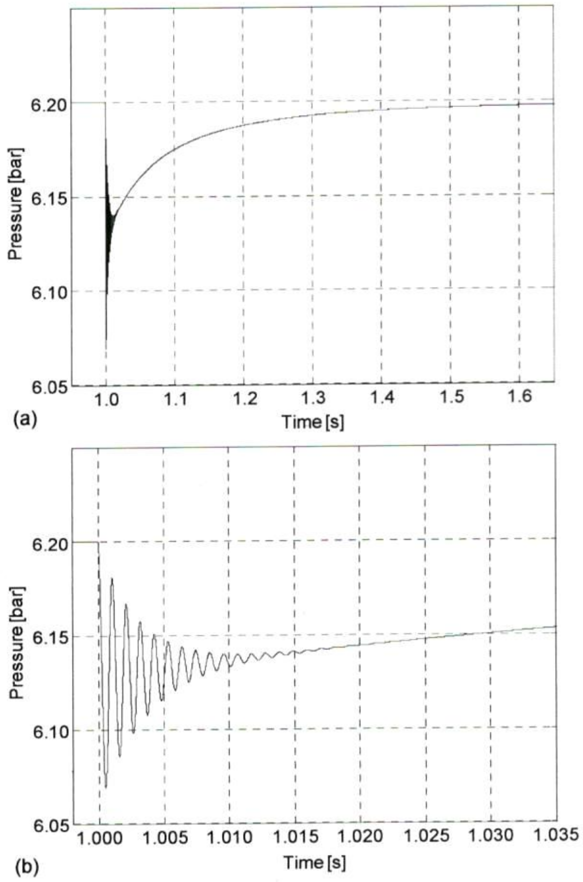

Figure 8: Pressure trend at the cylinder lower chamber after an upward step displacement (a), enlarged view in (b).

5. DISCUSSION

It is interesting to compare the pressure trends, when the head displacement generates an expansion or a compression. In the first case, Fig. 8(a), the pressure value has an abrupt decrease, followed by a quite rapid increase due to the compensating effect of the proportional valve. The enlarged view, Fig. 8(b), denotes several fluctuations in the globally ascending part of the curve, which are likely to be due to piston vibrations. The required value of pressure is established again after less than 1 s, but the difference between the required and the actual value comes to a negligible amount after just 0.1 s.

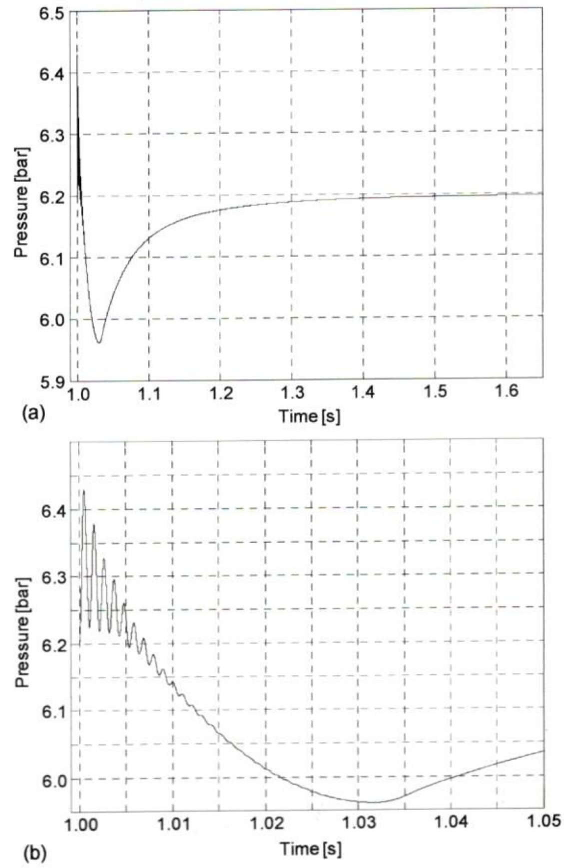

Figure 9: Pressure trend at the cylinder lower chamber after a downward step displacement (a), enlarged view in (b).

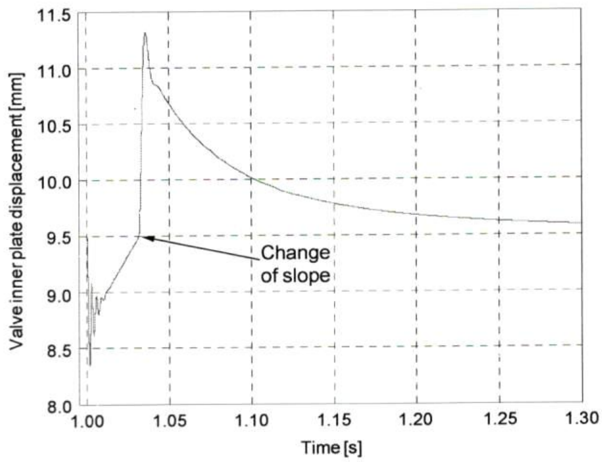

Figure 10: Displacement of the valve inner plate, when a downward step displacement is applied to the head.

In the case of the downward displacement, Fig. 9(a), with consequent cylinder compression, the pressure in the cylinder chamber experiences an abrupt increase, as expected. Just after, the valve exhausting outlet is opened, which implies a rapid and uncontrolled decrease. The consequence is that in just 5 ms time, pressure goes back to the reference value of

6.2 bar and then keeps decreasing, thus assuming lower values, like in the expansion case, Fig. 9(b). As pressure becomes lower than 6.2 bar, the supply inlet is opened and the reference value is established again after about 1 s from the step displacement. This occurrence can be better explained in the light of the graph in Fig. 10, where the displacement of the valve inner plate (indicated by x in the differential equation, Eq. (1)) is plotted. In the beginning, the plate is standing in its initial position, marked by x_0 in Fig. 2, corresponding to 9.5 mm. As a consequence of pressure increase, it moves upward, corresponding to lower values of x. Therefore, the exhausting outlet is opened with increasing conductance, as the plate moves upward. Upon air exhausting, pressure starts decreasing and the plate moves downward with some vibrations, corresponding to the pressure fluctuations of Fig. 9(b). After about 30 ms, the plate comes back to the initial position, thus closing the exhausting pipe, whose conductance drops to zero. The change of slope that can be observed at this time, indicating a steep increase of the plate velocity, can be easily explained and may be regarded as a proof of the high efficiency of the valve. As the pressure tends to decrease below 6.2 bar, the plate closes the connection with the exhausting outlet: it is at this moment that the area where pressure p_Z is applied (A_Z) gets instantaneously decreased. The reason for this occurrence is that the pipe cross section must be detracted from the total extension of the area where pressure is acting. The consequence is that the force transmitted by pressure p_S is contrasted by a much lower force (since p_Z acts on a significantly reduced area): this occurrence makes the plate accelerate in its downward displacement (x becomes higher than 9.5 mm). As a consequence, the supply inlet is rapidly opened, with consequent pressure increase. As the pressure increases to 6.2 bar again, the plate comes back to its initial position and the valve compensation task is concluded.

The final issue regarded the development of an equivalent simulative model, i.e. a model that was able to provide acceptable approximated results in a very short time. The main application of this equivalent model was in the simulation of the entire MC (considering all its five axes), where the here studied pneumatic-mechanical group is integrated. In this case the adoption of the complete model would have led to too long simulation times.

As it has been emphasized and clearly visible in Fig. 7, the system response in terms of the piston displacement (x_2) can be easily assimilated to that of second-order systems. In particular, the analysis of the displacement trend, exhibiting damped vibrations, leads immediately to the determination of the maximum sovraelongation and its period. The performed processing consisted in the estimation of the critical damping δ and of the related eigenfrequency ω_n, characterizing the whole pneumatic-mechanical system [30]. In other words, the entire device could be regarded as a conventional second-order system with the determined δ and ω_n parameters and, therefore, with the same dynamic behaviour. The calculated data, δ = 0,0496 and ω_n = 5935 rad/s, were then used to implement the simple mathematical model of the equivalent second-order system. By the integration of Eq. (6), it is possible to determine the piston displacement x_2', depending on the head motion law x_1. The superscript ′ is here appended to denote the aforementioned approximation.

$$\frac{\ddot{x}_2'}{\omega_n^2}+2\frac{\delta}{\omega_n}\dot{x}_2'+x_2'=x_1 \qquad (6)$$

Once estimated the trend of piston displacement, the last step consisted in the calculation of the amount of the pneumatic force that counterbalances weight. To get this result, the differential equation in Eq. (5) was inverted as in Eq. (7).

$$F=m_c\ddot{x}_2'+c_c\dot{x}_2'+k_c(x_2'-x_1)+P \qquad (7)$$

The adoption of the described simplified model made it possible to perform several simulations in a very short time and to investigate the mechanical response to many kinds of motion laws applied to the moving head. For instance, the response to a ramp is shown in the enlarged graph in Fig. 11. When considering a gradual movement of the head, representing the most frequent operating condition, rather than a step displacement, the vibration of the pistons has a really negligible amplitude, which is a further proof of the efficient pressure compensation operated by the proportional valve.

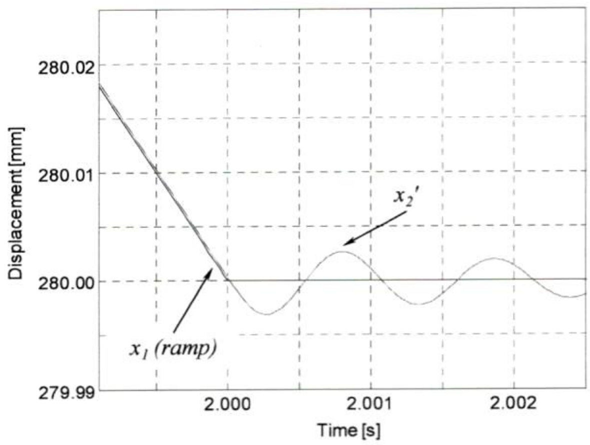

Figure 11. Response (x_2') to a ramp displacement (x_1) applied to the spindle head.

6. CONCLUSIONS

The present paper has dealt with the development of the analytical model of a pneumatic-mechanical device, whose main element is a proportional pneumatic valve. The main issues can be summarized in the following points.

- In the Introductive paragraph the field of application of the present research has been pointed out. Reference has been made to the design of a machining centre, whose z-axis consists of an integrated pneumatic and mechanical system.
- The main features of the pneumatic and mechanical system have been described, with particular reference to the operating mode of the proportional valve and to the procedure to dynamically counterbalance the weight of the spindle head.

- All the analytical equations regarding the mechanical issues and the air flow in the valve and in the circuit have been pointed out, along with their implementation in the Matlab-Simulink environment.
- The simulations dealt with the response to some critical operating tasks, where the moving head experiences step displacements with impulsive pressure variations. The results proved that the proportional valve is remarkably efficient, both when air exhausting and when supplying are required. The expected pressure value can be established again after less than 1 s.
- Finally, studying the response to step displacements made it possible to determine the main parameters (critical damping and eigenfrequency) of an equivalent second-order system, i.e. a system with the same dynamic behaviour.
- This outcome led to the development of a simplified Simulink model with a high efficiency from the computational point of view. Running this model, made it possible to test the efficiency of the compensation proportional valve in response to whatever motion law applied to the head.
- Regarding further applications, the second-order equivalent model can be incorporated into the entire model of the machining centre with excellent results in terms of accuracy and efficiency.

7. REFERENCES

1. Y. Altintas, C. Brecher, M. Weck and S. Witt, Virtual machine tool, *Annals of CIRP*, vol. 53(2), pp. 619-642, 2005.
2. M. F. Zaeh and T. Oertli, Finite element modeling of ball screw feed drive systems, *Annals of CIRP*, vol. 53(1), pp. 289-293, 2004.
3. J. S. Chen and W. Y. Hsu, Dynamic and compliant characteristics of a cartesian-guided tripod machine, *Journal of Manufacturing Science and Engineering*, vol. 128(2), pp. 494-502, 2006.
4. H. Zhang, W. Zhao, J. Zhang and H. Liu, Research on the modeling of dynamics for vertical axis ball screw feed system, *Proc. International Symposium on Assembly and Manufacturing, ISAM 2013 (Xi'an, China)*, Category number: CFP13ATP-ART, Code: 101457, 2013.
5. W. Z. Ding, X. D. Huang, M. L. Wang and B. S. Wang, Effect of trajectory-controlled algorithm on the mechatronical performance of high-speed machining, *Proceedings of the Institution of Mechanical Engineers, Part B: Journal of Engineering Manufacture*, vol. 227(9), pp. 1277-1286, 2013.
6. Y. Yang, W.-M. Zhang, Q. Zhou and T. Yang, Dynamic characteristic study of ball screw feed drive systems based on multi-flexible body model, *Journal of Donghua University (English Edition)*, vol. 30(2), pp. 118-124, 2013.
7. G. Olmi and G. Vassura, Modelling pneumatic circuits with variable-geometry nozzles: A hybrid approach, *Proc. 2006 SEM Annual Conference and Exposition on Experimental and Applied Mechanics (Saint Louis, MO, United States)*, vol. 1, pp. 447-457, Code 68397, 2006.
8. D. Croccolo, M. De Agostinis and N. Vincenzi, Structural analysis of an articulated urban bus chassis via FEM: A methodology applied to a case study, *Strojniski Vestnik/Journal of Mechanical Engineering*, vol. 57(11), pp. 799-809, 2011.
9. D. Croccolo, M. De Agostinis, G. Olmi and A. Tizzanini, Analysis of the stress state in brake caliper mounts of front motorbike suspensions, *Advances in Mechanical Engineering*, vol. 2013, article number: 525010, 2013.
10. M. Comandini, G. Olmi and A. Freddi, Fatigue performance of shot-peened gears investigated by experimental and numerical methods, *Transactions of Famena*, vol. 31(2), pp. 1-10, 2007.
11. G. Olmi, M. Comandini and A. Freddi, Fatigue on shot-peened gears: Experimentation, simulation and sensitivity analyses, *Strain*, vol. 46(4), pp. 382-395, 2010.
12. G. Olmi, Low cycle fatigue experiments on turbogenerator steels and a new method for defining confidence bands, *Journal of Testing and Evaluation*, vol. 40(4), pp. 539-552, 2012.
13. D. Croccolo, M. De Agostinis and G. Olmi, Fatigue life improvement of holed plates by interference fitted pins, *Proc. 2012 ASME International Mechanical Engineering Congress and Exposition, IMECE 2012 (Houston, TX, United States)*, vol. 8, pp. 65-71, DOI: 10.1115/IMECE2012-86118, 2012.
14. D. Croccolo, M. De Agostinis and G. Olmi, Fatigue life characterisation of interference fitted joints, *Proc. 2013 ASME International Mechanical Engineering Congress and Exposition, IMECE 2013 (San Diego, CA, United States)*, vol. 2B, pp. V02BT02A015 (10 pages), DOI: 10.1115/IMECE2013-63515, 2013.
15. G. Olmi and A. Freddi, A new method for modelling the support effect under rotating bending fatigue: Application to Ti-6Al-4V alloy, with and without shot peening, *Fatigue and Fracture of Engineering Materials and Structures*, vol. 36(10), pp. 981-993, 2013.
16. G. Olmi, A. Freddi, M. Bandini, M. Lanzoni and B. Riccò, Calibration and performance evaluation of hydrogenated amorphous silicon stress sensors, *Proc. 2006 SEM Annual Conference and Exposition on Experimental and Applied Mechanics (Saint Louis, MO, United States)*, vol. 3, pp. 1616-1626, Code 68397, 2006.
17. G. Olmi, A. Freddi and D. Croccolo, In-field measurement of forces and deformations at the rear end of a motorcycle and structural optimisation: Experimental-numerical approach aimed at structural optimization, *Strain*, vol. 44(6), pp. 453-461, 2008.
18. G. Olmi, Investigation on the influence of temperature variation on the response of miniaturised piezoresistive sensors, *Strain*, vol. 45(1), pp. 63-76, 2009.
19. G. Olmi, A new loading-constraining device for mechanical testing with misalignment auto-compensation, *Experimental Techniques*, vol. 35(6), pp. 61-70, 2011.
20. G. Olmi, A Novel Method for Strain Controlled Tests, *Experimental Mechanics*, vol. 52(4), pp. 379-393, 2012.
21. G. Olmi, An efficient method for the determination of the probability of failure on the basis of LCF data: Application to turbogenerator design, *SDHM Structural Durability and Health Monitoring*, vol. 8(1), pp. 61-89, 2012.

22. D. Croccolo, M. De Agostinis and G. Olmi, Experimental characterization and analytical modelling of the mechanical behaviour of fused deposition processed parts made of ABS-M30, *Computational Materials Science*, vol. 79, pp. 506-518, 2013.
23. G. Olmi and A. Freddi, Reliability assessment of a Turbogenerator Coil Retaining Ring based on Low Cycle Fatigue data, *Archive of Mechanical Engineering*, vol. 61(1), pp. 5-34; DOI: 10.2478/meceng-2014-0001, 2014.
24. D. Croccolo, M. De Agostinis, P. Mauri and G. Olmi, Influence of the engagement ratio on the joint strength of press fitted and adhesively bonded specimens, *International Journal of Adhesion and Adhesives*, DOI: 10.1016/j.ijadhadh.2014.01.017, In Press.
25. G. Olmi, An experimental investigation on a crack propagating from a geartrain housing in an asphalt milling machine, *Engineering Failure Analysis*, vol. 38, pp. 38-48, 2014.
26. International Organization for Standardization, *ISO 6358: 1989 Pneumatic Fluid Power - Components Using Compressible Fluids - Determination of Flow-Rate Characteristics First Edition*, Geneva, Switzerland, 1989.
27. International Organization for Standardization, *ISO 6358-1: 2013 Pneumatic fluid power - Determination of flow-rate characteristics of components using compressible fluids - Part 1: General rules and test methods for steady-state flow - First Edition*, Geneva, Switzerland, 2013.
28. International Organization for Standardization, *ISO 6358-2: 2013 Pneumatic fluid power - Determination of flow-rate characteristics of components using compressible fluids - Part 2: Alternative test methods - First Edition*, Geneva, Switzerland, 2013.
29. G. Belforte, *Manuale di Pneumatica – II Edizione*, Tecniche Nuove, Milan, 2005.
30. S. Doughty, *Mechanics of Machines*, John-Wiley & Sons, New York, 1988.