A dynamic load-simulator for servo-systems

A Dynamic Load-Simulator for Servo-Systems

C. J. Heuvelman (1), P. C. Mulders and M. van den Bersselaar

Servo-systems are widely used in machine tools, flexible automation systems and robots. A servo-system consists of a servo-motor with its drive and control, and the load. Often this load is a workpiece and its carrier, e.g. a workpiece and a robot-arm. The main task of the servo-system is to position the workpiece with a defined accuracy and speed. However, the properties of the servo-system depend very largely on the load of the servo-motor which must be well known. In general the load of a (rotating) servo-motor consists of a combination of inertia of mass, viscous damping and torsional stiffness. In many cases this load varies as a function of the time and so the properties of the servo system will vary. This paper describes an equipment which simulates these kinds of loads. A servo-motor connected to the load-simulator can thus be examined on its properties. The load simulator consists of a DC-motor reduction gear, a output-torque sensor and a microcomputer which calculates from the measured torque an angular frequency which belongs functionally to the desired load.
A servo-system in the simulator takes care that angular frequency of the output-shaft will follow the calculated value as close as possible.
The load simulator can handle torques up to 36 Nm and angular frequencies up to 4.8 rad/s. The bandwidth of the system is 31 Hz.

Introduction

With drives for machine tools, manufacturing systems and robotics servo-mechanisms are very common. A servomotor-drive system is able to follow a desired path and velocity accurately. This follow-property is very important and with the design of a servo system this has to be investigated thoroughly. With the aid of the transfer-characteristic a number of important properties of the servo system can be determined, either in the frequency or in the time domain. By measuring the transfer-characteristic a servo system can be adjusted, e.g. minimizing the overshoot in the step response. Optimizing adjustment of the servo system depends on the load of the system. The load of a rotational servo motor can be damping, mass of inertia or torsional stiffness or combination of these elements. When the load changes re-adjustment of the servo controller (e.g. a P.I.D.-controller) may be necessary. This is especially the case with flexible automation systems and robotics. The mass of inertia as a load of the servo motor of a robot working in polar coordinates changes as a function of time when the robot moves his arm in- or outwards. In this case, the load of the servomotor, a mass of inertia can very by a factor of 4.

Testing these effects in the design laboratory is rather difficult: changing the values of the mechanical load, e.g. the mass of inertia, as a function of time is not easy. This paper describes a device which can act as a load for a servo motor. The device can be programmed as a combination of a torsional stiffness with mass of inertia and damping, as well as statically as dynamically,
The load simulator can handle a maximum T = 36 Nm, maximum angular frequency Ω = 4.8 rad/s. The bandwidth of the system is B = 31 Hz(3dB).

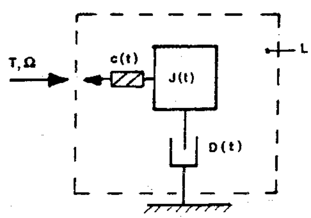

Fig. 1. Possible simulation of torsional stiffness c, mass of inertia J and damping D by the load simulator L.

Description of the system

For a servo system under test the load simulator must be a shaft which, when a torque T is applied to it, rotates with an angular frequency Ω according to:

$$\Omega = \frac{1}{J} \cdot \int [T - D.(\Omega - \frac{1}{c}\frac{dT}{dt})]dt + \frac{1}{c}\frac{dT}{dt} \qquad (1)$$

This equation corresponds to the combination of a mass of inertia J, a viscous damping D and a torsional stiffness c of the simulated load represented in fig. 1.

The load simulator consists of a motor which drives via a reduction gear the output shaft with angular frequency Ω, a torque measuring device and a unit which calculates the desired speed $\Omega = f(T)$. The input of the calculation unit is the measured torque T, the output is fed to the motor via an amplifier. Actually the amplifier-motor combination is in itself a servosystem in order to obtain an accurate output speed for the total system.
Fig. 2 shows the entire set-up of the load simulator

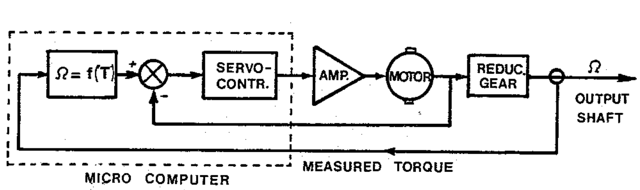

Fig. 2. The set-up of the load simulator

The calculation unit and the servocontroller operates digitally and is realized by a microcomputer based on the INTEL 8085. The application of digital techniques implies the need for sampling of the input data: angular frequency Ω and torque T so equation (1) has to be discretized. With backwards differentation this yields:

$$\Omega\,(n.T_s) = \frac{T_s}{J} \cdot \sum_{k=1}^{n} [T(k.T_s) - D.[\Omega(k.T_s) - \frac{1}{c}\frac{T(k.T_s) - T(k.T_s - T_s)}{T_s}]] + \frac{1}{c}\frac{T(n.T_s) - T\,(n.T_s - T_s)}{T_s} \qquad (2)$$

In this equation (2) is T_s the sample time. If T_s is small compared with the smallest mechanical time constant in the system, the calculated Ω may be considered as being continuously. In that case the term $\Omega(k.T_s)$ may be replaced by $\Omega(k.T_s - T_s)$, so equation (2) now can be resolved.

The mechanical part of the simulator consists of a dc-motor (Servalco TM 530) with a harmonic drive as reduction (HDUC 20-80-2A BL3), an incremental encoder mounted on the motor shaft as tachogenerator and a device to measure the torque applied to the output shaft. The motor is driven by a dc amplifier preceeded by a digital to analogue converter. The incremental tacho actually measures the motor angular frequency instead of the output angular frequency. However, besides of dynamic effects, this output frequency is known by the reduction ratio 80:1 of the harmonic drive.
The input torque is by measuring the reaction torque of the total system. For that purpose the motor-harmonic drive-output shaft combination is suspended by means of three springs in a ring. The deflection of a spring is linear to the applied torque and is measured by strain gauges.

The dynamic properties of the mechanic system

The dynamic properties of the mechanic system define the dynamic properties of the entire load simulator. These properties are a result of:

- transfer characteristics of the motor and tacho
- transfer characteristics of the harmonic drive
- the infuence of the torque measuring device
- the influence of the applied torque, the latter will have influence on the output angular frequency since the motor-reduction gearcombination has a an angular frequency source not a mechanicaloutput impedance equal to zero.

If no internal feed-back and no applied torque is assumed, the output angular frequency can be written in Laplace notation as:

$$\Omega(s) = H_1(s).\Omega_M(s) \tag{3}$$

where $\Omega_M(s)$ is the input voltage to the dc-amplifier and $H_1(s)$ the overall transfer characteristic of the amplifier, motor-tacho and harmonic drive, which can be written as

$$H_1(s) = V(s).\frac{1}{i}\frac{\mu_1}{1+\tau s} \tag{4}$$

with
$V(s)$ amplification of the dc-amplifier,

i the reduction ratio of the harmonic drive

$$\mu_1 = \frac{K}{R_A\ D_t + K^2}$$

$$\tau = \frac{J_t\ R_A}{R_A\ D_t + K^2}$$

In these expressions are:
K motorcontante
R_A rotor resistance
D_t and J_t total viscous damping and mass of inertia respectively of the motor and harmonic drive, all reduced to the motorshaft angular frequency.

In equation (4) the influences of armature self-inductance, damping and torsional stiffness of the incremental encoder and torsional stiffness of the motor output shaft have been neglected.

An external torque applied to the output shaft of the entire system causes a change of the angular frequency.
Applying the superposition theorem (for linear systems) this extra angular frequency will be:

$$\Delta\Omega(s) = (H_2(s) + H_3(s)).T(s) \tag{5}$$

with:

$$H_2(s) = \frac{1}{i^2}\frac{\mu_2}{1+\tau s}$$

$$H_3(s) = \frac{s}{c_1}$$

and:

$$\mu_2 = \frac{R_A}{R_A\ D_t + K}$$

c = torsional stiffness of output shaft of the harmonic drive.

In this equation the same neglections as in equation (4) have been used.

In equation (5) the extra angular frequency consist of two contributions, one caused by H_2 and the other by H_3.
The reason for that is that the contribution caused by H_2 can be measured by the tacho, the contribution by H_3, viz. by the torsional stiffness of the output shaft cannot. The contribution by H_2 can be reduced by applying a feedback loop.

Schematically equations (3) and (5) can be represented by the diagram of fig. 3.

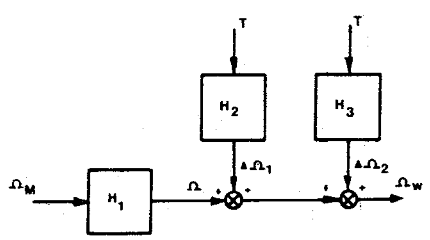

Fig. 3. Schematic representation of the mechanical system of the simulator. Applied torque T causes deviation of the desired output angular frequency by $\Delta\Omega_1 + \Delta\Omega_2$.

As mentioned before, the properties of the simulator can be improved by applying a feedback loop for the angular frequency. The angular frequency of the motor shaft, measured by an incremental encoder, is compared with the desired frequency delivered by the calculation unit. The measured angular frequency is assumed to be linear to the reduced angular frequency, which is present just behind the harmonic drive reduction but before the output shaft of the harmonic drive.

In fig. 4 the mechanical system with a velocity control loop is given. The output of the incremental encoder is transferred to a digital signal whose is proportional to the angular frequency $\Omega + \Delta\Omega_1$. The difference of this signal with the output of the calculator Ω_M is fed to the proportional controller R which output is connected to the motor amplifier via a digital to analogue converter.

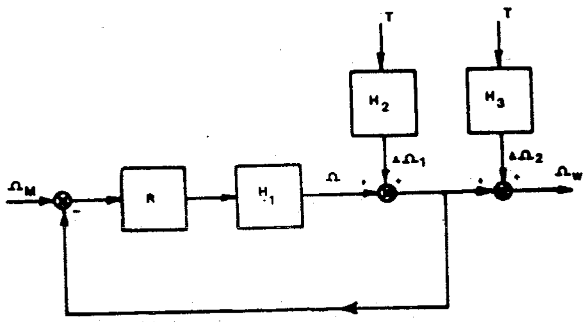

Fig. 4. Schematic diagram of the mechanical system with velocity feedback. The measured angular frequency $\Omega + \Delta\Omega_1$ is fed back and compared with the desired angular frequency Ω_M.

The control loop causes a apparent reduction of the transfer-characteristics H_1 en H_2 by a factor of:

$$\frac{1}{1 + R\ H_1(s)} \tag{6}$$

Applying this reduction factor to equations (3) and (5) yields the output angular frequency Ω_{out}:

$$\Omega_{out}(s) = \Omega(s) + \Delta\Omega_1(s) + \Delta\Omega_2(s)$$

$$\Omega_{out}(s) = \frac{R.H_1(s)}{1+R.H_1(s)}\Omega_M(s) + \frac{H_2(s)}{1+R.H_1(s)}T(s) + H_3(s)\ T(s) \tag{7}$$

$$\Omega_{out}(s) = H_1'(s)\ \Omega_M(s) + H_2'(s)\ T(s) + H_3(s)\ T(s) \tag{8}$$

The values for H_1, H_2 and H_3 are calculated with the aid of equations (4) and (3) data of the motor and harmonic drive.
In fig. 5 the modulus of H_1' is plotted versus the frequency for three feed back conditions; R=0 no feed-back: $H_1' = H_1$ and for R=1 and R=6. It is obvious that increasing the feed-back (increase of R) causes an increase of the bandwith. With R=6 the bandwith is 28 Hz.

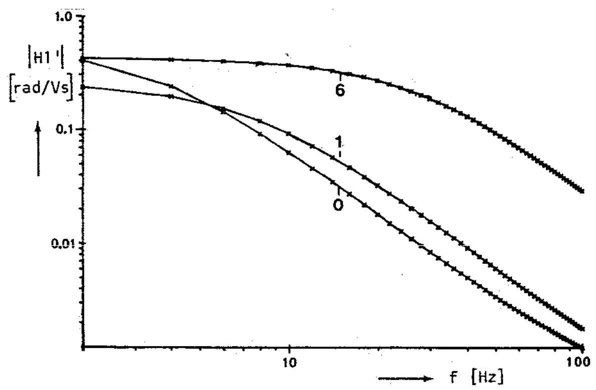

Fig. 5. Calculated values of H_1' versus the frequency for R=6, R=1 and no feed-back (R=0).

The modulus of $H_2' + H_3$ is plotted in fig. 6 for the three feedback conditions. It is clear that feed-back has no influence on H_3, which represents the torsional stiffness of the output shaft. The leftside of the plot of fig. 6 is caused by H_2.

For R=6 the value of $H_2'(o) + H_3(o)$ equals to 0,015 (rad/Nms). With the maximal permissable torque of T_M = 36 Nm this causes an increase of the angular frequency of the output shaft of $\Delta\omega(o) = (H_2'(o) + H_3(o)).\ T_M$ = 0,54 rad/s.
The maximum value of the angular frequency is 4.8 rad/s, so in this case the relative accuracy of the system will be 11%.

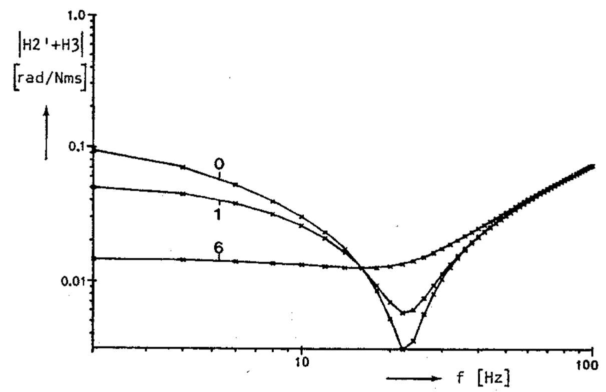

Fig. 6. Calculated values for $H_2' + H_3$ versus the frequency for R=6, R=1 and no feed-back (R=0).

Verification of the model of the simulator

From the load simulator the values of H_1', H_2' and H_3 have been measured.
From equation (8) it is obvious that H_1' can be measured by taking T=0 so

$$H_1'(s) = \frac{\Omega_{out}(s)}{\Omega_M(s)}$$ or in the frequency domain:

$$H_1'(j\omega) = \frac{\Omega_{out}(j\omega)}{\Omega_M(j\omega)}$$

The measurement is carried out by applying an electrical signal $\Omega_M(j\omega)$ to the servo amplifier and recording Ω_{out}. Since the angular frequency of the output shaft is rather low (max. 4.8 rad/s or 46 rev/min), this is rather difficult with a conventional tachogenerator. In this case the angular accleration $\dot{\Omega}_{out}$ is measured with a suitable transducer (Hottinger type BD). The resonance frequency of this transducer is 3Hz, above this frequency the instrument acts as an angular displacement transducer. Since the frequency response above 3 Hz is of interest the output signal has been differentiated. In fig. 7 the result of this measurement is given for R=6. The falling curve below 3Hz is due to the properties of the transducer: the curve has to be horizontal. The measurement shows a fair agreement with the theoretical value (broken line); the measured bandwidth is 31 Hz.

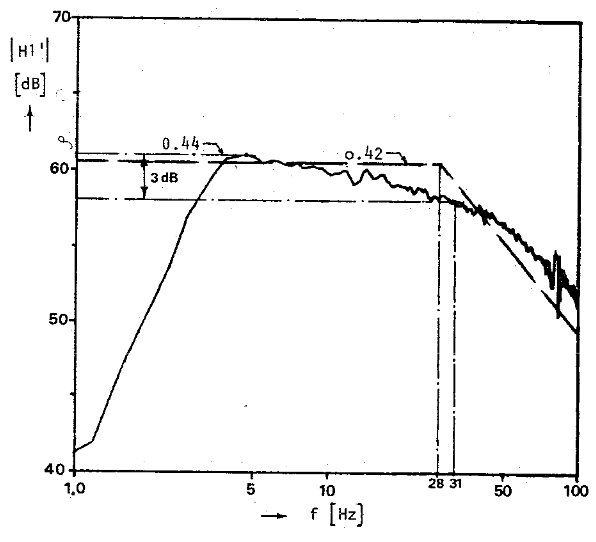

Fig. 7. Measurement of $H_1'(j\omega)$ for R=6 the falling curve below 3Hz is caused by the acceleration transducer.

The measurement of $H_2' + H_3$ was more difficult.
According to equation (8) $H_2' + H_3$ can be measured by taking Ω_M=0, so:

$$H_2'(j\omega) + H_3(j\omega) = \frac{\Omega_{out}(j\omega)}{T(j\omega)}$$

The torque is applied to the output shaft of the simulator by using a similar simulator which was coupled with shaft to the shaft of the simulator under test. The introduced torque could easily be measured with its own torque measurement device.

However, since both shafts of each simulator were tightly coupled to each other, a tachogenerator or accelerometer could not be used at that point. However, since both simulators were similar, the average value of the output of each incremental encoder was taken. This value can be considered to be linear with the angular frequency of the coupling points of both shafts.
This measurement shows a value for $H_2'(o)$ = 0,010 with R=6. This value is lower than the theoretical value 0,015. The cause of this discrepancy is not quite sure, the damping (viscous and coulomb) is probably higher than assumed.

Conclusions

The load simulator shows to be a practical device for testing servo systems under varying loads. The simulator can simulate loads rather accurately (10%). Since the bandwith of the system is approximately 30Hz the device can easily simulate varying loads. Perhaps for very fast servo systems under test the bandwidth is not enough. Further research is necessary in that case.

Acknowledgement

The authors wish to express their thanks to Mr. G.J. Theuws for his help and advice with the realisation of the mechanical construction of the load simulator.

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