ACTIVE VIBRATION CONTROL OF A PAPER MACHINE ROLL

Juhanko,J; Porkka, E.; Kuosmanen, P.; Valkonen, A; Järviluoma, M.

Abstract: In process machines unexpected vibration behaviour may cause severe problems when passing the resonance frequencies. In paper machines it is often impossible to avoid sub-critical rotation speeds because of great number of different type of rolls and continuously changing process parameters. Uneven moment of inertia of the rotor and second harmonic of rotational error motion of the bearings are the main reasons for the second harmonic response at critical frequency which often restricts the running speed of a paper machine. By using active vibration control better runnability, less quality variation of paper and increased life time of roll covers and bearings could be achieved.

Key words: rotor dynamics, active vibration control, roll

1. INTRODUCTION

Resonant vibration of large-scale rotors used in turbines, generators, paper machines etc. is normally an unwanted effect and may cause major problems. Typically there are several rotation speeds (sub-critical speeds) even below the first critical speed where superharmonic hits and excites the narrow band resonance. In process machines unexpected vibration behaviour may cause severe problems when passing the resonance frequencies. In paper machines it is often impossible to avoid sub-critical rotation speeds because of great number of different type of rolls and continuously changing process parameters. Uneven moment of inertia $[^1]$ of the rotor and second harmonic of rotational error motion of the bearings are the main reasons for the second harmonic response at critical frequency which often restricts the running speed of a paper machine $[^2]$. By using active vibration control better runnability, less quality variation of paper and increased life time of roll covers and bearings could be achieved.

2. EXPERIMENTAL SETUP

In this research a servohydraulic device was built for active vibration control of large-scale rotors. The test rotor was 5 meter long paper machine roll, which was supported by standard roller bearings. The device was a rigid frame with three servohydraulic cylinders loading the third bearing unit at the open end (Fig. 1). Measurement arrangement is described in Fig. 2.

![Fig. 1. Test arrangement with hydraulic active damper.](image)

2.1 Control hardware

The control system hardware consists of sensors, analog electronics, digital signal processing board DAP300a/415 with analog output extension board, a standard PC and three electrohydraulic servo valves and cylinders.
Fig. 1. Measurement quantities in x-y directions: laser displacement sensors 1-6, acceleration sensors 7-10, relative motion between the bearing housing and the shaft 11-14, and rotary pulse encoder p. Location of the actuator is marked with red arrows. The control system uses signals p, 3 and 4.

The commands for the signal processor card are given from the PC using DAPL command language and DAPVIEW user interface. The signal processor card is used for generating reference signals for the analog PID controllers, which control the forces of the cylinders.

The oil pressures in both sides of the cylinder pistons in the three hydraulic cylinders are measured with pressure sensors. The analog force signals are generated from pressure measurements with analog amplifiers proportional to the piston areas. The force signals are fed both to the digital processor board and the analog PID controllers. The force control is implemented using servo valves with analog amplifiers. The digital processor card gives the set value signals for the PID controllers.

Manual input to the processor board is provided by manual potentiometers. These are used for force adjustment during start up and shutdown operations.

The vibration of the rotor is measured in horizontal and vertical directions (x- and y-directions) with laser distance sensors (resolution < 1 µm) in the middle (lengthwise) of the rotor. In active damping the force set value signals are then generated in the processor board by analysing the vibration signals and using information of the transfer characteristics of the actuator/rotor system. The force, vibration and manual input signals are fed to AD-converters through antialiasing filters.

The rotation speed of the rotor is measured with a pulse sensor and separate pulse counter electronics with analog output. This pulse counter output signal gives the position angle of the rotor and the rotation speed signal is generated with software in the processor board by counting the number of control intervals which are spent during ten full revolutions. This way the maximum error in measuring the duration of ten full revolutions is one sample interval and maximum error in rotation speed is roughly $f^2 T/10$, where $f$ is the actual rotation speed and $T$ is the sample interval.

2.2 Control method

The aim of the active vibration control is to reduce the measured horizontal and vertical vibration in the centre (lengthwise) of the rotor by controlling the forces of the actuator system with feedback from the vibration measurements. The basic structure of the control method is shown in Fig. 3. The force control loop operates in horizontal and vertical directions and it is fed with sinusoidal set value signals. The frequencies, amplitudes and phases of these set value signals are determined according to the measured rotation speed and rotor displacements in two directions.

Fig. 3. Control structure in active damping [3].

The damping control requires the following operations [4]:
- The frequency response, i.e. frequency dependent gain and phase lag, between
force controls and the measured vibration signals must be known. Hence, frequency response measurements and modelling are needed.

- The amplitude and phase of the base frequency and/or higher harmonic frequencies in the vibration measurement signals must be known. Hence, an estimation method for these is needed. The amplitude and phase should be obtained relative to some reference sinusoid signal. The reference signal frequency is related to the rotation speed of the rotor.

- Automatic adjustment of the phases and amplitudes of the control signals is needed. This uses the above mentioned frequency response models and phase and amplitude estimates of the vibrations to produce forces, which compensate the vibration forces generated by the rotating system.

- To cope with changing rotation speed the above mentioned adjustment should be made adaptive to the measured rotation speed.

If the change in rotation speed is assumed to be slow, the adjustment of the phase and amplitude of the compensation signal can be done with larger intervals than the control interval. This is also practical since the estimation of phase and amplitude requires analysis of large data sets. For this reason the automatic phase and amplitude adjustment is often called open loop or feedforward control instead of feedback control.

The amplitudes and phases of the vibration measurement signals were calculated using Fourier transforms for single frequencies. Recursive adaptive synchronization [4, 5] was used for the adjustment of the amplitudes and phases of the control signals.

3. RESULTS

In the active vibration control tests the resonance response at half-critical speed reduced from 150 μm to 5 μm (Fig. 4). The resonance response at one third critical speed reduced from 35 μm to 2 μm respectively.

Fig. 4. Horizontal vibration spectrums in the middle cross-section without (upper) and with simultaneous active damping (lower) during the acceleration of the rotor. Natural frequency fc (32 Hz) of the roll is excited at rotation speeds of 1/3 and 1/2 of the resonance speed fc.

4. CONCLUSION

In this research active damping at the rotation speed range from one third- to half-critical was tested. The system was capable of minimizing runout in the critical frequency of the test roll (32 Hz). Even several harmonics including the first harmonic could be damped simultaneously. Future work includes integration of the actuator into the bearing unit. This technology makes it possible for example to change the design criteria of paper machine rolls, which reduces the diameter and the mass of inertia of the rolls.
5. REFERENCES


AUTHORS

Professor Jari Juhanko
Professor Petri Kuosmanen
M.Sc. (Tech.) Esa Porkka
M.Sc. (Tech.) Antti Valkonen
Helsinki University of Technology
Department of Design Engineering and Production
Otakaari 4, P.O.Box 4100
FIN-02015 TKK
Phone: +358-9-4513564
Email:Jari.Juhanko@tkk.fi

Lic. Sc. (Tech.) Markku Järiviluoma
VTT Electronics
Kaitoväylä 1, P.O.Box 1100
FIN-90571 Oulu
Phone: +358-8-551 2111
Email:Markku.Jarviluoma@vtt.fi