Design of a high speed low friction XY-table

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Summary

This paper is about about the replacement of the current mechanical XY-guidings with airbearings to be able to reach higher speeds and an increase in accuracy.
The same building volume and weight as the current mechanical XY-table had to be used to be able to make an in situ replacement. Also cost had to be in the same order of magnitude as the current table.
In Adams a model has been made to obtain insight in the influence of the supporting stiffness and the dynamical behaviour of the whole machine with the new table.
Airbearings have frequency dependent damping which was implemented inside ADAMS using the internal controls toolbox.
The control loops and setpoint generation has been done in Matlab/Simulink and final performance evaluation in a co-simulation of Matlab and Adams.
The simulation reveals that it is possible to implement the frequency dependent damping. It is also advantageous to implement the damping in this way, because there is a clear difference with constant stiffness and damping.

1 Introduction

Within Philips many applications use XY-tables for point to point movements. They consist of two stacked 1D axes (sledges). The sledges can be driven directly (mostly spindle) or for more high performance applications directly at the load with two linear motors.
The main functions of the table are to prevent Rz rotation and give vertical support. The Rz-rotation is important because the point where high accuracy is needed is often placed outside the centre of gravity.
Mostly used are XY-tables with mechanical bearings. They have friction and a limited X-, Y- and Rz stiffness for given dimensions and weight. The vertical stiffness is high. Air bearings have the advantage that high axial stiffness can be achieved for a low mass. Rotational stiffness is dependent on the length of the guiding.

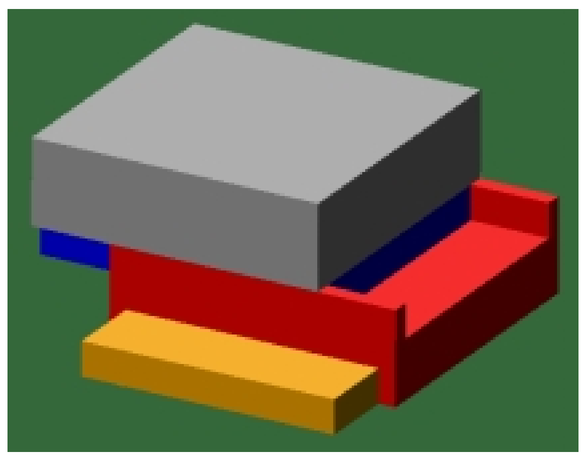

Figure 1 : Model of XY-table

Due to the limited stiffness of mechanical bearings perpendicular to the required motion two 1 dimensional position sensors are not possible. An expensive 2 dimensional sensor which measures directly from the load towards the ground is needed to reach high performances.
The speed of these tables is limited if accuracy in the order of magnitude of 1 micrometer and below are needed. Movement times below 20 milliseconds are hard to achieve.
This study is to see the effects of replacing the mechanical guidings with airbearings. The same building volume and weight as the current mechanical XY-table must be used to be able to make an in situ replacement. The cost of the more expensive air bearings must be counter balanced by a better performance and lower costs of encoders by using two 1 dimensional encoders.

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2 Behaviour of air bearing

A very simple model of an (circular) air bearing is depicted in Figure 2.

d = diameter of restriction
D = diameter of the bearing
h_a = hight of air film in bearing
p_a = ambient pressure
p_s = pressure in restriction

Air bearings have a frequency dependent behaviour. This behaviour can be characterised in a low frequent and a high frequent behaviour with a transition area in between. At low-frequencies the stiffness comes from the difference in flow-resistance in the restriction and the air film. Which requires finite element calculations. At high frequencies the restriction is choked and the stiffness can be calculated with the gas law, with the assumption that the amount of air in the bearing remains constant and the temperature does not increase (isotherm).
For a certain air bearing the frequency dependent behaviour has been calculated (See Figure 3).

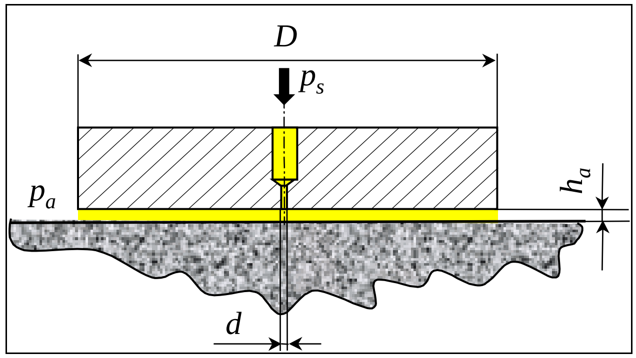


Figure 2 : Simple model of air bearing

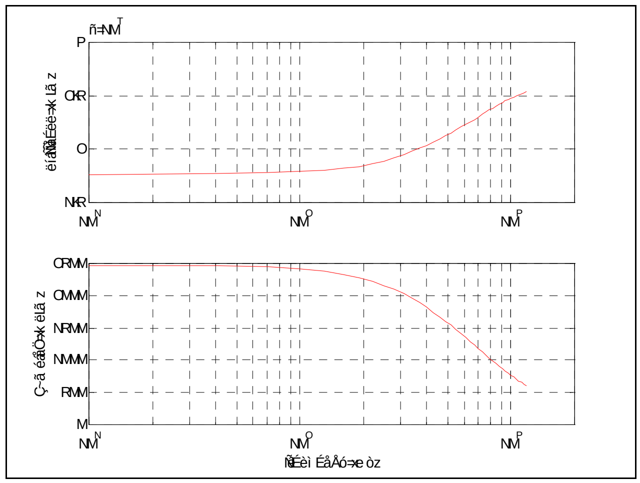


Figure 3 : Calculated data for airbearing

2.1 FIT TO CALCULATED DATA

The force in a spring-damper combination can be written as:

$$F = F_{spring} + F_{damper} = K \cdot u + C \cdot \dot{u} \qquad (1)$$

with:
u = displacement in spring damper
K = stiffness
C = damping

Representing this function in the frequency domain using Laplace; d/dt can be replaced with the operator s and for sinusoidal signals s = jω so:

$$\dot{u} = s \cdot u = j \cdot \omega \cdot u \qquad (2)$$

Substitution of (2) in (1) yields:

$$F = K \cdot u + C \cdot \dot{u} = (K + j \cdot \omega \cdot C) \cdot u = K_{complex} \cdot u \qquad (3)$$

So the stiffness can be seen as the real part of a complex stiffness and the damping as the imaginary part divided by the frequency (in radians).
In this way it is possible to interpret the frequency dependent air bearing behaviour as a single complex stiffness. It is namely possible to fit a lead-lag filter through the calculated data. The advantage is then that only a single measure is needed. Filtering this measure the lead-lag gives the spring-damper force. This is easy to implement inside ADAMS with the help of the control toolbox.

The lead-lag filter is defined by three parameters:

- Stiffness at 0 Hz (K_0)
- Stiffness at frequency infinite (K_∞)
- Damping at 0 Hz (C_0)

A lead-lag filter can be seen as the sum of a constant and a first order highpass filter:

$$K_{complex} = N_1 + N_2 \frac{s}{s/\omega_b + 1} \tag{4}$$

$N_1$and N_2 are two constants and ω_b is the break point of the high pass filter
Splitting this function in a real and imaginary part with the help of (2) reveals the stiffness and damping :

$$K_{complex} = N_1 + N_2 \frac{j\omega}{j\omega/\omega_b + 1} = N_1 + N_2 \frac{\omega^2/\omega_b}{\omega^2/\omega_b^2 + 1} + j\omega N_2 \frac{1}{\omega^2/\omega_b^2 + 1} \tag{5}$$

$$K = \mathrm{Re}(K_{complex}) = N_1 + N_2 \frac{\omega^2/\omega_b}{\omega^2/\omega_b^2 + 1} \qquad (5).a$$

$$C = \frac{\mathrm{Im}(K_{complex})}{\omega} = N_2 \frac{1}{\omega^2/\omega_b^2 + 1} \qquad (5).b$$

Substitution of ω=0 rad/s in (5).a en (5).b gives $N_1 = K_0$ en $N_2 = C_0$.
Taking the limit of ω towards infinity in (5).a gives $K_\infty = N_1 + N_2\omega_b$ thus $\omega_b = (K_\infty - K_0)/C_0$.
This results in:

$$K_{complex} = K_0 + \frac{C_0 \cdot s}{\frac{C_0}{K_\infty - K_0} \cdot s + 1} \tag{6}$$

In Figure 4 formula (6) is plotted against the data showing an excellent fit.
Air bearings become instable if the low frequent stiffness is higher then the high frequent stiffness. Formula (6) satisfies this behaviour: if the low frequent stiffness is higher the damping must be negative, resulting in an instable system. On the other hand if the low frequent stiffness is made lower due to a different design of the airbearing, higher damping results.

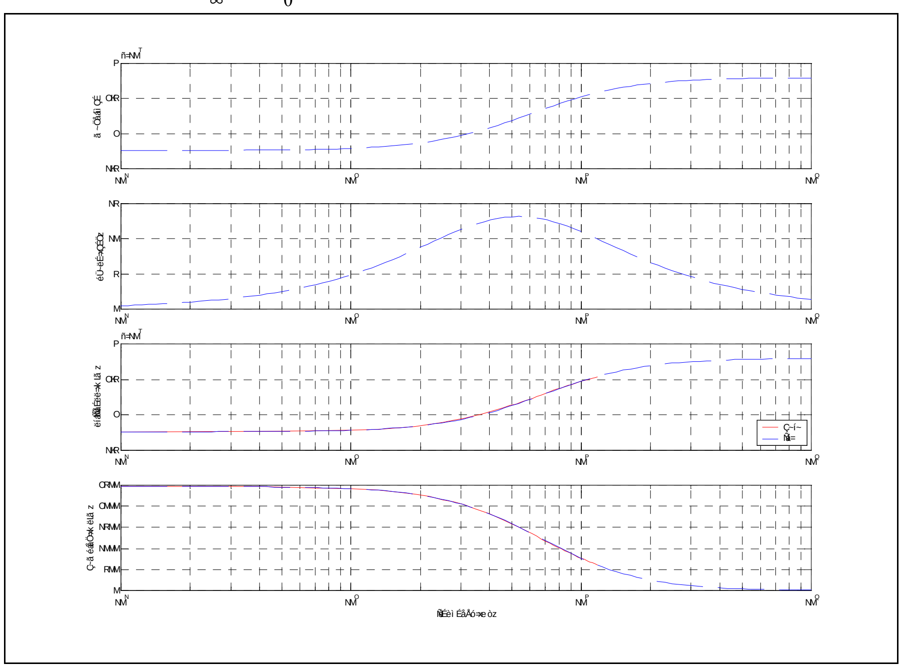


Figure 4 : Fit of lead-lag filter on airbearing stiffness and damping

3 Approach

- The mechanics are modelled in ADAMS including forces, encoders and a measurement of the tip position
- In the first model the damping and stiffness are taken at a fixed value corresponding to the stiffness and damping at 0 Hz, in the second model the frequency dependent damping is implemented in ADAMS using the controls toolbox
- Controllers and setpoint generation are modelled in Matlab-simulink.
- To tune the controllers are tuned with a linearised model using ABCD-matrices of the mechanics including the frequency dependent damping exported from ADAMS.
- Final time-domain performance analysis is done with the help of a non-linear co-simulation of ADAMS with Matlab-simulink.

4 Adams and simulink models

4.1 ADAMS MODEL

The guiding consists of 3 parts. The bearings are between the lower and middle part and the middle and upper part. The lower part is connected to ground, the upper part to the load. The bearings are modelled using a general force.
A complete bearing consists of 8 small air bearings which together give support in 5 dof's. From the characteristic of the single bearing the supporting stiffness in each direction can be constructed. The distances and angles between the two parts of the complete bearing are measured and put into a lead-lag function and then the result is put into the general force function. The measure connected to the wanted motion has been put into a state-variable as the encoder-measure
The driving forces are each a function of a state-variable which can be used in Matlab-simulink. Also a measure has been made at the actual end-effector position where the real accuracy is needed.

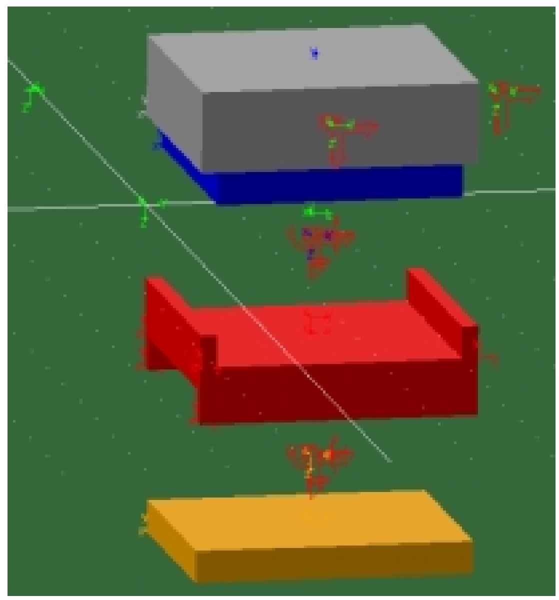


Figure 5 : Adams model

4.2 SIMULINK MODEL

In simulink two models are used.
The first one is only used to tune the control loops. (see Figure 6). It consists only of two control loops, both control loops have only a P-action and a lead-lag filter.

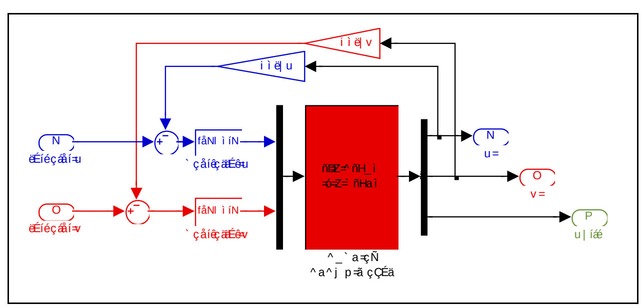


Figure 6 : frequency domain model

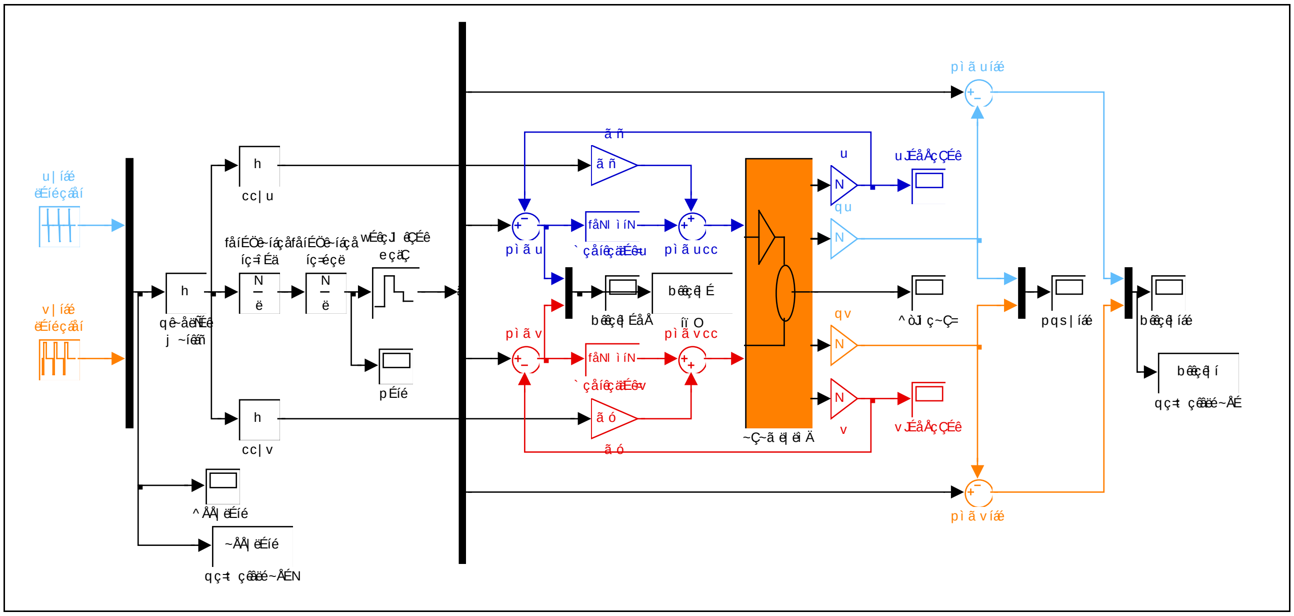


Figure 7 :Simulink model for time-domain simulations

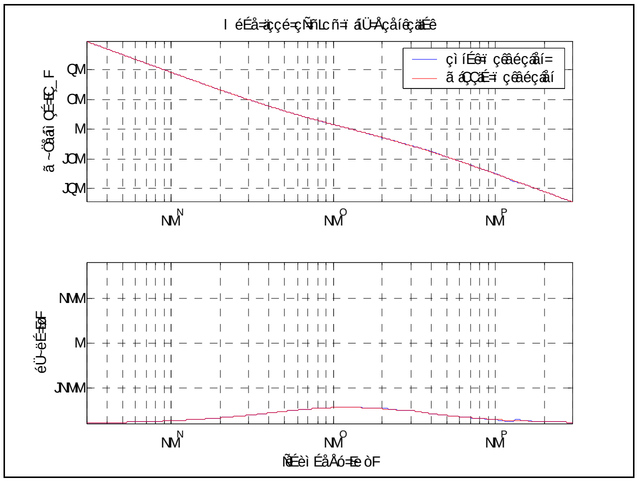
The second simulink model (see Figure 7) represents the whole model for the control of the XY-table.
It is important to be aware that the main movement direction is at an angle with respect to the orientation of the XY-table. And also that the load must be positioned with respect to a point outside the table.
The setpoints are specified in accelerations and must be integrated twice to obtain the position
From left to right:

- The setpoints are constructed in the main movement directions of the table. (X-tip and Y-tip in Figure 7)
- With a gain-balancing matrix these setpoints are translated to the axes of the XY-table. With repect to the table-axes the setpoint positions and feedforwards are generated.
- The zero-order hold is necessary to keep the setpoint position constant in the periods between the data exchange. Otherwise the discrete positions coming from the ADAMS model would give a saw-tooth like error and a very noisy control behaviour.
- The controllers for both axis
- The ADAMS plant
- Errors at both encoder and tip are measured for various positions

5 Simulations

5.1 TUNING THE CONTROLLERS

In Figure 8 and Figure 9 a simple PD controller is added to the mechanics resulting in 130 Hz bandwidth with 35 degrees phase margin . Bandwidth is in this case defined as the 0dB cross-over frequency in the open loop. Actually the bandwidth could be chosen higher, but in reality the bandwidth is limited by internal modes of the load which are not modelled. It is also clear that the position dependent mechanics are almost non-observable at the encoders. Especially interesting is that the resonance in Figure 9 lays sufficiently high frequent to allow the 130 Hz bandwidth.
So from the controller point of view it is indeed possible to use two 1 dimensional encoders.



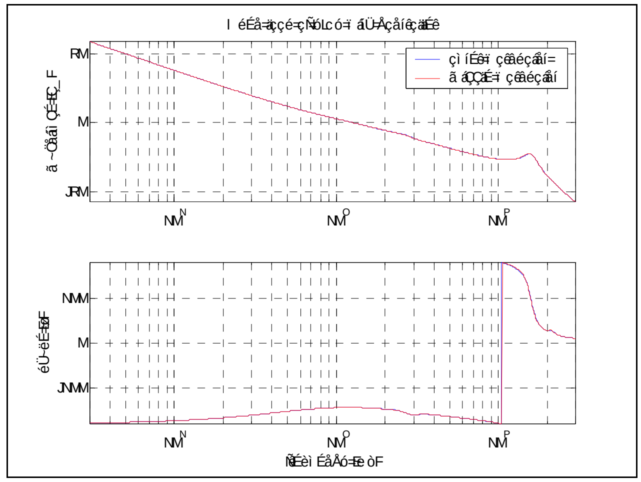


Figure 8 : Openloop of x/Fx

Figure 9: Openloop y/Fy

5.2 TIME DOMAIN: FREQUENCY DEPENDENT VERSUS CONSTANT STIFFNESS AND DAMPING

In Figure 10 a typical 10 ms, 1mm stroke set-point can be seen. A third order setup function is used from which the acceleration is plotted. It is clear that the frequency dependent stiffness and damping results in less error and also better damping of the error. The difference in settling time at the tip is large.

The settling time is measured at the tip and defined as the time needed after the end of the set-point to come within specification.

For a 3 micrometer specification the settling time is 7 milliseconds for the frequency dependent stiffness and damping and 17 ms for the constant stiffness and damping (see Figure 10 bottom graph). The first settling time is acceptable, the second not. Also the difference in rotational eigenfrequency is considerable: 230 Hz for the frequency dependent damping versus 200 Hz for the constant stiffness and damping. So implementing a frequency dependent stiffness and damping is favourable for the performance and resulting in less initial stiffness and damping needed, which make the design easier.

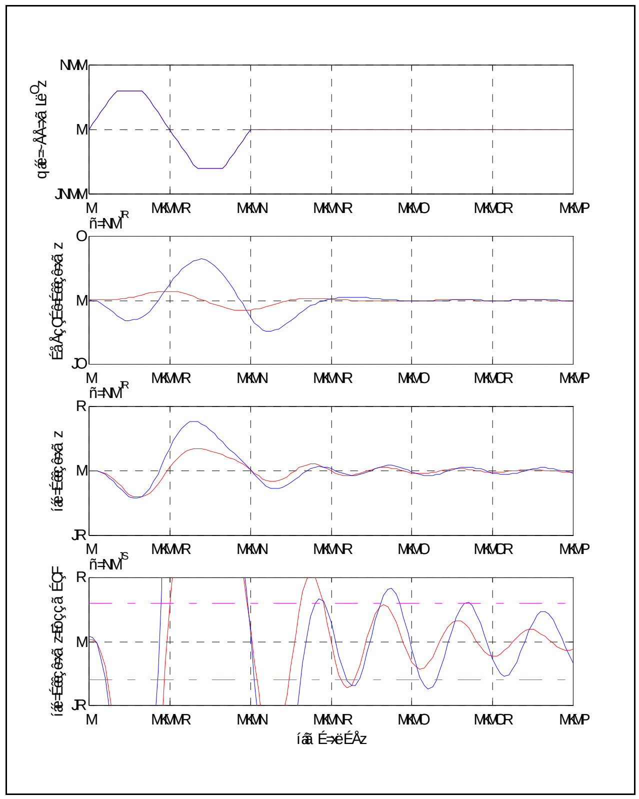


Figure 10 : Frequency dependent (red) versus constant (blue) stiffness and damping

5.3 TIME DOMAIN: POSITION DEPENDENT BEHAVIOUR

In Figure 11 the same typical 10 ms, 1mm stroke setpoint can be seen for different working positions.
The error at encoder level does not differ much. This is due to the fact that the controllers don't see much differences in the various positions (Figure 8 and Figure 9). This is also a measure for the fact that the cross-talk between the two axes is low.
The more forward the position, the better the performance.

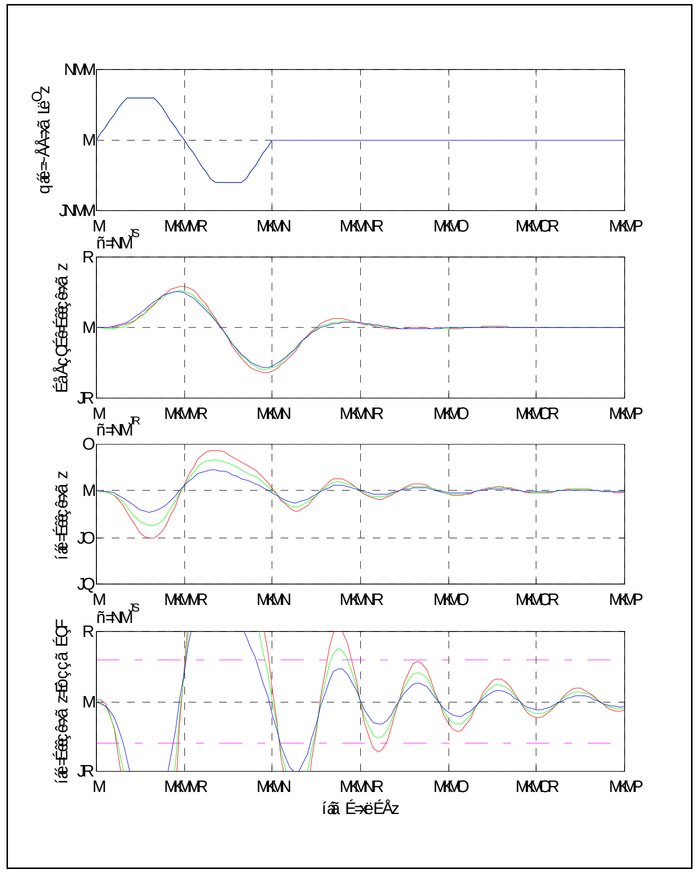


Figure 11 : Performance at three working positions: front (blue), middle (green) and aft (red)

6 Concluding remarks

- Using the frequency dependent stiffness and damping give a better performance then only using a constant stiffness and damping. A first step has been made in the insight of the frequency dependent behaviour of air bearings on performance. It is now possible to make a trade off between low frequent stiffness versus low frequent damping.
- It is possible to use two 1 D encoders, which help keeping the cost low for the XY-table with air bearings .
- Currently the XY-table with air bearings is being made and test results will be available for model verification in December 2000.