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Delay-resonance Control of Roll Press by Speed Variation Approach

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Abstract

Roll pairs in rolling contact are commonly used in paper machines. To improve the process at least one of the component rolls is covered by a soft layer, which is the case in calenders and coating units. The larger contact area allows longer manipulation time in the nip, and therefore faster line speeds can be achieved when using such soft covered rolls compared to hard non-covered rolls. This improvement was widely utilised in start-ups of fast next generation machines late 90’s. Several examples, however, proved that the new polymers in the cover materials were a source of new-type of oscillations arising at higher running speeds making it difficult to reach the planned production rates. As understood quite early, the reason for these instabilities is the exponential and thus too slow recover of the cover penetration during each roll revolution leading to self-regenerative normal oscillations of the roll pair. Further investigations indicated that: 1) these instabilities exist at certain discrete running speeds beyond a critical speed level, 2) the vibration frequency was always the natural frequency of the roll stack and 3) the highest amplitude peaks correspond to rotation frequencies, which are integer fractions of the natural frequency. These rotational frequencies are called delay-resonance speeds referring to the feedback mechanism of the one revolution earlier cover penetration.

Dynamical instability of rolls is not accepted as it marks the roll covers by wave formations, which in turn cause variations in the line load, and deteriorates the quality of the product. Vibration also overloads mechanical structures by means of fatigue and can cause damage of vibration sensitive components.

This thesis is introducing methodology and tools to control the self-excited delay-resonance vibrations at speeds, which are higher than the critical speed level making it possible to extend the production speed range. The methodology is based on systematic variation of the running speed by three different ways: 1) by setting the speed to fixed value between two successive resonance speeds, 2) by active change of the speed for avoiding the regular formation of waves on the cover and 3) by controlling phase-shift of rolls to damp cover-induced vibrations. The control methods are verified in a laboratory unit scaled down to half size of the existing industrial units.

Preface

The work presented in this thesis was carried out at the Laboratory of Machine Dynamics at Tampere University of Technology. The research idea was composed during the projects funded by the Technology Development Agency TEKES, which is gratefully acknowledged, as well as Graduate School Concurrent Mechanical Engineering for the financial support. My thanks go also to the research partners of the TEKES projects Metso Paper Inc. Järvenpää, ABB Industrial Drives Helsinki, M-real Oy and Stora Enso Fine Paper Oulu Mill.

I am grateful to the advisor of this thesis, Prof. Erno Keskinen for his encouragement, support and guidance during this thesis. I would also like to thank Docent Juha Miettinen for his advice and participation during the work as well as Dr. Lihong Yuan for her kind help in simulation and modeling.

The assessors of this manuscript, Professor Matti Pietola from Aalto University, Professor Jussi Sopanen from Lappeenranta University of Technology and Dr. Timo Holopainen from ABB, are kindly acknowledged for their valuable comments to improve the manuscript. In addition, my warmest thanks to Mikko Enqvist for revising the language of the manuscript.

I wish to express my warm gratitude to my parents Hannu and Tuija for their encouragement during this study. I dedicate this thesis to my dear, Heli, for the patience, love and support during the course of this work as well as for my sweet daughter Eevi.

Tampere, December 2013

Pekka Salmenperä

Contents

Nomenclature

A_p, A_m	pressure area at plus and minus chambers
B	bulk modulus of fluid
c_1	damping coefficient
c_f	valve coefficient
c_n	nip damping
d	inside diameter
E	elastic constant
f	rotational frequency of polymer roll
f_{barr}	barring frequency
f_{barrex}	barring excitation frequency
f_{nat}	natural frequency
F	hydraulic actuator load
F_d	desired nip load
g	acceleration of gravity
h	roll cover thickness
j_w	number of sinusoidal waves
k	torsional stiffness of drive
k_1	stiffness
k_h	constant of hydraulic spring
k_n	nip stiffness
K_p	proportional gain
ℓ	length of nip line
m_1, m_2	mass of upper and lower roll
N	nip load
p	contact pressure
p_p	pressure at pushing chamber of the hydraulic cylinder
R	electrical resistance
t	time
T	roll revolution time at production speed
T_c	cover temperature
u	control input to proportional valve
v	paper web line speed

w	weight per length
x	element of the fuzzy set $\tilde{A}$
x_d	displacement matrix
x_1	displacement of upper roll
x_2	displacement of lower roll
z	shape excitation
δ	compression in cover
ε	cover penetration
ϕ_1, ϕ_2	angular position of drives
η	viscosity of polymer
γ	delay factor
μ_A	membership function
$\mu_A(x)$	degree of membership of element x to the fuzzy set $\tilde{A}$
τ	roll revolution time
τ_{cr}	critical roll revolution time
τ_r	time constant in cover recovery
ω	angular frequency
Ω	angular frequency of excitation
θ_1, θ_2	angular position of upper and lower roll
ψ	angular coordinate at roll surface
$\tilde{A}$	fuzzy set
A_{beat}	amplitude of beating
CD	machine Cross Direction
DE	Drive End of roll
DTC	Direct Torque Control
FEM	Finite Element Method
FFT	Fast Fourier Transform
MD	Machine Direction, direction of material flow through nip
MEC	Department of Mechanics and Design
RPM	Revolution per minute
RMS	Root Mean Square value
TE	Tender End of roll

1. Introduction

1.1 Background and motivation

Rolls are widely used in web handling processes. A common unit process is based on the use of two rolls in line contact rolling against each other and the raw material is traveling through this narrow contact area. This kind of roll contact is called the nip contact. Nipped rolls are typically used in paper machines, printing machines, in aluminum folium mills and in rubber industry machinery. In metal industry the line contact is usually between two steel rolls. In paper machine sections, like in calenders and coating units, soft roll applications are more common. The schematic drawings of the calender nip and the coater nip of a paper machine are shown in Figure 1. The line load between the rolls is typically generated with different kind of mechanisms actuated by fluid power cylinders. Coverings are used to give more radial compliance for nipped rolls. It softens the contact and even relaxes the stresses inside the material. This decreases product thickness variations and enhances the quality of the paper web.

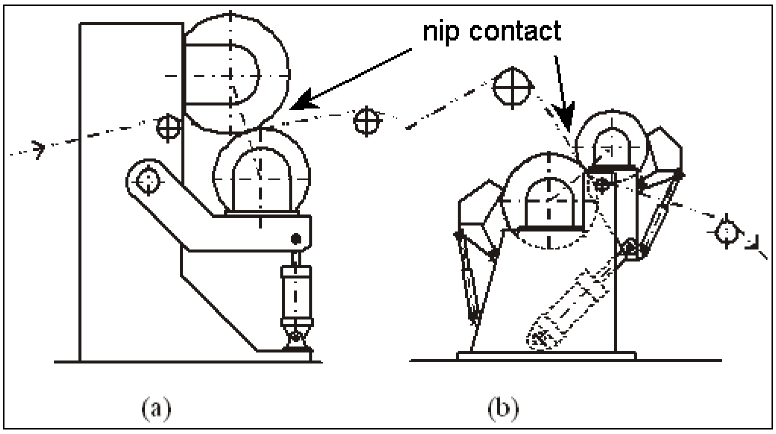


Figure 1. a) Calender nip and b) coater nip of a paper machine (Keskinen 1998).

A combined paper machine and coater line as shown in Figure 2 is a representative example of a machine configuration with single or multiple soft nips in various sections for coating and calendering.

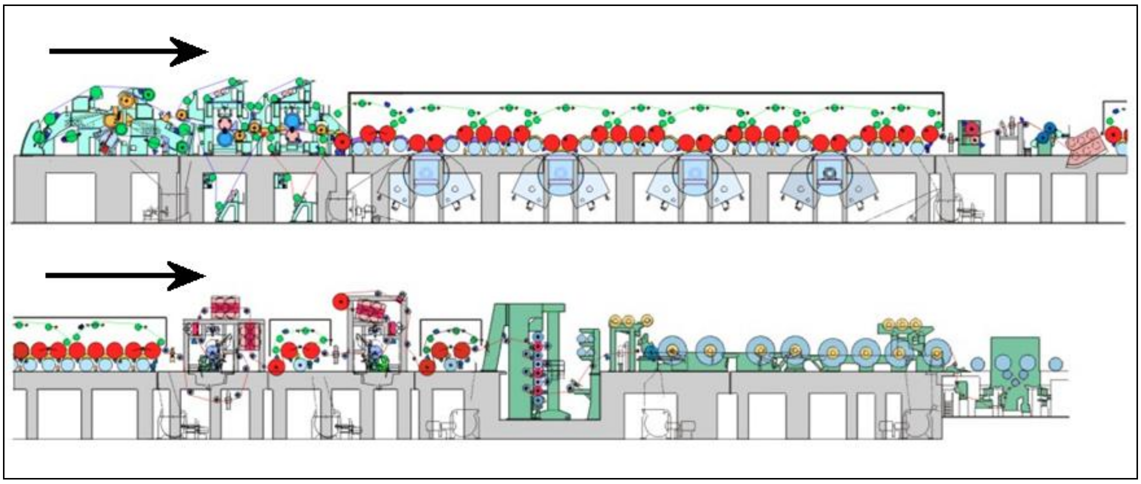


Figure 2. Combined paper machine and coater (Metso Paper Inc.).

Contact vibration in a roll pair is caused by normal motion of rolls and some vibration is always present when the rolls are running. This vibration driven by internal or external sources (Figure 3) causes unexpected variations in line load and can deteriorate the quality of the end product or damage the surface of the roll. In addition, vibration causes noise, overloads mechanical components and can cause damage of electrical components. The problems that especially are associated with rolls in contact can be classified to internal and external excitations (Salmenperä, Miettinen 2003, Keskinen et al. 1998). In the case of paper machines, internal vibration sources are the unbalance forces and out-of-roundness of roll cover profiles and external vibration sources are the periodic paper thickness variations. Periodic thickness variations are not necessarily related by an integer number of wavelengths to the circumference of any of the rolls (Emmanuel 1985). Internal and external excitation types are presented in Figure 3.

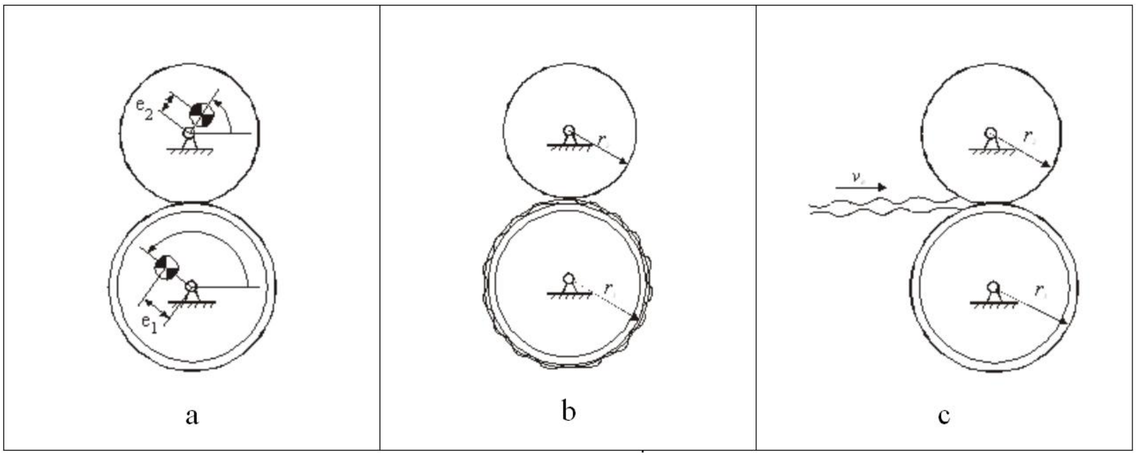


Figure 3. Sources of vibration in nip contact. Internal sources: a) roll unbalance and b) out-of-roundness of roll cover profile. External source: c) paper thickness variations.

In some running conditions the nip vibration has a nature of resonance vibration. This means that the amplitude of the vibration can rise in very high values. A difficult resonance vibration situation is the self-excited delay-resonance. The delay resonance exists especially in soft nip contact and the source mechanism comes from the incomplete recovery of the penetration of the soft roll cover during one revolution of the roll. Figure 4 represents the situation of the delay resonance. Above a certain critical rotational frequency the time of one revolution of the roll is not long enough for the recovery of the soft cover penetration. In this situation the nip penetration begins to work as an excitation and becomes larger in amplitude after each roll revolution. If in this case accidentally the integer number multiple of the rotational frequency of the roll is the same as the natural vibration frequency of the nip contact, the self-exited delay resonance starts (Vinicki 2001, Yuan 2002). This means that there is some threshold rotational frequency above which several delay resonance states can be found when raising up the line speed of the machine. This limits the production speed in real paper machines.

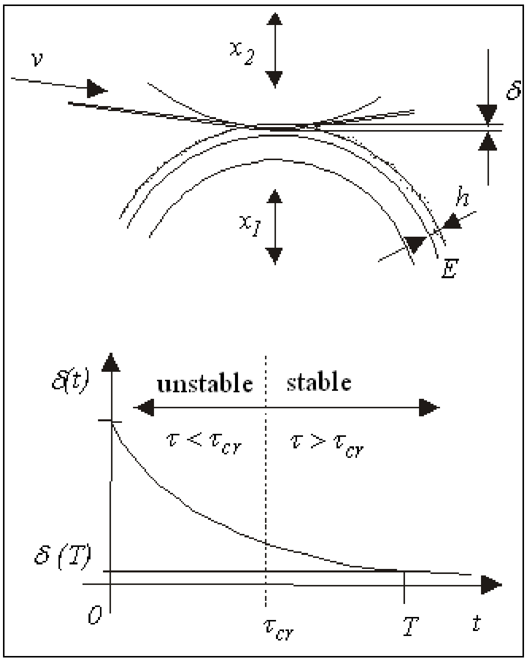


Figure 4. Exponential recovery of cover penetration δ. Unstable delay resonance states can exist with roll revolution time τ shorter than a certain critical revolution time τ_{cr}, above which a stable region exist. This enforces the paper mills to use long enough revolution times *T*, which correspond to lower speeds and reduced production rate.

The delay vibration of the rolls can cause machine cross-directional (CD) line patterns on the surfaces of the rolls. These are called barring marks (Chinn 1999) and they can appear on the surface of one or both rolls. If these phenomena are strong enough they damage the surface of the roll rather quickly. This is due to a cumulative deformation in the cover material which will form a regularly wavy shape.

The delay resonance vibration of roll contact is a common problem with continuously increasing speeds of production (Kustermann 2000). Nipped rolls may also have low resonance frequencies because nips are typically much softer than other machine elements and the fluid powered loading circuit makes the effective support stiffness of the movable component roll much lower than the bearing stiffness of the non-movable backup roll. The large width of paper web creates problems in terms of long and flexible rolls. Contact vibration weakens the quality of the product and limits production speeds. It is possible to influence on the vibration behavior by design aspects. Large roll diameters, roll pairs with different diameters or cover materials are used in order to decrease vibration. Structures can also be designed to reduce vibration by increasing or decreasing mass or stiffness in appropriate points of machine. In addition, certain class of actuators or accessories can be used to attenuate vibration. These devices have been classified to passive, semi-active and active vibration dampers. The purpose of such methodologies has clearly been: 1) to improve the stability at the design speed area and 2) to extend the speed area from the original one. Such developments have mainly been successful, but in some cases total costs of additional investment together with the value of lost production during assembly brake have been criticized.

To avoid such costs and losses, a question arising from practical needs has been set:

Could it be possible to combine the on-line information of the system state and the theoretical knowledge on the response behavior in such intelligent way that a roll system could be driven in the required speed range just by avoiding the forbidden speeds?

There are few known examples of some other processes starting from rock-drilling in mining industry and ending to machining in mechanical workshops, in which the systems can be driven softly with high performance or less effectively with high noise and vibration by selecting the drilling or machining speeds differently. Such vibration-critical speed setting problems are common also in transportation sector.

When practically all other material-based solutions for reducing roll nip vibrations have already been evaluated, this purely algorithm-based approach could open a totally different and promising research line. Moreover, such immaterial solution strategies belonging to intelligent technologies are today popular in many sectors and activities of modern society.

1.2 The research problem

As pointed, there is a clear need to develop methods, which make it possible to reach the promised design speeds of new start-up lines or extend the production speed range of existing machine lines to higher speeds by means of an update package. In order to build a clear picture, which speed values are possible to use and which one should be avoided, the mechanisms behind the delay-resonance behavior of soft nips have to be scientifically explained. The next step will then be a systematic generation of an algorithm family to manage this vibration control problem in a real production line.

This problem setting leads to a set of research questions, which have to be opened and solved in order to develop scientifically argued algorithms:

1. Which mechanism is behind the wave formation of roll cover and which factors are controlling this phenomenon?
2. Is the behavior of cover polymer regular enough to produce repeatable response behavior?
3. How does the spectrum of resonances look like and how are the frequencies distributed? How are the fixed running parameters modifying the spectrum?
4. How can the resonance speeds be experimentally detected? Is there enough space for stable run in the speed domain?

Figure 5 illustrates the research field focusing to factors controlling the nip vibration, to damping techniques and to intelligent vibration control, which is the target of this thesis.

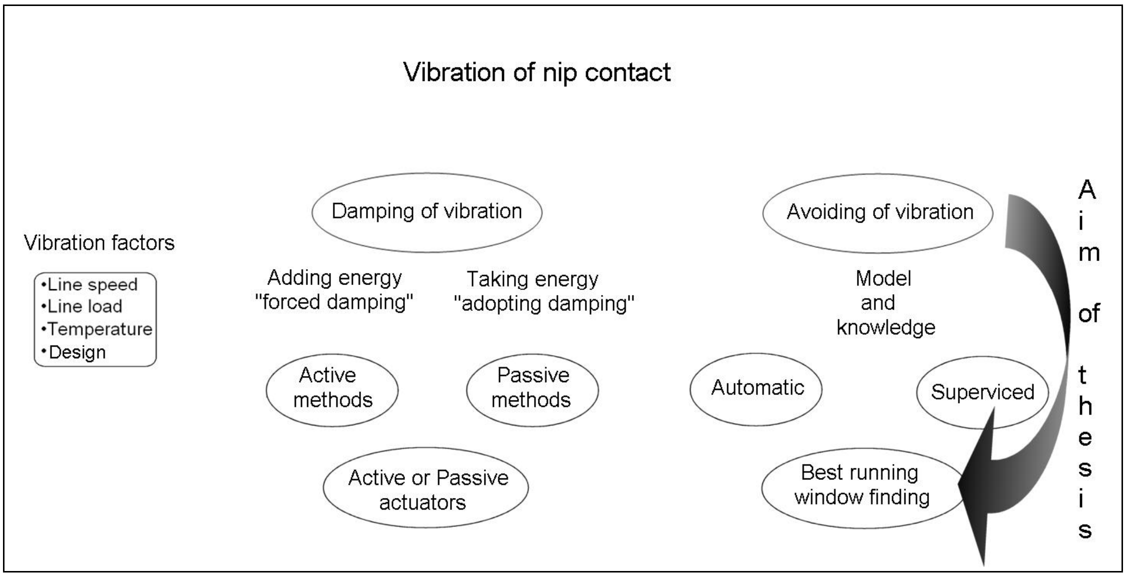


Figure 5. The field of the research problem including the control of nip contact vibrations by damping methodology or by intelligent speed control.

One of the key questions in developing the control strategy is the understanding of the feedback mechanism between the excitation and the response. It has been accepted that, as in any vibrating mechanical system, the natural vibration of the nip contact can be initialized by external random shocks or excited by broad-band background noise. Such motion, in which one fixed frequency dominates, generates sooner or later regular wave-valley formations to the roll surface. The accumulated deformations will mark the surface the faster the better the multiple of the rotation frequency is matching to the natural frequency. If the nip rotational frequency is set to lie between two successive integer fractions of the natural frequency, the contact-driven excitation and the natural vibration will be desynchronized and resonance vibration can be compensated or totally avoided. This is the most straightforward way to manage the resonance under manual control. More advanced control is based on systematic avoidance of the rotation frequencies corresponding to the integer fractions of the natural frequency, which leads to the selection of favorable running speed values.

The second strategy is based on slow oscillation of the running speed, the purpose of which is to eliminate the regular wave formation mechanism. In this method, the integer number of roll surface deformations is switched between another integer number, typically with the nearest smaller or nearest larger in the set. This procedure is generating a wave formation with varying wave length. By avoiding the match between the multiple of the speed oscillation frequency with the rotation frequency, the rolls can not repeat the same cover marking history and the condition for a fully developed nip vibration is effectively eliminated.

The third method for avoiding resonance vibration studied in this thesis is the transient shift of the phase of roll vibration compared to roll deformation. Only a very small delay of the phase of the rotation of the roll is needed to decrease the resonance vibration, as can be seen in results of this study. A clear technical constraint in this method is the performance limit of the drive control.

It has been identified six dominating parameters, which have an influence on vibration behavior of the nip contact in terms of resonance vibration amplitude and frequency. These parameters are the stiffness of the nip contact, stiffness of the hydraulic loading circuit, recovery speed of the cover material of the soft roll, running temperature of the cover, line load of the contact and rotational frequency of rolls or line speed. However, three of these parameters can change online and can be modified or measured online in the test nip. Nip load and line speed are parameters which can be changed online while temperature can be measured through non-contacting techniques. Nip stiffness and recovery speed are functions of nip load and roll cover temperature. The increase of temperature softens the nip contact and thus lowers the resonance frequency. Correspondingly in higher temperatures, the resonance starts with lower roll rotational frequency. However, stabilizing temperature is a slow process, even if a backwater system is available for temperature control.

Line load has also a strong influence on the resonance, in terms of frequency and amplitude. Unfortunately, line load changes are usually prohibited in production machinery. The rotational frequency of rolls or line speed is the essential element in avoiding the resonance state. The line speed changes are allowed for off-line paper machinery, while in on-line machinery combinations, the chance of line speed of one unit or whole line is not simple to carry out. The adjustment of line speed of one unit can lead to conflict with running speed constraints coming from another unit.

When the system response is depending on so many different design and running parameters, it is necessary to identify the operational state by means of a monitoring system.

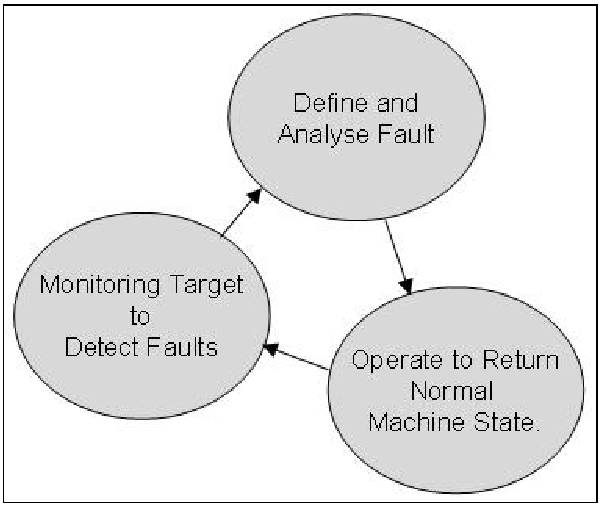


Figure 6. The idea of the diagnostic and control method in the thesis. (FDIR 2002).

Monitoring information is used to make diagnosis of the system state – in this case to detect whether it is in a delay resonance state or not. If a variation from normal running behavior is the result of the diagnosis, an action based on reasoning will change the running parameters to slide the system to acceptable state. This control loop developed in this thesis is illustrated in Figure 6.

1.3 Contributions

The purpose of this thesis has been to develop intelligent line speed variation methods for avoiding the resonance vibration of roll nips. The control methods avoid resonance by introducing techniques of reasoned speed changes under a supervised control. The decisions in such control approach are based on mapped data of vibration experiments taken against a wide enough running parameter space and on theoretical knowledge about model-based resonance states. Because the parameters may vary during the process thus shifting the resonance speeds, an observer block monitors changes and makes necessary operations for keeping up optimal drive speed. The developed methods are verified and argued by a large program of experimental tests with the pilot roll press and by utilising the system model of the same unit.

The solution of the research question led to coupling work of two interacting scientific sub-problems:

1. Experimental and theoretical response analysis to explain phenomena behind the roll nip vibrations and for building the knowledge basis for vibration control.

2. Systematic generation and implementation of intelligent ways to vary running speed in order to avoid falling to delay-resonance vibration state.

The developed control methods have been implemented and verified with the pilot roll press. Vibration control principles in the thesis are based on variations in line speed while line load and temperature are the given fixed process parameters. Developed methods are suitable for hard-soft or soft-soft roll contacts, in which cases the cover response is dominating the nip interaction.

The following contributions were developed in the course of this work:

1. Development of three different line speed variation methods for systematic avoidance of the delay-resonance vibration.

2. Implementation of the delay-resonance avoidance methods for direct drive roll units.

3. Systematic analysis of the influences of running parameters on response behavior of rolls in delay-resonance.

4. Utilisation of theoretical system model to predict the regular distribution of delay-resonance speeds.

The research work has been mainly carried out during the following research projects funded by TEKES:

Polyroll, 2003. Dynamics and operation monitoring of polymer covered rolls.
Activeroll, 2005. Active control of rolling contact.
Smartroll, 2007. Dynamics and control of direct drive rolls

2. Rolling contact mechanisms

2.1 Literature survey on roll system vibrations

As long as rolls have been used in production purposes of web-like products, vibrations have been present. Rolls are circular cylindrical bodies with large length to diameter ratio. When such elastic objects are rotating on bearings with varying speeds, there are two kinds of dynamic responses to be observed. The first is the bending effect of the roll itself making the roll to experience whirling motion in a deformed state and the second one is the vibration of the structure, on which the bearing houses are mounted. These motions are usually driven by imbalance forces, by roll-roll interaction forces or by roll-tool interaction forces. Accordingly the vibrations of rolls can be classified primarily to two categories:

A. Vibrations of individual rolls
B. Vibrations of rolls in rolling contact

A. Vibrations of individual rolls

These vibrations appear in manufacturing phases and in web or felt guiding positions in paper machines. In manufacturing, the purpose is to reduce mass imbalance and to reach smooth enough surface. When the roll is a long body, the imbalance forces are distributed on the roll span. Such balancing problem leads to application of multi-plane (Bishop 1959, Lund 1972, Vance 1988, Ehrich 1992) or continuous balancing mass techniques (Keskinen 2002a).

Manufacturing of the roll surface includes rough turning, fine turning and grinding phases. When the tool is in contact with the surface of the rotating roll, the material removal process leads to similar delay-controlled contact force problem than the rolling contact problem of polymer covered rolls represent. The physical reason for this well known chatter vibration (Merritt 1965, Hahn 1977) is the overlapping of the machining zones belonging to successive roll revolutions. The solution strategy to reduce chatter vibrations is the on-line variation of process parameters: time delay (roll revolution time), chip thickness, axial tool speed etc. (Moon 1998, Altintas 2000, Xu 2004, Suh 2013, Yuan 2002a, Yuan 2002b). The result of chatter vibration is a wavy spiral street covering the whole roll surface. This finishing profile remains as a bottom disturbance of the surface and can be one of the mechanisms initializing the nip vibrations in the later rolling contact situations. Due to the similarities between the chatter vibrations and nip vibrations of soft polymer rolls, the published chatter literature has a specific role when developing vibration control methods.

B. Vibrations of rolls in rolling contact

Nip vibrations and the corresponding marking of the roll surfaces and/or the web by axial stripes have been widely reported in magazines published by the pulp and paper industry (Matomäki 1963, Wahlström 1963, Parker 1965, Emmanuel 1985, Bradford et al. 1988, Hermanski 1995, Shelley 1997,) and with some delay also in academic journals (Sueoka 1993, Vinicki 2001). When fast paper machinery have been built everywhere since middle nineties, the unexpected behavior of new polymer covers in rolling contact opened a totally new research and publication period. The papers can be classified to those explaining the instabilities (Chinn 1999, Keskinen 2000, Yuan 2002a, Yuan 2009) and to the ones introducing some technical devices or methodologies to overcome the instabilities (Kustermann 2000, Virtanen 2006) thus improving the response behavior at higher running speeds. Cover material responses at high strain rates have been investigated and reported in scientific papers as well (for instance Vuoristo 2002).

Polymer covers and systems damping the vibrations have been under high innovation activity. In contrast to the recipes of polymer composites, which have been carefully protected by the manufacturers, the damping mechanisms have been published in patent applications and in industry driven seminars. When using patent applications as a source material, one has to consider that these innovations have normally not been built nor evaluated in practice at the date when they have been introduced.

2.2 System description

Many production processes utilise roll nip treatments. In paper industry processes like calendering, coating and printing are the most typical ones. However, nips are very intolerant systems from a viewpoint of vibration. Rolling contact is a complex process, where the system response depends on the fixed design parameters of the rolls, roll covers and loading mechanism as well as on the adjustable running parameters including the control gains. In order to drive a nip in controlled conditions, a pilot roll press (Figure 7, Kivinen 2001) has been down-scaled to half size from the ones used in industry exhibiting unstable response behavior by means of delay-resonance vibration. Early start-ups showed, that similar resonance phenomenon was successfully incorporated to the laboratory unit providing an excellent environment for the experimental analysis as well as for the control of the resonance states.

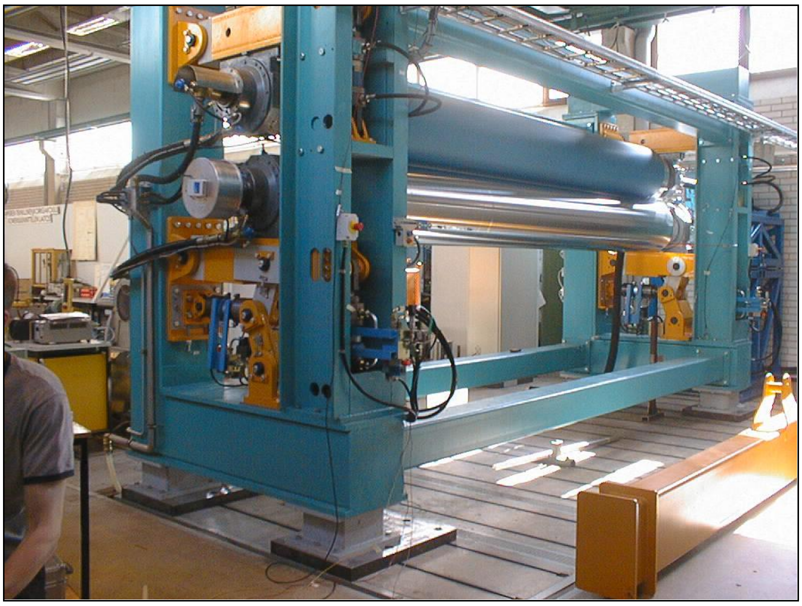


Figure 7. The pilot roll press at TUT machine dynamics laboratory.

Roll drives can be roughly divided to geared and direct drive systems, where the electric motor is connected to the roll with a coupling or by integrating it to the roll end. Control demands have an influence on the selection of drive type. Control performance can be measured for example by control accuracy of static speed regulation. The most effective, so called tight drives can hold speed to within 0.01 % from the line speed value (Roisum 1998). This is important because web tensions must be held within given tolerances to avoid web loosen or break.

Two types of drives were installed consecutively into the test nip system. The first drive type was a traditional AC motor drive with vector control including gear transmission and the second drive was a new type permanent magnet AC direct drive motor. First one is a general solution in paper machinery, latter one is a more sophisticated and modern drive, whose large operating range eliminates the need of gearboxes. Also, because of the reduced mass of the permanent magnet motors, they can be installed directly to roll ends instead of using separate bed for motors and gearboxes. The geared and direct drive assemblies of the pilot roll press are shown in Figure 8. The advantages of the direct drive solutions is to decrease the sensitivity of the roll system to torsional oscillations, to reduce the number of components in the power train and save floor space when customized beds are not needed for a complete roll drive unit as found out from TEKES project Smartroll - Dynamics and control of direct drive rolls.

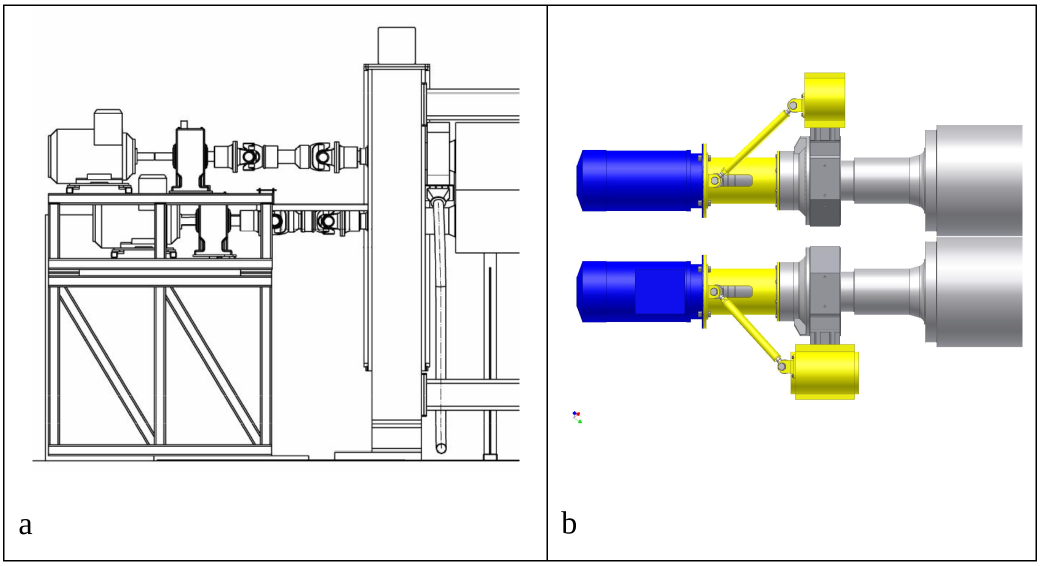


Figure 8. Geared (a) and gearless (b) drive units of the pilot roll press used in this investigation.

In direct drives the control strategy is called Direct Torque Control (DTC) (ABB 2010). With DTC technology the orientation of the magnetic field is achieved without encoder feedback from the rotational frequency of the motor. The DTC control algorithm calculates the motor state e.g. torque and magnetic flux by 40 kHz frequency (ABB

2010). The controlling variables are motor magnetizing flux and motor torque. With DTC there is no requirement for a tachometer or position encoder to feed back the speed or position of the motor shaft. Because torque and flux are motor parameters that are being directly controlled, there is no need for a modulator, as used in conventional drives, to control the frequency and voltage. This, in effect, speeds up the response of the drive to changes in required torque. DTC uses digital signal processing and advanced mathematical description of motor functions. The result is a drive with a torque response that is typically 10 times faster than with conventional AC or permanent magnet AC drives. The resonance vibration avoidance methods are tested with the direct drive design.

The power input of motors is consumed by elastic deformations in soft roll cover, roll accelerations, thermal losses caused by dissipation in soft roll cover, bearing frictions, damping of loading actuator and frictions of other interfaces. Experiences from industrial units and from laboratory size sheet calenders show that web corrugation can be avoided by eliminating the tangential traction between the rolls. This is: the rolls are not allowed to be driven over the nip friction. This situation can be reached by driving the rolls with almost similar torques. Hard roll rotational frequency is usually controlled with speed control, so hard roll is the master and the soft roll is controlled with torque control, so soft roll is the slave. This principle helps to synchronize the surface speed of the rolls with the line speed, because the effective radius for hard roll is known.

Paper web entering the nip needs certain amount of time for proper processing between the rolls. Because machines tend to reach ever higher production speeds, the nip manipulation time might get too short with solid cast iron rolls. To increase the nip contact time, soft rolls are nowadays widely used. The soft roll cover distributes pressure in the nip more uniformly and produced paper’s density is more constant (Jokio 1999).

In film size presses or in calendering units, the nip load is one of the main control parameters. Nip load is conventionally produced by hydraulic actuators at both ends of the roll and is usually expressed as nip force between the rolls per unit width of the roll or width of the web between the rolls. A typical nip loading system is schematically presented in Figure 9. Cylinder pressure creates force to loading arm counteracting to the weight of the lower roll and the arm. Nip force can be calculated from geometry of the design. A fixed bottom roll would be easier to design, but loaded bottom roll design provides more safety because loading and unloading the nip is quicker and easier just by closing or opening the nip by moving the lower roll with the hydraulic cylinder.

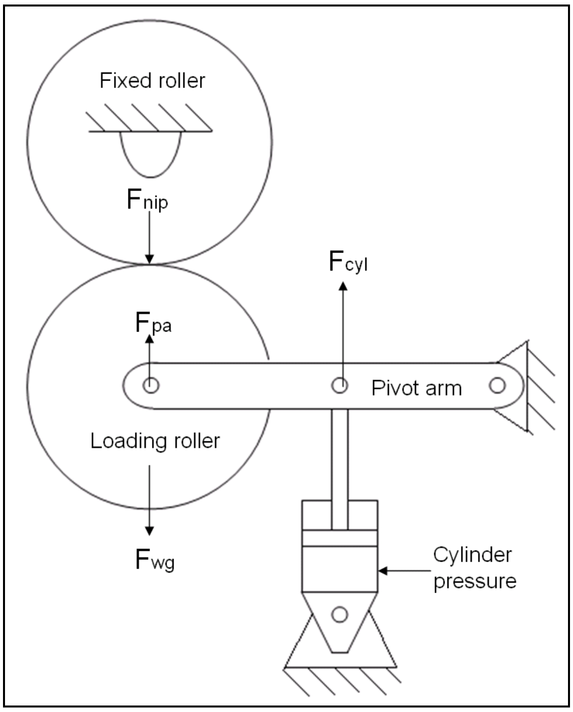


Figure 9. Schematic example of a typical nip loading system.

The nip is very sensitive to disturbances, and main problems are controlling the load as well as keeping the run free of vibrations (Lehtinen 2000). The nip load in machine cross direction (CD) should be an even line load to confirm uniform product quality. There are typically two ways to ensure even CD line load. The methods are crowning of rolls and zone-controlled rolls. In crowning, the roll is ground into a barrel-shaped form to compensate the deflection of the tubular roll structure mainly arising from beam bending and shell flattening effects. Crowning is performed for a certain nominal nip load. If nip load is varied, also the nip CD load profile varies and loses its uniform shape. In zone-controlled rolls this compensation has been done by using a set of hydraulic shoes mounted on equidistant locations at roll span and pressing the counter roll to produce a constant load over the whole nip line.

2.3 The design and instrumentation of the pilot roll press

A pilot unit designed to the scale 1:2 as compared to the corresponding mill unit operating in a specific industrial plant has been built for experimental testing of roll interactions under rolling contact conditions. The system consists of a frame, two rolls in nip contact, joint-supported arms on which the upper roll is mounted, hydraulic loading mechanism for generating the nip load and electro-mechanical drives for rotating rolls. The nip load mechanism lifts the lower roll up into the contact with the upper roll, see Figure 10. The line load characterizing the contact is evaluated by using load cells, which are at the moving ends of the support beams. This loading system is able to produce more accurate line load generation than conventional mill units, which utilise hydraulic pressure sensing in line load control loop.

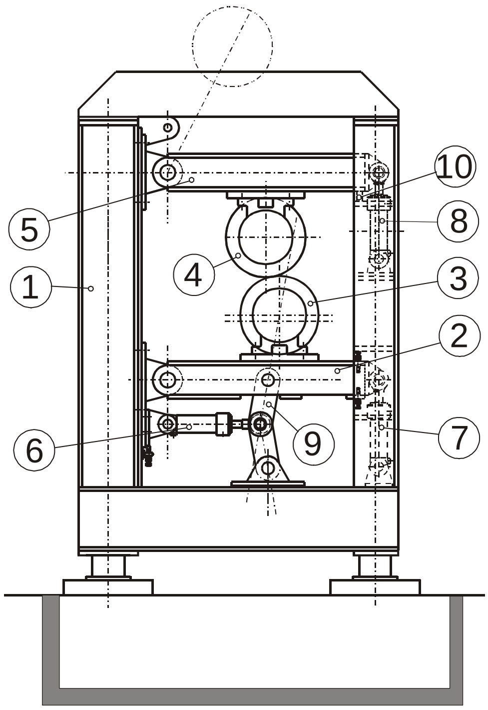


Figure 10. The components and structural parts of the pilot roll press: frame (1), loading arm (2), lower hard roll (3), upper soft roll (4), support arm (5), loading cylinder (6), alternative loading cylinder (7), locking cylinder (8), loading mechanism (9) and load cell (10).

The roll press is equipped with a fixed online operation monitoring and diagnostic system. Furthermore the hard roll contains internal temperature, acceleration, acoustic emission and strain gauge sensors. The measurement signals are transferred from rotating roll via wireless local area network bus into the measurement computer for further analysis. The role of these measurement facilities is to give enough information on system dynamics during running and cover the actuation and the physical aspects, deformation, vibration, stress and temperature evolution. The soft roll is equipped with temperature sensors in the metallic part of the roll and the information is sent through a wireless radio link to the computer processing unit, where the signals are saved before analysis. The technical data of the pilot roll station is presented in Table 1.

Table 1. Technical data of the pilot roll press.

Roll casing width	4.4 m
Roll diameters	0.525 m (hard roll) and 0.547 m (soft roll)
Electric drive motors	2 x 55 kW and 2 x 11 kW (both used consecutively)
Line load	up to 50 kN/m (with soft cover up to 25 kN/m), present rolls design
Roll mass	3.5 tons
Total mass	15 tons

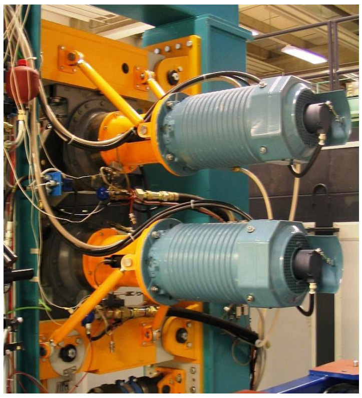
The soft roll has the same original diameter as the hard roll, but after coating of 11 mm, the diameter reaches 0.547 m. The coating is polyurethane, which is typically used in applications, where soft contact and abrasion resistance are required for coating (Roisum 1998).

During this study, three different types of drive unit solutions have been tested in pilot roll station, in order to measure their effect to roll vibrations. Speed control properties are different, and different motor types were tested with proposed speed control methods. Three tested configurations were:

- conventional AC motors with gearboxes and universal shafts
- conventional AC motors with universal shafts, without gearboxes
- direct drive permanent magnet AC motors installed to the end of the shafts

Measurements for resonance control in this study were performed with the last configuration, utilising direct drive motors. In Figure 11 the drive unit with conventional motors without gearboxes and the direct drive installation are shown. Motors are connected to rolls with universal shafts.

a



b

Figure 11. a) Conventional AC motors, where gearboxes have been removed and only universal shafts are in use, b) direct drive AC motors fitted on the ends of the rolls.

A high temperature of the soft roll may make the cover more tender or even thermally degrade it. On the other hand, many processes like calendering, embossing or laminating benefit from higher temperature, because high web plasticity makes these processes more effective. Hysteresis of cover deformations in nip contact heats up rolls and also heating or cooling systems are common in nip applications.

The pilot unit is equipped with heat exchanger with 30 kW cooling power. A simple monopass channel for fluid flow goes through the roll and further to a water tank, see Figure 12. A backwater pump circulates water back into the roll. The pump produces inverter controlled volume flow and is equipped with flow measurement. System is also equipped with 20 kW resistor elements for water heating. Corrosion prevention agents are added into the water. Temperature of the water is measured from points where it enters the roll as well as where it comes out from the roll. Temperature control is performed with a controller.

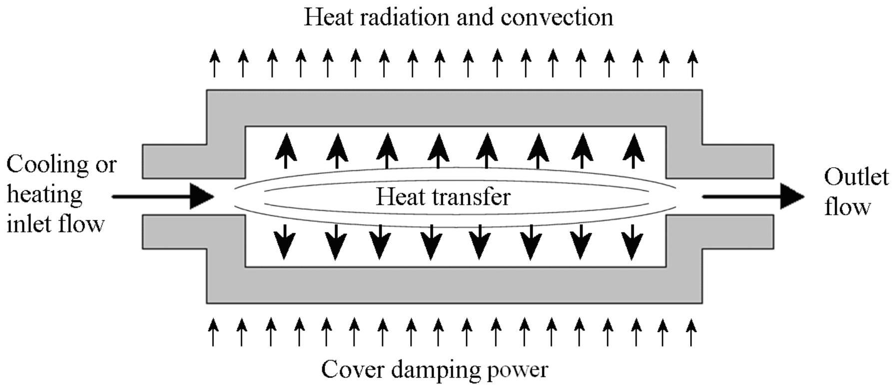


Figure 12. Simple monopass fluid flow trough the soft roll.

However, in this study, no cooling or warming of the roll was used. Instead, only nip hysteresis warms up the rolls. Reason for this is different temperature gradients inside the rolls depending on the methods. Backwater warming or cooling has an effect on the rolls from inside direction, when hysteresis mostly warms up the soft cover material. This study considers roll temperature and nip operation dependencies, and therefore it is reasonable to warm the rolls only by one method, since only roll surface temperature is measured.

Temperature is measured with infrared sensor. Measurement cone is 2:1, corresponding to 0.25 m diameter wide measurement area with distance of 0.5 m. The sensor is mounted to measure temperature in the middle of the soft roll. Current signal of the sensor is changed into voltage signal with the resistor arrangement presented. The voltage signal is calibrated with touching surface temperature measurement device.

It is suggested that the most practical method to measure coating temperature is to use an appropriate contacting thermometer immediately after the line has stopped (Roisum 1998). However, testing phase of temperature measurements showed that cover temperature can warm up a few degrees of Celsius immediately after stop. This is probably due to the accumulated temperature inside the coating, which is not cooled anymore by air when rotation stops.

Data acquisition is performed via data acquisition card having a 14 bit resolution and maximum sampling rate of 48 kS/s.

Vibration was measured with B&K accelerometer sensor and B&K amplifier. The measurement chain was calibrated to correspond 0.1 V/g. Attachment of the sensor is magnetic and measurement angle 10° from vertical direction, corresponding to nip orientation angle. The sensor is situated on bearing housing of top soft roll tender end. The amplifier is equipped with high- and low-pass filters, which were set to 100 Hz and 300 Hz -3dB limits respectively. Limits were selected to detect resonance vibration of about 120 Hz at a suitable bandwidth. Lower frequencies than 100 Hz contain noise as well as frequencies over 300 Hz.

2.4 Vibration characteristics of roll press

One of the dominating parameters of the nip unit is the natural frequency of the rolls in normal motion. Because this depends on the whole design of the press unit, it can not be estimated with acceptable accuracy by elementary beam bending formulas, which way is a common mistake as the rolls are tubular beams at the first look. So, the bending frequency of an individual roll has not much to do with the lowest frequency of the whole press system representing rather a multi-body system. Depending on the radius to length and thickness to radius ratios, the shell flattening effect may be present leading to completely different shell frequencies and shell modes when the rolls are in line contact when comparing to the ones of a separate beam. The accurate and reliable enough methods to determine natural frequencies of even simplest roll presses are modeling based methods like finite element method (FEM) to support the design, or, experimental methods like modal analysis to investigate the existing units.

Machine vibration is a complex issue because there are so different factors causing vibration particularly when rotating components are present. Rolls and shafts are never perfect in shape and may be unbalanced statically or dynamically. Also, misalignments are possible in the assembly phase. Rolls may contain variations in hardness, and in process of time they may run out of shape. Gears, clutches, brakes and bearings may create chatter or speed variations to rolls. Drives themselves can also create small torque variations. External shocks or broadband disturbances from the entire mill environment can also travel to the nip over the foundation blocks and bearings. All these problems can generate vibration and in worst cases also nip resonance. Machine vibration has harmful effects to the quality of production and lifetime of components. According to

Roisum (1998), too much vibration is leading to waste production due to poor product quality or web breaks. Sometimes these problems have to be avoided by using lower production line speeds, which also leads to production losses. Rolls or other components may break or wear up due to vibration, which results to increased maintenance time and costs. Also, vibration can create safety problems or factory personnel discomfort in form of noise.

In order to attenuate vibration, feedback information is usually needed from the system. This is usually implemented with vibration measurements. Measured vibration parameter can be acceleration, velocity or displacement. In measurements of rotating machinery acceleration is usually measured. If needed, the acceleration signal can be converted into velocity- or displacement domains through integration procedures. The vibration sensor should be placed as close as possible to the vibration source. In most cases, the sensor is attached to bearing housing. However, generally vibration amplitude is strongest in the nip area in the middle of the roll. The sensor should be oriented in the direction of strongest vibration amplitude, which in nip contact is the direction of the nip load (Roisum 1998).

Efforts for attenuating machine vibration include several approaches from vibration predictive design to adding accessories and actuators to reduce vibrations of running units. In design phase, the system can be modeled and simulated in order to avoid vibration. This leads to optimization of roll cover and roll shell in the parameter space. Also roll frame can be designed to reduce vibrations, and foundation properties for machinery can also be designed same way. Accessories and actuators include different kind of dampers. Passive dampers typically change vibration energy to heat through different kinds of friction phenomena, which is the case when utilising fluid viscosity or mechanical sliding. Classical literature (den Hartog 1947) already knows passive mass dampers (Frahm damper), which have been developed recently towards tunable semi-active dampers by adjusting the stiffness (Karhunen 2005) or by tuning the control gains to adapt the damper to the observed frequency band (Virtanen 2006). Active dampers including actuators are generally designed to counteract vibration, but involve a risk to pump more energy to the system, if the monitoring information is not processed correctly or the actuator performance is not suitable for this application.

2.5 Delay vibration in rolling contact

In paper making generally known phenomenon, barring, which is a barrel-like deformation shape appearing at the roll surface, causes regular periodical variations in paper quality. Geometrically this kind of marking appears in a periodical formation of parallel cross-directional stripes at the whole width of the web. Measured variation can be in weight, thickness and gloss. Simultaneous observations of barring shape in roll cover and in paper web in link these unwanted anomalies strongly together. Surprising is that barring effect has been observed earlier at purely metallic roll surfaces and later on hard-soft rolling pairs with rubber, polyurethane, composite surfaces at the softer side. This has been explained by different damage mechanisms of the roll surfaces: in metallic contacts a wear mechanism is present while in softer nonmetallic surfaces it is more question of plastic deformations or on delayed recovery response of polymeric cover materials.

If temporary roll shaping in sense of barring exists at today so popular polymer covers, self-excited vibration can occur and it can lead to delay resonance if particular speed matching conditions are full-filled. There are a lot of process and design parameters, which influence on the barring behavior like line load, machine speed, roll temperature, and the fixed parameters related to roll, actuator and frame design. There can be also problems with roll grinding or with other finishing treatments as well as paper properties can have some additional effect. This phenomenon is so parameter sensitive that even small changes in these parameters can start or eliminate barring (Shelley 1997). Changes can have an influence on the barring frequency or amplitude. The dependence of the parameters from each other makes the situation even more difficult. Because each paper making process is unique, there is no unambiguous extract mechanism for the delay resonance. In reality, resonance speeds should be avoided, but running near a resonance is not restricted. The amplitude of vibration may perhaps increase as much as 10 times at critical speed than at speed only 10 % away from the critical speed (Roisum 1998).

The vibration behavior of the soft nip contact is nonlinear. While the roll tubes, machine frame, oil lubricated rolling bearings and the hydraulic loading circuit behave in small amplitude vibration linearly, the polymer cover under rolling contact exhibits two types of nonlinearities. The first one is the geometrical effect of the varying contact area, which is present when two convex bodies are in non-conformal contact. This leads to nip stiffness, which is increasing with the load. Such stiffening type nonlinearity makes the vibration frequency to depend on the vibration amplitude. When amplitude levels showing this phenomenon correspond to extremely high nip load variations, this nonlinearity has more academic interest. The conclusion is that barring vibrations represent dominantly small amplitude vibrations whether it marks the web or not. The other one is the delay-type nonlinearity related to the recovery history of the cover penetration. This nonlinear effect, which is the main research subject in this thesis, makes the response of the roll press to depend on the earlier motion history.

The natural frequency of nip contact is primarily design-dependent. However, process parameters have an effect on the natural frequency also. Higher line load increases nip stiffness and natural frequency. Higher temperature softens the soft roll cover, which lowers the stiffness and increases the damping of the cover and decreases natural frequency. This means that the natural frequency of nip vibration is a function of line load q and cover temperature T_c

$$f_{nat} = f_{nat}(q, T_c) \tag{1}$$

When the higher line load and higher speed are creating more damping power in the polymer cover, the cover temperature may increase bringing a new dependence

$$T_c = T_c(q, f_{rot}) \tag{2}$$

This phenomenon can be only partly compensated by the heating/cooling circuit of the roll, because the temperature gradient over the different material layers of the roll wall is complicated and follows very slowly the fluid temperature.

In delay resonance, the rolls vibrate in opposite direction against each other, when the rotation frequency f_{rot} of the roll and the natural frequency of the contact vibration f_{nat} are matching by rule

$$n \cdot f_{rot} = f_{nat} \tag{3}$$

in which n is the integer number of waves at the roll surface. This rule actually is the definition of the set of resonance rotation frequencies given by

$$f_{rot} = \frac{f_{nat}}{n} \tag{4}$$

Two important facts can be noticed. The first is that the delay-resonance speeds are forming a discrete spectrum in the rotation frequency domain. The second is that the barring frequency in delay resonance is always the natural frequency of the nip contact

$$f_{barr} = f_{nat} \tag{5}$$

If process parameters during delay resonance state are changed, the wave formation takes time to reform to the new running situation. Before reforming, the natural frequency and the frequency at which the waves are driven through the nip are different. When the new running state has stabilized the delay resonance may wake up or not, depending on whether the relationship (3) or (4) hold or not in the new process state.

This situation is complicated, because natural frequency and barring frequency can be changed independently. Barring frequency can be varied, if there is a need to adjust the line speed of the production. Natural frequency will also travel, if line load or temperature is changed. Essential is that in the beginning of the new situation condition (3) does not anymore hold leading to interference of the vibrations at natural frequency and barring frequency. If these frequencies are close enough to each other, a strong beating in the vibration is generated, but will slowly die out because of mismatch of rule (3). When this phenomenon is over, the remaining vibration represents nip oscillation at natural frequency.

Such beating is a special case of vibration, where the frequencies of two component vibrations are nearly equal to each other (Hartog 1947). This has been detected in mills and during laboratory experiments artificially produced with the pilot roll press. The dominating feature is the pulsation of the interference response at the beating frequency. When the component vibrations are in same phase they are gaining each other while in opposite phase they compensate each other. Mathematically the beating vibration of two component vibrations with different amplitudes and different frequencies (in this case barring frequency and natural frequency) can be presented in combined form (Flügge 1962) by

$$\begin{aligned} A(t)_{beat} &= a_1 \sin(2\pi f_{barr} t) + a_2 \sin(2\pi f_{nat} t) \\ &= [a_1^2 + a_2^2 + 2a_1 a_2 \cos(2\pi(f_{barr} - f_{nat})t)]^{½} \sin(2\pi f_{nat} t + \psi) \end{aligned} \tag{6}$$

with

$$\tan\psi = \frac{a_2 \sin(2\pi(f_{barr} - f_{nat})t)}{a_1 + a_2 \cos(2\pi(f_{barr} - f_{nat})t)} \tag{7}$$

The amplitude of the combined vibration is oscillating with frequency

$$f_{beat} = |f_{barr} - f_{nat}| \tag{8}$$

Depending on the situation, which one of the component frequencies is higher, the barring frequency can be estimated from the monitored beating frequency by rule

$$f_{barr} = f_{nat} \pm f_{beat} \tag{9}$$

This rule needs additional information to be used. One has to know, whether the running frequency or natural frequency has been changed to set the sign correctly.

In the special case, when the component amplitudes are equal

$$a = a_1 = a_2 \tag{10}$$

the combined vibration gets a simple form

$$A(t) = 2a\cos(2\pi \frac{f_{barr} - f_{nat}}{2} t)\sin(2\pi \frac{f_{barr} + f_{nat}}{2} t) \qquad (11)$$

In nip oscillations this situation is possible only at short moments, because the amplitudes of the component vibrations vary independently.

3. System model of the hydraulic roll press

3.1 Introduction

Roll systems represent multi-technological multi-component machinery, which have complex response behaviour. The scientific tradition of analyzing such systems requires the derivation of mathematical models, in which the essential interaction mechanisms are included. Such models can be used for three main reasons:

1. To show interaction mechanisms between different parts of the system.
2. To explain physical phenomena observed.
3. To predict quantitatively system behavior by means of numerical computation.

A system model has been built to describe a complete rolling contact system, which can be driven as a feedback controlled system like any industrial roll press system. This model consists of two interconnected sub-models to model correspondingly

- roll vibrations in normal direction of the contact surface
- roll motions in angular direction under the speed control

3.2 Roll motion in normal direction

The developed models are based on the geometry and mechanical properties of the press unit in TUT, but by changing input parameters similar system with different dimensions can be analyzed as well. The sub-model of normal motion consists of two degree of freedom vibrating system that also contains the delay phenomenon which is caused by the soft nip contact between the rolls, of which the upper one is covered by a highly viscous polymer layer. This is the simplest still realistic way to model rolls with such cover.

According to investigators (for instance Kustermann 2000), the rolls in nip contact exhibit motion related to beam bending, shell flattening, bearing elasticity, and cover compression. As the three first ones represent effectively a system consisting of three springs in series, they can be reduced in one spring constant. Furthermore, the value of this reduced spring constant can be chosen so that the combined stiffness effects of the structural elasticity and the nip stiffness correspond to the measured eigenfrequency (about *100* Hz) of the entire roll system in nip contact.

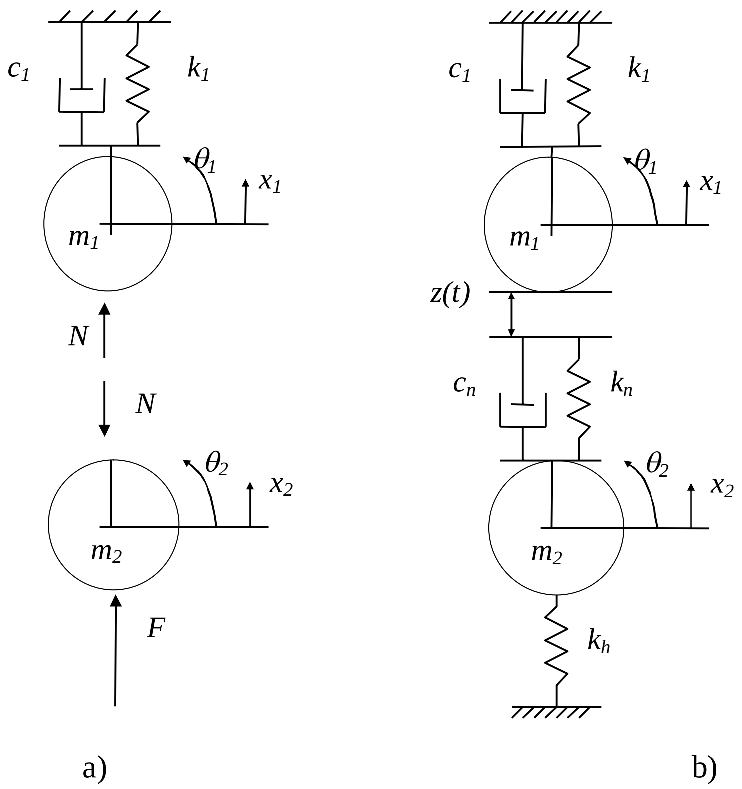


Figure 13. Loading situation (a) and vibration model (b) of the roll press.

By measuring the vertical positions of the roll centers with x_1 (upper covered soft roll) and x_2 (lower metallic hard roll), the dynamic equations of motion of the roll system in Figure 13a get form

$$m_1\ddot{x}_1 + c_1\dot{x}_1 + k_1 x_1 = N(\varepsilon, \dot{\varepsilon}) \tag{12}$$

$$m_2\ddot{x}_2 = -N(\varepsilon, \dot{\varepsilon}) + F \tag{13}$$

where N is the dynamic contact force between the rolls and F is the actuator load

$$F = r(p_p A_p - p_m A_m) \tag{14}$$

to make over a kinematic ratio r the required average line load by using the pressure p_p in the positive (pushing) chamber actively while the pressure p_m in the negative (pulling) chamber is a static counter pressure. The active pressure is governed by first order differential equation

$$\dot{p}_p = B \frac{uc_f \sqrt{p_s - p_p} - rA_p \dot{x}_2}{A_p (z_p + rx_2)} \tag{15}$$

in which B stands for the bulk modulus of the fluid, c_f is the flow coefficient of the valve. The control input u governing the opening of the proportional directional control valve is regulated by simple control law

$$u = K_P (F_d - F) \tag{16}$$

This is a rather unusual way to make force servo by using a flow control valve instead of pressure control valve, but has shown excellent performance.

The dynamic force N is a nonlinear function of cover penetration and penetration speed, which depend on the relative roll position and thickness variation $z(t)$ of cover or paper entering the nip

$$\varepsilon = x_2 - x_1 + z \tag{17}$$

In order to describe correctly the resisting force of the cover under normal penetration, one has to include the nonlinear effect of the changing contact area and the complex stress response of the cover polymer to the compressive strain. A precise modeling of this non-conformal contact problem leads to complicated numerical analysis (Tervonen 1997), which is difficult to be included in the computer simulation of a long-lasting dynamic process. To overcome this difficulty, a simple analytical model based on elastic foundation theory (Johnson 1985) has been developed (Keskinen 2002). Supposing that stress-strain relationship of the cover polymer can be described by a visco-elastic Kelvin-Voigt model with E and η representing the elastic and viscous constants of the polymer (Ward & Hadley 1993), one can link the compressive pressure p and the cover displacement δ together by equation

$$p(x) = E \frac{\delta(x)}{h} + \eta \frac{\dot{\delta}(x)}{h} \tag{18}$$

This relation holds on the contact zone on a very short time period, which is actually the relaxation phase. Outside of the contact zone the cover compression undergoes much longer free recovery phase, which is governed by initial value problem

$$\eta\dot{\delta} + E\delta = 0 \tag{19}$$

$$\delta(0) = \delta_o \tag{20}$$

The solution path is an exponential free recovery curve

$$\delta(t) = \delta_o e^{-\frac{E}{\eta}t} \tag{21}$$

In a non-conformal contact situation where a hard cylindrical body (lower roll) is penetrating into a softer cylindrical body (upper roll), the resulting nip force can be integrated over the contact zone bringing an exponential force-penetration law

$$N = a\varepsilon^b + c\sqrt{\varepsilon}\dot{\varepsilon} \tag{22}$$

in which the contact parameters

$$a = \frac{4}{3}\frac{E\ell}{h}\frac{1}{\sqrt{\beta}} \tag{23}$$

$$b = \frac{3}{2} \tag{24}$$

$$c = 2\frac{\eta\ell}{h}\frac{1}{\sqrt{\beta}} \tag{25}$$

$$\beta = ½(R_1^{-1} + R_2^{-1}) \tag{26}$$

depend on the cover thickness h length of the contact line ℓ and on the radii R_1, R_2 of the contacting rolls.

Due to the recovery behavior of the polymer cover, the contact model has to be updated to include the effect of the incomplete recovery period during the contact free zone

$$N = a(\varepsilon - \gamma\varepsilon_T)^b + c\sqrt{(\varepsilon - \gamma\varepsilon_T)}\dot{\varepsilon} \tag{27}$$

where $\varepsilon_T = \varepsilon(t - T)$ is the penetration one revolution earlier and T is the time of one roll revolution. Because the nip force is depending on the current and on the previous penetration, this actually leads to a delay-differential equation problem, in which the delayed response history brings the memory effect to the current nip loading.

This effect is controlled by factor

$$\gamma = e^{-\frac{T}{\tau_r}} \tag{28}$$

in which the time constant of polymer recovery is given by

$$\tau_r = \frac{\eta}{E} \tag{29}$$

The shape excitation can represent a time-dependent thickness profile of the paper web

$$z = z(t) \tag{30}$$

which typically has a connection to the web speed. This represents for the roll an external excitation. A typical example is a sinusoidal profile of a paper web traveling with speed v and carrying a wave with height Z and length λ

$$z = Z \sin \Omega t \tag{31}$$

where the angular frequency of the excitation is

$$\Omega = \frac{2\pi v}{\lambda} \tag{32}$$

The external excitation speed is then for the harmonic variation

$$\dot{z} = \Omega Z \cos \Omega t \tag{33}$$

In contrast to that, a shape profile of the roll cover is an internal excitation. If the roll profile around the roll perimeter is given in form

$$z = z(\psi) \tag{34}$$

the angular distance ψ from the roll-fixed reference line to the lowest point of the roll has to be updated during the motion by rule

$$\psi(t) = \frac{3}{2}\pi - \theta_1(t) \tag{35}$$

If the profile is representing *i*th harmonic component

$$z = Z \sin i\psi(t) \tag{36}$$

the speed gets expression

$$\dot{z} = -i\dot{\theta}_1 Z \cos i\psi(t) \tag{37}$$

When the primary interest is in monitoring the normal oscillations, a useful approach is to split the nonlinear quasistatic state from the small-amplitude and linear oscillatory response. Formally this leads to the splitting of the roll displacements and active pressure to static (^) and dynamic (δ) parts

$$x_1 = \hat{x}_1 + \delta x_1 \tag{38}$$

$$x_2 = \hat{x}_2 + \delta x_2 \tag{39}$$

$$p_p = \hat{p}_p + \delta p_p \tag{40}$$

Substitution of these quantities representing the static state and the dynamic response to the original general nonlinear equations (12)-(13) splits the whole problem to time-independent static equations

$$k_1\hat{x}_1 = a[\hat{x}_2 - \hat{x}_1 - \gamma(\hat{x}_2 - \hat{x}_1)_T]^b \tag{41}$$

$$a[\hat{x}_2 - \hat{x}_1 - \gamma(\hat{x}_2 - \hat{x}_1)_T]^b = F_d \tag{42}$$

$$F_d = r(\hat{p}_p A_p - p_m A_m) \tag{43}$$

and to the time-dependent dynamic equations of the spring –mass system in Figure 13b

$$m_1\delta\ddot{x}_1 + (c_1 + c_n)\delta\dot{x}_1 - c_n\delta\dot{x}_2 + (k_1 + k_n)\delta x_1 = \gamma k_n(\delta x_1 - \delta x_2)_T - \gamma k_n z_T + c_n\dot{z} + k_n z \tag{44}$$

$$m_2\delta\ddot{x}_2 - c_n\delta\dot{x}_1 + c_n\delta\dot{x}_2 - k_n\delta x_1 + (k_n + k_h)\delta x_2 = \gamma k_n(\delta x_2 - \delta x_1)_T + \gamma k_n z_T - c_n\dot{z} - k_n z \tag{45}$$

where the hydraulic spring constant of the loading circuit has expression

$$k_h = B\frac{r^2 A_p}{z_p + r\hat{x}_2} \tag{46}$$

and the linearized nip stiffness and nip damping have formulas

$$k_n = b\frac{F_d}{\hat{\varepsilon}} \tag{47}$$

$$c_n = c\sqrt{\hat{\varepsilon}} \tag{48}$$

By knowing that the current and delayed static states must be equal

$$\hat{\varepsilon} = \hat{x}_2 - \hat{x}_1 = (\hat{x}_2 - \hat{x}_1)_T \tag{49}$$

the static system can be solved

$$\hat{x}_1 = \frac{F_d}{k_1} \tag{50}$$

$$\hat{x}_2 = \frac{\left(\frac{F_d}{a}\right)^{1/b}}{1-\gamma} + \frac{F_d}{k_1} \tag{51}$$

$$\hat{p}_p = \frac{F_d}{rA_p} + p_m A_m \tag{52}$$

In case one prefers the use of the nonlinear dynamic equations (12)-(13), the static solution can be used for initial values in cases, where the simulation will be started at full speed in order to avoid long-lasting sliding to the final speed level.

3.3 Response to stochastic loading source

The use of the system model has been mainly addressed to the analysis of the following interactions:

Nip vibration response to different loading sources.
Development of the wave profile for different values of polymer parameters.
Memory model of the cover.
Interactions between natural vibration and barring vibration in sliding speed situation.

A typical feature of describing such complex interactions in time domain is the need of relatively long simulation, which in turn brings large amount of response information. To avoid difficulties in processing such less informative data, there are two possibilities to proceed.

The first is the change to frequency domain at system equation level. This leads to a description, which links the amplitude of the input to the amplitude of the output by means of complex frequency response functions. This analysis can be done in two different ways (Yuan 2002): one can fix the input frequency to the rotational speed (internal source) or slide them independently (external source). The second way to utilize frequency domain model is to calculate system response to stochastic nip loading (Keskinen 2005), which can be carried in case of cyclostationary (internal source) or general stationary (external source) loadings.

A representative input function when using such models is the white noise input. The physical meaning of this input is the random deviation of any of the cover parameters (E, η, h) or paper thickness z. If the source is the random variation of the elastic constant δE, the random load gets form

$$\delta N = \frac{N}{\hat{E}} \delta E \tag{53}$$

which model also includes the gaining effect of the nip load N to the stochastic input.

It has been discussed whether such stochastic analysis is missing some information of the cumulative time evolution of the wave profile on the roll cover. To be sure that such filtering of the output is not present, the time domain analysis has been done also under stochastic load by means of random input values obtained from different statistical distributions like from the uniform or the Poisson ones. Actually they are known solutions to describe roughness information of contacting surfaces (Johnson 1985).

The response of the rolls to such random input has been computed in long lasting simulation, when the roll rotation frequency has been sliding from *3 Hz* to *8 Hz* in time window of *200 s*. Material parameters of the roll cover of thickness *15 mm* have been identified by VTT Jyväskylä unit to be: $E = 40\ MPa$, $\eta = 0{,}3\ MPa\ s$, while the other system parameters are the ones of the roll press specified elsewhere in this thesis. The waterfall output of spectral densities of the roll displacements have been presented in waterfall plots in Figure 14, which is a popular way in vibration measurement technology to show spectral changes of system response under time-varying input.

The waterfall plots predict similar behavior of the rolling contact, which has been observed in the experimental studies. The existence of threshold speed, regular distribution of resonance speeds and the concentration of vibration energy to the natural frequency of the roll system can be clearly seen. When the magnitude of the response is highly controlled by the random input, the output of numerical computation has to be understood to be a conceptual one. The location of the threshold speed, above which the peaks are rising, is also sensitive to damping constants and viscoelastic material parameters of the roll cover. For these reasons the output plots have been normalized to range [*0,1*].

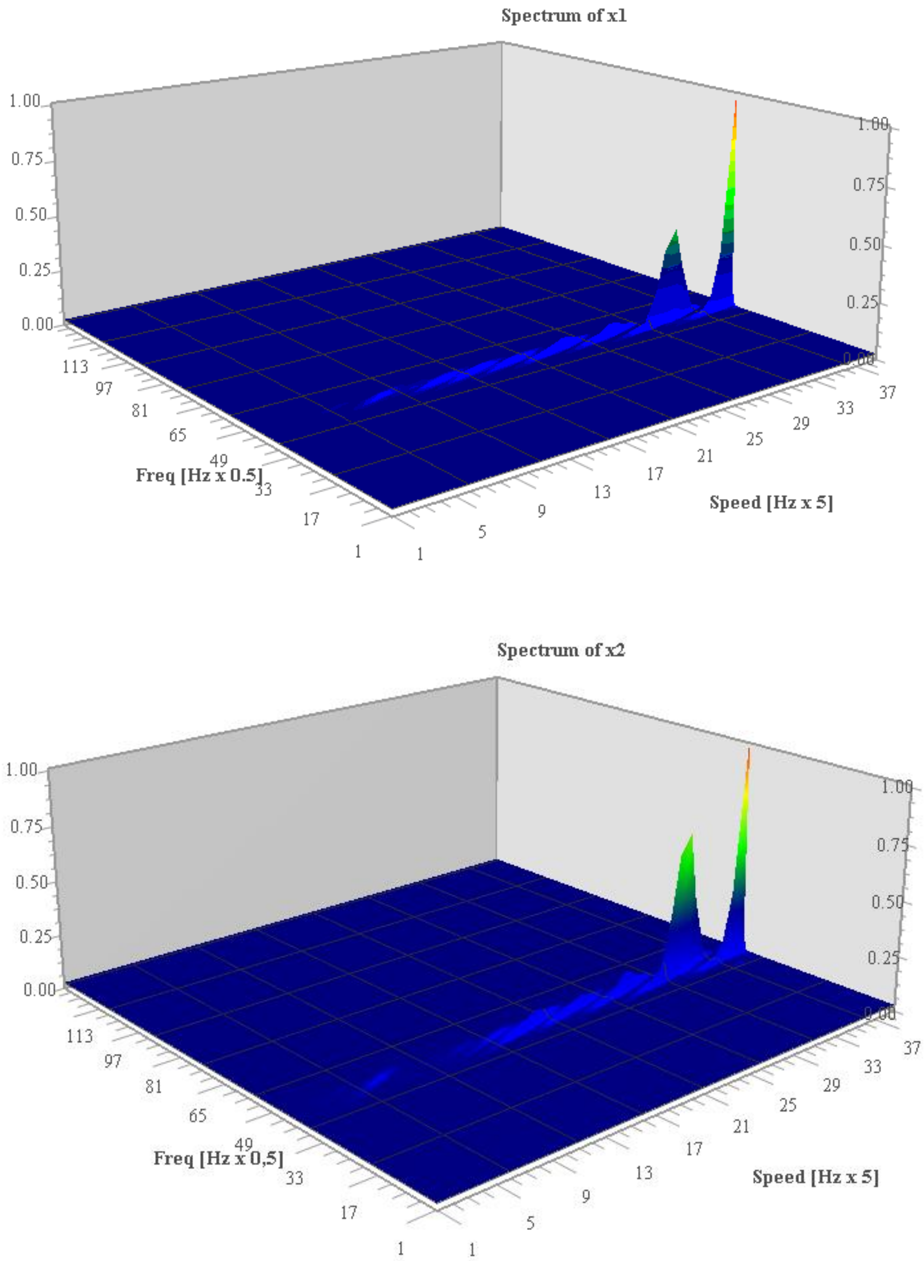


Figure 14. Normalized power spectrum densities of upper x_1 and lower x_2 roll displacements under random cover elasticity deviation load.

4. Vibration control methods

Vibration control in industrial applications usually means keeping vibration of machinery under accepted limit values. These limit values can be determined based on noise, quality of product or durability of the design or components. Also standards give some guideline values for the limits. Sections 4.1 and 4.2 review the literature about vibration control by dampers and actuators and by speed scheduling approach.

4.1 Vibration control by dampers and actuators

When the aim is to reduce vibration, the applications of vibration control techniques are commonly utilizing classical passive damping principles. These applications are based on fluid viscosity like car shock absorbers are working, on friction of two or more compressed plates in relative sliding (Osinski 1998) or on dynamic mass damper principle (den Hartog 1947). A tunable mass damper as shown in Figure 15, adapts to varying excitation frequencies. The basic rule of dimensioning the dynamic damper is

$$k / m = \omega^2 \tag{54}$$

where, according to Figure 15, k is the spring constant of the beam spring (3), m is the mass (4) and ω is the angular frequency of the vibrating system (2).

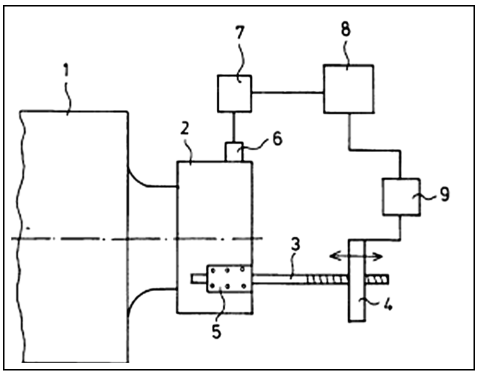
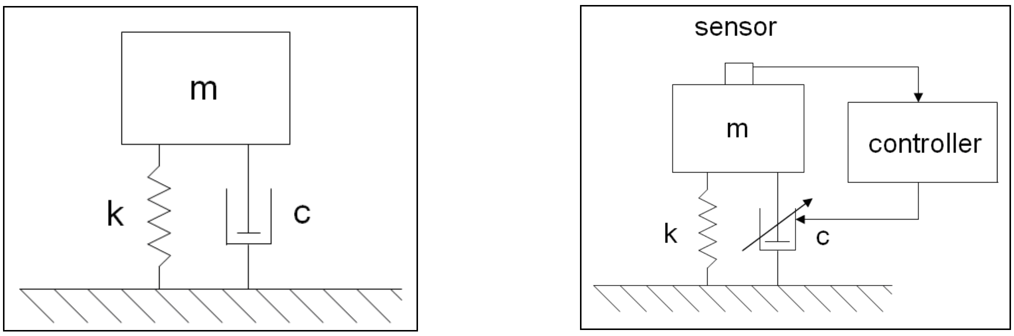


Figure 15. Tuned mass spring system. 1) roll, 2) bearing housing, 3) rod spring, 4) mass adjusted of its position, 5) fitting on the bearing housing, 6) vibration sensor and 7-9) control electronics (Karhunen 2005).

If the natural frequency of the primary system is varying, which is actually the case in roll press under different temperature and line load levels, see equation (1), the spring constant has to be adjusted by rule

$$k = 4\pi^2 f_{nat}^2(q,T)m \tag{55}$$

Such tuned and more sophisticated self tuning mass dampers have been successfully used in paper machinery applications. Improvement of dynamic damper can be done by adding a viscous damper parallel with the spring element, Figure 16a. This makes the tuning procedure less critical, when the damping property makes the effective running area wider in frequency domain.



a b

Figure 16. a) Principle of damping carried out with passive fluid damper. b) Principle of a semi-active vibration damping system.

Semi-active, Figure 16b, also known as adaptive-passive vibration control is based on tuning of damping or stiffness, but the coefficients can be altered during operation with relatively low energy. Therefore, the control can adapt to different kind of vibration overall states. Semi-active control is more versatile than passive control technique, and usually cheaper to implement than active control schemes. Measurements have shown that semi-active vibration damping can decrease resonance vibration of roll contact (Virtanen 2006, Salmenperä et al. 2005, Filipovic et al. 1997).

Passive and semi-active damping strategies become ineffective when the dynamics of the system or the frequencies of the disturbance vary with time. In those cases active vibration damping approach can be applied. Active damping system (Fuller 1996), which is also developed from the classical mass damper, Figure 17, contains an actuator that is controlled based on the feedback of the frequency content of the vibrating component.

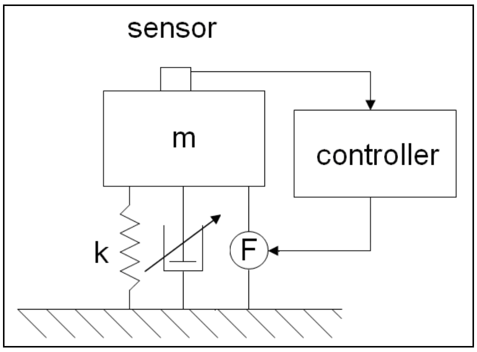


Figure 17. Principle of an active vibration damping system.

An arrangement, in which a secondary active damping actuator has been installed parallel with the primary loading actuator, is shown in Figure 18. The secondary actuators are mounted between roll shafts, which in turn have some distance from the support bearings. This kind of installation produces also bending reactions. By knowing that contact oscillations typically exist at frequencies beyond *100 Hz,* the secondary actuator has to be significantly faster than the primary hydraulic actuator in order to bring the required damping power to the system. In some applications a piezoceramic actuator has shown the required performance in this task (TEKES project Smartroll - Dynamics and control of direct drive rolls).

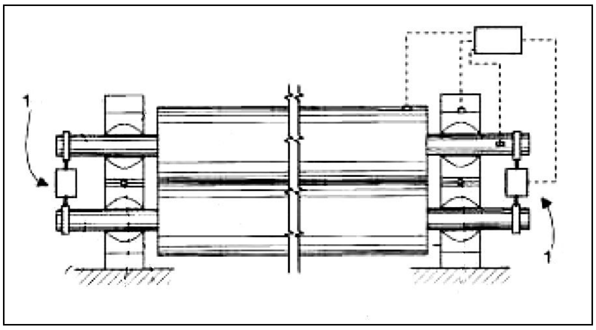


Figure 18. Active vibration damping system for nip vibration (Korolainen et al. 2010).

In the active vibration damping system the actuator feeds opposite directional damping force with the same frequency, compared to the excitation force, against to the vibrating component. Recent improvements in actuator, sensor and computing technologies have done it possible to find cost-effective solutions for active vibration control problems. Measurements have shown that active vibration damping can clearly decrease resonance vibration of roll contact (Virtanen 2006, TEKES project Smartroll - Dynamics and control of direct drive rolls). However, active vibration damping is often utilised only when passive methods are inadequate (Linjama et al.1996).

4.2 Vibration control by speed scheduling

When a machine is running, it can generate for many physical reasons periodic loadings, which force the whole system to oscillatory motion. This is avoidable but very common situation particularly in rotating machinery. The resonance state appears when the excitation force has the same frequency as the natural frequency of the system. Characteristic is that the excitation force can be very weak and still can generate very strong resonance amplitudes especially if the damping of the system is low. In the viewpoint of vibration control, speeding up or slowing down the running speed of a machine is the first aid to avoid the resonance. In some cases the resonance stage is so dominating that the raising of the speed is not possible because the resonance condition dissipates too much energy with respect to the performance of the actuator. Such situations have appeared for example in running up of large power generators.

Avoiding resonance vibration by changing running speed of a machine is not a new idea. One problem in the principle is that it is not easy to give a universal method because the vibration, especially resonance vibration behaviors of machines and designs and their running environments can be very different. The situation comes challenging if the speed change method has to optimize for example production speed or quality of the product.

Speed change control solutions for avoiding resonance vibration can be found from different industrial and also household areas. In Figure 19 speed change control method for avoiding intensive vibration areas or resonance of a paper winder is shown.

Characteristic feature of such machine is that the nip stiffness between the paper reel and the drums as well as the mass of the paper roll is changing when the radius of the paper roll is increasing. This means that for a constant web speed the rotation frequency and the natural frequency are changing during the process. It has been shown that there is a risk of resonance in two phases of the process, when the frequency curves are crossing. A particular speed adjusting program has been developed for fast passing of these crossings.

In this method, the running speed (S) of the winder is controlled based on the frequency of rotation of the roll. When the frequency of rotation of the roll approaches a range of oscillation (1,2), i.e. a range of frequency of rotation of the roll in which intensive oscillation occurs, the running speed (S) is lowered quickly. Then the speed of rotation of the roll is reduced to a level lower than the lower frequency of the range of oscillation (1,2). After this the running speed (S) of the winder is increased so that the frequency of rotation (F) of the roll remains invariable until the original running speed (S) of the winder is reached. (Virtanen 2006, Jorkama et al. 2001)

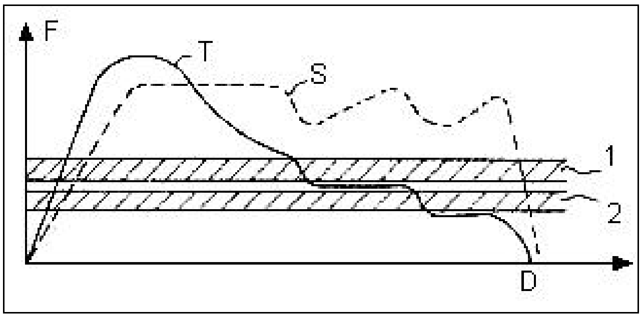


Figure 19. Resonance avoidance method in a winder. *D* is roll diameter, *S* is winder running speed, *T* is roll rotational frequency and F is frequency and 1 and 2 are areas of strong vibration. (Jorkama et al. 2001)

In grinding machines the speed or rotational frequency change control method has also been used (Altintas et al. 2000). Reference (Tokiwa et al. 2006) presents a grinding method for preventing chattering vibration during grinding process. In the method the rotational frequency of the grinding wheel is controlled by the vibration of the wheel and a speed change is performed to the grinding wheel if chattering vibration occurs. Reference (Yongyi et al. 2008) presents a speed change method in cam non-circle grinding. The idea in this is to get invariable grinding amount in every position of the cam. According to the reference, this method is more beneficial for avoiding impact and vibration caused by violent change of grinding force, thus obtaining higher processing precision and better surface quality of the work piece. There are also open loop vibration avoidance control methods in use in grinding machines (Xu et al. 2004). In those methods the rotational frequency of the work piece or grinding wheel is varied continuously in some small percentage. Reference (Hiroshi 2010) presents a speed change vibration avoidance method for a machine tool. In the method a stable rotation speed is acquired by finely changing rotation speed of the tool based on an expected stable rotation speed. Then some key figures will be calculated based on the rotational frequency and the geometry of the tool. Based on these key figures, a rotational frequency for the tool with lower vibration can be achieved.

In inverter driven fans speed change methods based on resonance vibration are also used (Onuma 1994). If a rotational frequency is reached during operation at which resonance occurs, the vibration value of the fan rises. Then the speed of the electric motor is changed for example with a predetermined amount of frequency. Reference (Lifson et al. 2006) presents a speed change resonance vibration avoidance method for electric motor and its accessories. In the method control of an electric motor is utilised to avoid operation in or near the resonance frequencies for the electric motor and its associated system components. The resonance frequencies are identified analytically at the design stage, or experimentally during operation for a component and electric motor. During start-up, shutdown or frequency adjustment, the control drives the speed through the resonance frequency zones more rapidly, and also avoids operation in or near those resonance frequencies during steady state operation. This method can be applied also if the electric motor is associated for example with fan or pump.

The speed or rotational frequency change method has been applied also in avoidance of torsional vibrations. Reference (Vyaas et al. 2009) describes a method where control is applied without using an adaptive speed signal and without attempting to actively damp the shaft resonance; instead the machine is controlled to avoid producing a torque component at the critical frequencies by estimating the torque disturbances from machine voltage or current measurements.

In household machines speed change control applications are also in use. Some methods get feedback signal from foaming of the washing soap (Ushijima et al. 2008) and some applications use adaptive vibration control speed change method (Son 2009).

As a conclusion one can state that speed change method for avoiding resonance vibration has been used in different industrial areas and there are several applications available. The question arising now is how speed scheduling can be practically done.

One can find two main principles in speed control. They are the closed loop control in which the feedback signal is taken from some process parameters, typically from vibration of the machine or its components. If high vibration, for example resonance vibration is detected the speed is changed. Anther principle is the open loop control in which the speed is changed continuously inside some limits, so that conditions for steady vibrations are disturbed to eliminate regular wave formation process on the contacting surfaces.

In the cases presented here the resonance vibration is typically resonance in one narrow frequency band, which means that when the band is passed, it is possible to raise the speed very freely. In the case of delay-resonance in rolling contact, the resonance situation is nearly always waiting beyond some threshold running speed, and in contrast to classical resonance cases, a discrete spectrum of delay-resonances has to be avoided by smart speed scheduling. This situation appearing in Figure 20 is the main difference and major challenge of this thesis problem.

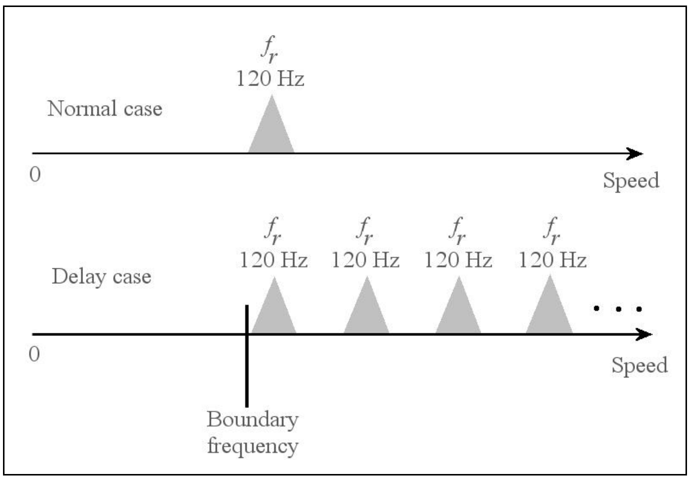


Figure 20. Comparison of classical resonance situation and delay-resonance situation.

4.3 Soft computing approach for processing incomplete information

Soft computing methods have been widely used in a number of areas like economics, cosmology, biology, health care and engineering sciences. All kinds of modern vehicles and household items utilise soft computing in control and information presentation. Soft computing represents a set of computational intelligence-based methodologies which are used to deal effectively with systems characterized by complex structures and possibly subjected to incomplete knowledge and ill-defined dynamics (Karray et al. 2004). Soft computing techniques allow taking advantage of the knowledge accumulated by the system either in a linguistic form or in numerical data form. Unlike hard computing schemes, which strive for exactness and full truth, soft computing techniques exploit the given tolerance of uncertainty for a particular problem.

The rolling contact system with polymer cover at the contact interface is an example of a system, whose response behavior can not be precisely predicted by purely computational methods. Uncertainties related to the sensitivity of cover response to temperature variations and parameters of material equation (stress-strain relationship) require the delay-resonance states to be experimentally detected. In addition to that, the measurement of the temperature gradient is very challenging. Instead of measuring or modeling the temperature gradient, a single point measurement of the temperature of the soft roll cover can be used as a predictive indicator for delay-resonance type vibrations. However, these resonance states for different surface temperatures have to be measured beforehand to collect information for the developed resonance control system.

Nip load also has effects on vibration behavior. Load influences on many parameters of rolls and the nip, like on nip load profile, on the size of the compression of soft roll cover and nip contact length. It is possible to approach the influences of the nip load on the vibration in a similar experimental way as on the influence of the temperature. Resonance measurements with different load parameters can be used as predictive indicators. The results can be combined, in order to create multi-parameter maps for resonance states for different load and temperature combinations. Such information is then stored and available for the control computer to make it possible to use rule-based speed scheduling algorithms in varying process conditions.

4.4 Speed variation methods for control of delay-resonance

The proposed speed-variation methods are based on experimentally determined vibration charts, where process parameters are rotational frequency, line load, roll temperature while the response parameter is vibration acceleration amplitude of the resonance vibration. Based on this study, the mentioned running parameters have strong influence on the resonance vibration. The flow diagram of the developed control methods is presented in Figure 21. The vibration avoidance methods use knowledge about the influence of cover temperature, nip load and rotational frequency on the resonance vibration of the nip contact. The knowledge is summed up experimentally in to vibration charts, which are utilised as supervised rules to define line speed where resonance vibration does not exist. Supervised in this context means ‘out of control loop’. Decisions for line speed changes are only suggested for operator.

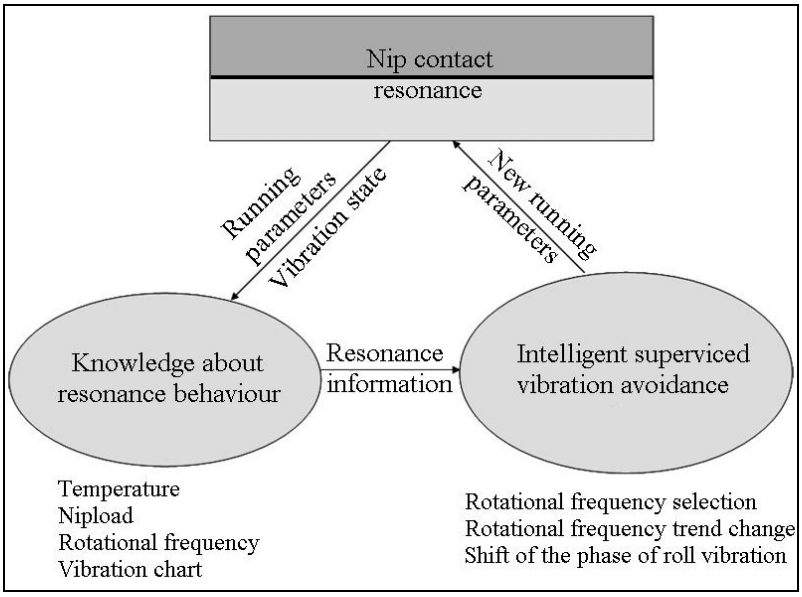


Figure 21. Flow diagram of the vibration control principle developed in this study.

The rules contain the comparison of barring frequency, based on the integer multiplies of the rotational frequency of the covered roll, to the nip contact natural frequency of the present state. This rule is essential in avoiding the nip resonance. Changing the line speed has an effect on the roll wave formation. On a definite barring-excited resonance vibration frequency, increasing line speed increases the wavelength in wave formation. On certain wavelength, one of its integer multiples matches the perimeter of the roll. In this case, the amplitude of the wave profile increases as long as affecting running parameters are maintained stable. If the wavelength of the barring profile is selected to stay in the middle of two successive integer fractions of the perimeter, the wave shape excitation entering the nip is in opposite phase with the natural oscillation. This action can be carried out by changing the rotational frequency of the roll and it will eliminate the delay-resonance.

The three nip resonance vibration avoidance control principles of rotating rolls in line contact developed in this study are based on selection of drive parameters in the following way:

The rotational frequency has to be selected to desynchronize deformation accumulation. If the roll rotational frequency is multiplied with current soft roll's barring wave number, the product should not be the natural frequency of the nip system. Instead, a wave number between two integers, close to current wave number of soft roll, should be selected.

Change the trend of rotational frequency. By this method, the whole integer of roll deformations is changed continuously and the running situation is changing continuously respectively. This can be done by speed sliding or speed oscillation principle.

Shift of the phase of roll vibration compared to roll deformation can be implemented through speed changes. The idea is to create a quick beating phenomenon described in chapter 2.2. When beating amplitude is in minimum, speed will be restored, leaving the roll vibration in opposite phase with roll deformation excitation. Of course, in reality speed must be restored before reaching minimum beating, because speed changes are transferred through a 'soft step' PI-controller, causing time lag to changes. Exact way how this is performed experimentally is presented in chapter 5.5 Phase change method.

When the first two methods are making trend changes to the roll rotation speed while the third method is making speed pulses in limited time span, the corresponding approaches can be classified to be slow and fast methods as well.

5. Experimental analysis of roll response

5.1 Introduction

Vibration response of roll system with delay-type nonlinearity differs significantly from the response of a classical standing or rotating structure. Such complex phenomenon has to be observed in a systematic way. This chapter separates the important factors controlling the response. Moreover, a platform of different running mode settings and monitoring tools has been built in order to run pre-programmed test procedures in accurate enough and repeatable way. Phenomena characteristic for a system in delayed rolling contact can then be identified from the responses.

5.2 Response behavior for different running parameter combinations

In the course of the study some component additions were done and also some modifications from which the largest are drive motor type changes and installation modifications and drive electronics modifications were carried out. In the first design the drive line of the rolls was composed of an electric motor, reduction gear transmission and universal shaft and couplings as shown in Figure 8 a. With this design it was studied the influence of different running parameters on the behavior of the vibration of the nip contact especially in nip resonance situation. The studied parameters were rotational frequency, nip load, temperature of the soft roll cover, torque of electric motors and so called “torque transient” which means a short impulse type high torque addition generated by electric drive motors. The “torque transient” can be called as the first stage of the phase shift method presented later in this thesis. Further torque share between hard master roll and soft slave roll were studied as well as force and position control of nip load. Negative property of this kind of power transmission line was the gear transmission. Gear transmission has rather large backlash, which is a nonlinearity in speed variation situations especially if the speed change is impulsive like in phase shift method. Gear transmissions also generate forced vibrations at teeth number fixed frequencies, which are transferred into the rolls disturbing vibration measurements. With the conventional AC motors, where gearboxes had been removed and only universal shafts were in use, some tests were carried out but the final development of the resonance control methods were carried out with the permanent magnet direct drive

AC motors mounted on the ends of the rolls. The direct drive motor installation is a compact and room saving design and presumably its applications in process industry, for example in paper machinery will become common. For these reasons, the resonance control methods were developed mainly for the direct drive case. Moreover, the transient speed changes in the phase-shift control work better, when the design has no backlash property.

Experience from the test pilot unit has shown following characteristics of the system:

- Vibration levels of test pilot unit tend to increase during time. This may be due to warming of the soft cover, which seems to increase amplitude.

- There exists speed windows where vibration stays low, but these windows can change during time. Sometimes even small changes in speed can diminish or excite resonance.

- Vibration level on a certain speed can be different depending of the direction of the last speed change.

- Vibration amplitude has generally high level on high nip load as well as high cover temperature.

- If constant speed is interrupted by a short period speed change, vibration can decrease significantly. This might be due to change in phase between roll deformation excitation and phase of vibration mode.

Figure 22 represents roll vibrations in run-up. On frequencies over 500 Hz can be seen strong noise, which is not nip-dependent and is caused by special motor setup. Filtering is performed to exclude the noise from resonance signal in the other measurements of this study.

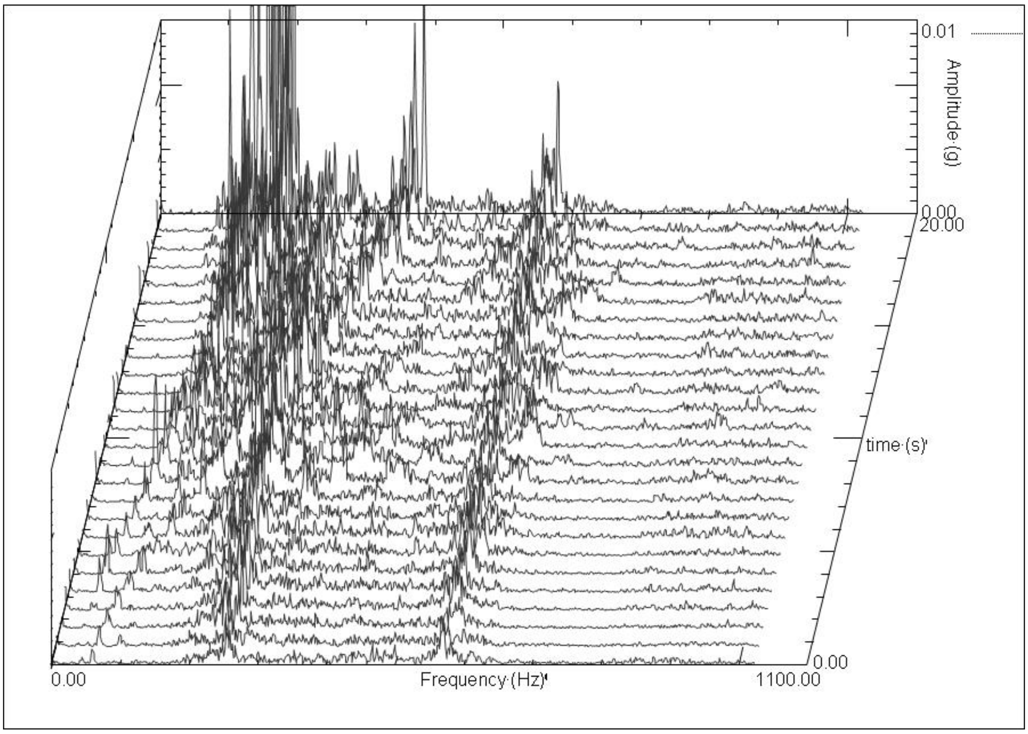


Figure 22. Typical lower roll vertical vibration result of the pilot roll press with direct drive in run-up from 100 m/min to 600 m/min.

5.3 Speed calibration

Operator interfaces in paper machines usually employ line speed as process parameter, for example in unit meter per minute, rather than roll rotational frequency. Main focus of the operator is in line speed, because it corresponds to production speed. Also, line speed control is essential with many lined up processing units, to keep paper web tension optimal. Operator is not interested in rotational frequencies or nip compressions though line speed is usually calculated from these parameters. It is very hard to consider the cover material flow and slippage in the nip or true effective radius for exact line speed calculation based on soft roll diameter. Effective radius depends on the nip load, line speed, cover material and its possible speeding up in the compressed nip.

Figure 23 describes the qualitative relationship of the effective diameter of the roll to the free diameter of the roll as a function of nip load with three values of Poisson ratio (υ) in the case of soft nip contact, assuming that the soft roll operates as master drive roll. Figure 23 is designed so that the line speed is assumed to have some fixed value in order to keep the tension of the paper web in right level. In the case of compressive roll cover ($\upsilon < 0.3$) the effective diameter of the roll is too small to produce the fixed line speed so the roll operates in "underdrive" state or in other words rotates too slowly. This running state creates an increment in web tension and may lead to break the web. In the case of less compressible cover material ($\upsilon > 0.3$), like polyurethane, the roll operates in "overdrive" state so the effective roll diameter is too large or in other words the roll rotates too fast. This running state may lead to slack of web tension.

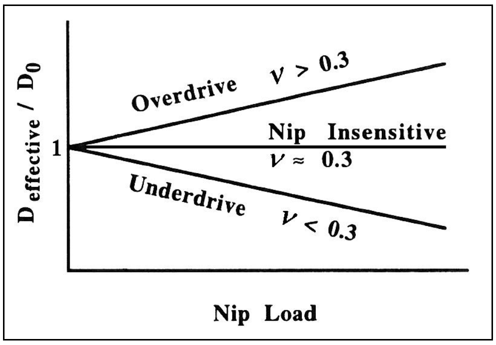


Figure 23. Relationship of the effective diameter of the roll to the free diameter of the roll as a function of nip load with three values of Poisson ratio (υ) in the case of soft nip contact, assuming stable paper web tension (Roisum 1998).

The situation described in the Figure 23 is problematic especially if the master drive roll is soft roll. In the pilot roll station used in this study the master drive roll is the hard roll when the line speed stays in principle very stable as a function of nip load. In the case of phase change control the nip's line speed is temporarily changed in order to counteract roll barring vibration which is excited by the soft roll wave profile. This vibration avoiding method needs to know a very accurate rotational frequency of the soft roll. Because the process parameter in paper machine normally is line speed, a correlation between line speed and soft roll rotational frequency in pilot roll press was clarified to be utilised in phase shift method.

Real line speed and machine set line speed were compared through measurements, to increase knowledge about possible errors. These errors might be due to small errors in roll diameters or due to the crowning of the soft roll and due to the penetration of the hard roll in to the soft roll cover making the rotation diameter smaller. It is also possible that slippage exists between the rolls, especially on lower nip loads. In this case, the speed controlled hard roll has a different line speed than the torque controlled soft roll by means of Figure 23. The excitation frequency caused by the roll wave profile is dependent on the soft roll speed. The line speed of the hard and soft roll is of course the same, when no slippage exists. When the wave profile locates at the surface of soft roll, it is natural to fix the line speed and rotation speed to the soft roll. Line speed of the soft roll under nip penetration was calculated from measured roll radius in the nip and rotational speed of the soft roll. Soft roll radius in the nip is the free radius of roll minus compression in the nip and was measured indirectly through lower roll position by compensating first the effect of roll bending. Because load has a strong influence on compression, it was varied between 5 … 20 kN/m during measurements. Temperature was 25 … 26 °C, corresponding to normal test roll operating conditions. The measurement results are shown in Figure 24.

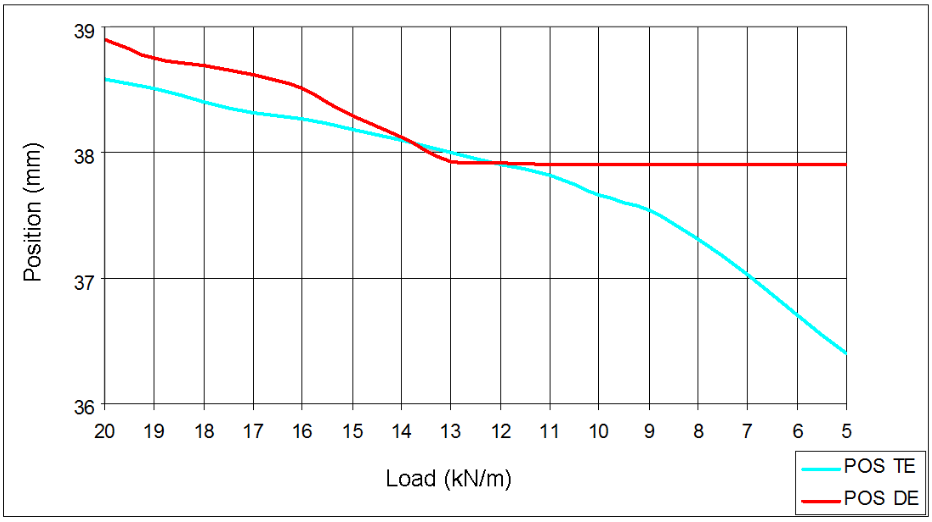


Figure 24. Nip load vs. position of the lower roll as shared to the tender end (TE) and roll drive end (DE).

Somewhat non-symmetrical behavior can be seen between roll end positions. With lower nip loads than the nominal 15 kN/m, the load decrease seems to have an effect to only one end of the roll. This might be due to the crowning, which makes the rolls to have irregular swinging in pitch-mode as well. Thus, lower loads than the nominal create non-uniform line load distribution in CD, caused by the barrel-shaped soft roll.

Roll end positions were averaged to form a curve of soft roll average displacement in the nip, see Figure 25.

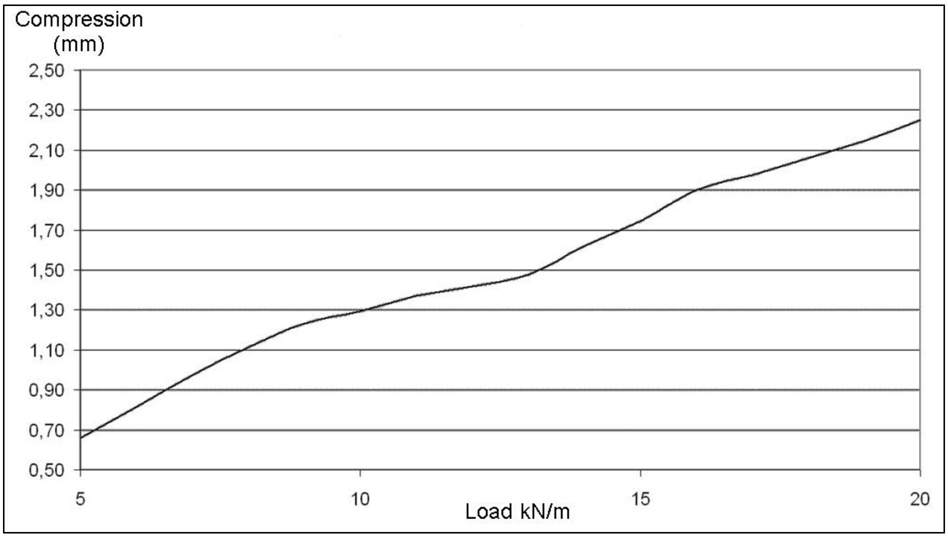


Figure 25. Nip load vs. average surface approach of the soft roll. Average is calculated from displacements in roll ends.

This problem of non-symmetrical behavior of the nip loading mechanism is not new in the paper production lines. The classical solution used in many roll press sections is to use a synchronizing shaft, which purely mechanically forces the moving roll to contact and penetrate against the nonmoving roll in exactly same orientation. Today this problem falls to category parallel manipulation, for which the servo technology brings a solution. When the rolls by nature are in relative pitch motion, the end reactions and thus the load sensing and actuator forces are not anymore independent. Such cross-coupling effect can be compensated by changing from the control of the end loadings separately to the control of the total load by keeping the load difference zero.

Exact line speed calculation is very difficult with soft rolls despite of exact measurements of effective roll diameter and roll rotational speed. Most soft roll cover materials are relatively incompressible and cover material has to speed up in the nip so that the flow rate of material is the same at the nip as elsewhere (Roisum 1998). This is the “overdrive” running state as described in Figure 23. The test roll station’s soft roll cover is polyurethane, having a Poisson ratio near of 0.5, which means that it is rather incompressible. However, this should not have effect on barring frequency since the same amount of deformation waves still reach the nip in the same time interval. The Table 2 indicates set line speed values and measured rotational frequency values and calculated compressed-cover surface speed of the soft roll, based on the measurement shown in Figure 24. In Figure 26 the calculated error between set line speed and calculated compressed-cover line speed of the soft roll in the case of nip load of 15 kN/m is shown.

The nearly incompressible polyurethane cover causes the roll to act bigger than its free radius and thus turn faster than expected in order to achieve the set line speed (Roisum 1998). One has to remember that in pilot roll station the line speed is determined by the hard roll. This study of the speed of the soft roll is utilised in phase shift control principle.

Table 2. Measurement results of machine set line speed and corresponding real rotational speed and frequency and calculated compressed nip line speed of the soft roll.

line load	15 kN/m	20 kN/m	15 kN/m	20 kN/m	15 kN/m	20 kN/m
Set line speed (m/min)	Real rot. speed (rpm)	Real rot. speed (rpm)	Real rot. frequency (Hz)	Real rot. frequency (Hz)	Calculated compressed line speed (m/min)	Calculated compressed line speed (m/min)
300.0	173.4	173.1	2.890	2.885	297.3	295.0
350.0	201.7	201.4	3.362	3.357	345.8	343.2
400.0	230.4	229.8	3.840	3.830	395.0	391.7
450.0	258.8	258.2	4.313	4.303	443.7	440.1
500.0	287.3	286.7	4.788	4.778	492.5	488.6
550.0	316.0	315.4	5.267	5.257	541.7	537.5
600.0	344.1	343.4	5.735	5.723	589.9	585.3

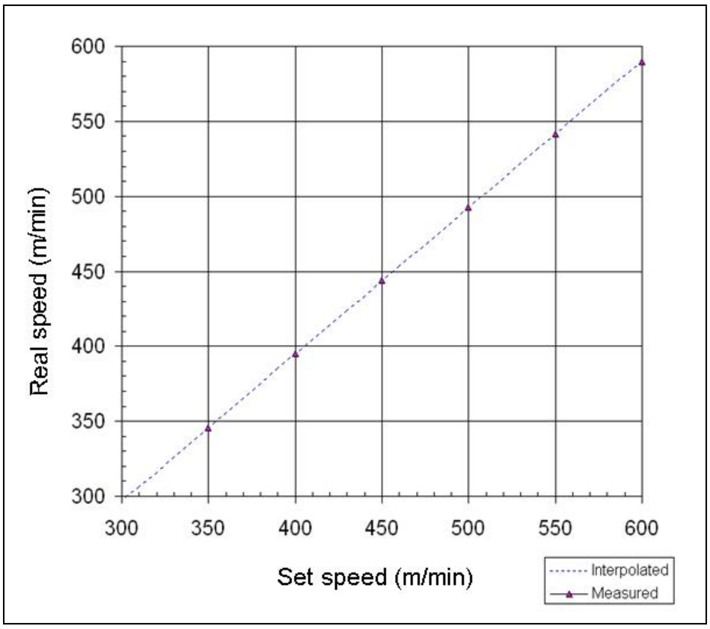


Figure 26. Machine set speed vs. real speed or compressed-cover line speed of the soft roll on load of 15 kN/m. Interpolated line fits data quite well.

On the basis of the study of set and real line speed values, an interpolation program was constructed for other loads and speeds by using commercial LabView software. The developed program interpolates results for other nip loads between two loads which were measured; 15 kN/m and 20 kN/m. Polynomial interpolation has been used for results between seven measured speeds. Block diagram and user interface for real roll speed calculation are presented in Figures 27 and 28. The parameters which can be varied are the line speed (Set line speed) and line load (Line load) and the output is the real line speed of the soft roll (Real line speed).

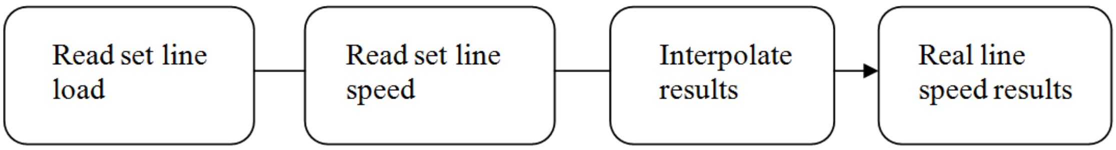


Figure 27. Block diagram of speed calibration.

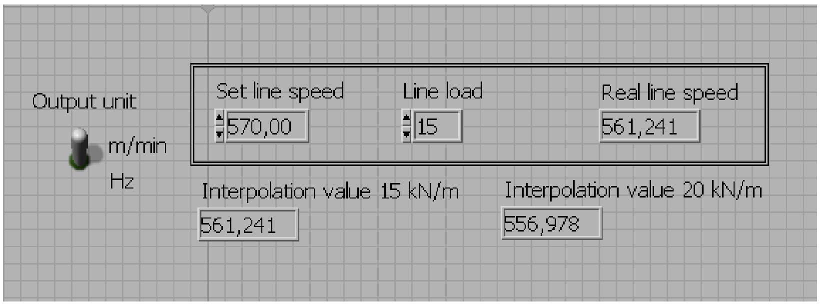


Figure 28. LabView user interface for speed calibration.

The real line speed calculation program is used within phase change control principle, in order to create half a wavelength offset between roll wave profile and vibration phase. Without calibration, values for the length of roll wave profile would be distorted and results of phase change operations less effective.

5.4 Vibration charts

In order to determine nip behavior and vibration dependencies from speed, load and temperature, measurements were performed with combinations of most essential process parameters. These parameters were selected because they are the most affecting to vibration (Roisum 1998). Speed and load were obtained from process control parameters. In order to guarantee similar preconditions for each measurement, an automated measurement program was used. Block diagram and user interface are presented in Figures 29 and 30.

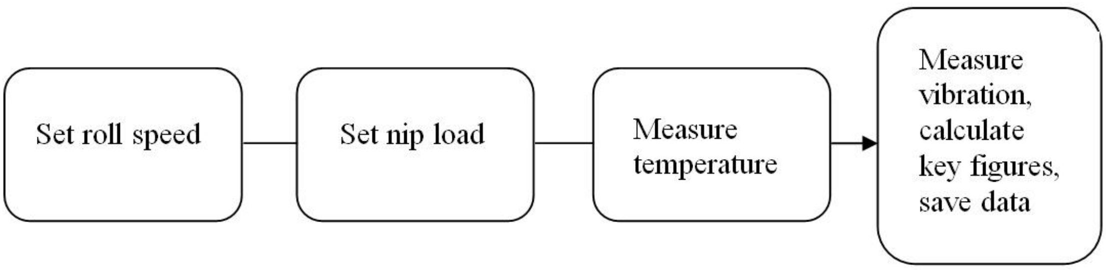


Figure 29. Block diagram of the model measurement program.

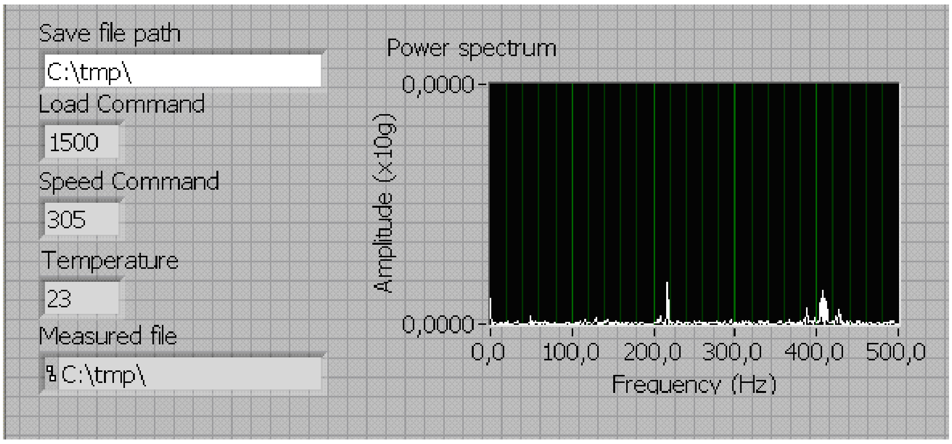


Figure 30. User interface of the model measurement program.

In Figure 31 the measurement procedure is shown. Measurement program contains a loop structure, where temperature of the soft roll cover is measured first. Then nip load is set to a valid value and after this the next speed without a measurement result is selected. This way all combinations of process parameters will finally have a vibration measurement result. When the speed and load have settled to selected values, the measurement of vibration can be started. The program saves from the measurement vibration acceleration RMS of the signal, the highest peak of acceleration amplitude in resonance frequency window and corresponding frequency of the peak. Resonance frequency window is defined as 100 – 150 Hz.

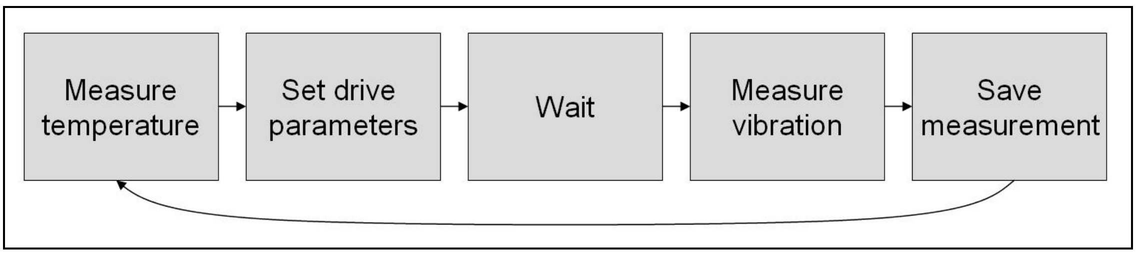


Figure 31. Measurement procedure of vibration charts.

The results form an experimental model of vibration behavior. Two vibration models were created based on a large number of measurements. First one was used to gather initial understanding of the vibration behavior in the nip, and it consists of measurements with load parameter varying between 15 … 20 kN/m and a speed step of

5 m/min running upward from 300 m/min to 600 m/min. Temperature varied between 24 … 27 ºC.

Second model includes only the nominal nip load 15 kN/m, but more accurate speed step of 1 m/min with running upward and downward. The measurements were performed this way to confirm that the response curve for non-linear system would appear (Chinn 1999). This means that upward- and downward measurement result are different within each other. Therefore the second model consists of both measurements running upward and downward. Example of resulting vibration chart is presented in Figure 32 and block diagram of vibration chart drawing program is presented in Figure 33. Totally 300 measurement results are collected into a chart describing nip vibration on each temperature and separately for upward and downward speeds. Amplitude has ‘peaks’ and ‘valleys’ and periodic vibration behavior through line speed domain is clearly visible.

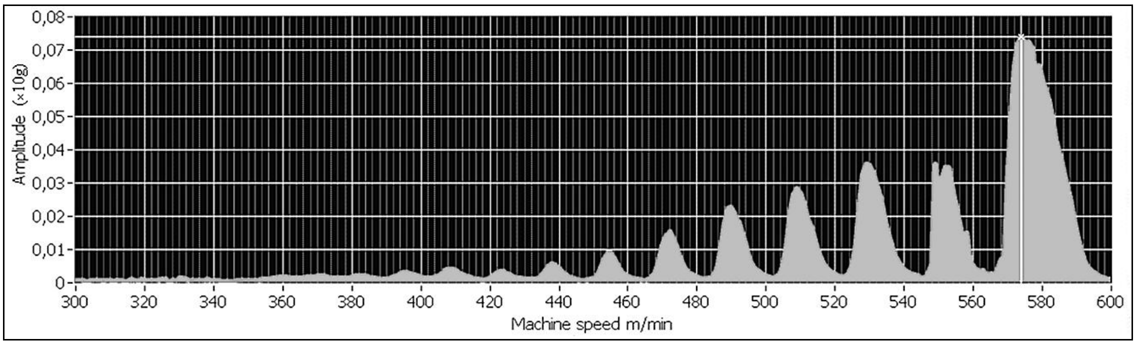


Figure 32. Example of a vibration chart of running downward measurements with load of 15 kN/m and temperature of 27 ºC.

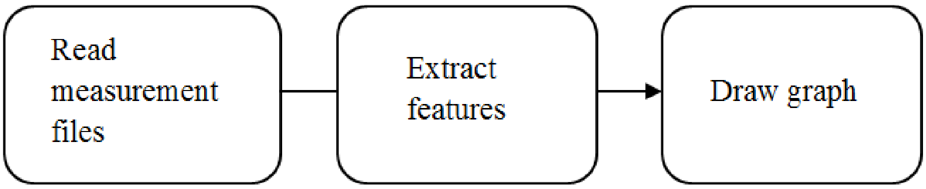


Figure 33. Block diagram of vibration chart drawing program.

Temperatures varied between 25 … 27 ºC in the measurements of the second model. Process parameters and their varying ranges are presented in Table 3 and Table 4. The temperature of soft roll cover affects strongly the nip’s vibration behavior. A temperature gradient inside the cover can vary, and surface temperature does not tell everything about cover properties. However, thermo elements inside the cover have a short lifetime, and therefore the temperature gradient could not be investigated in this thesis. Instead, a non-contacting surface temperature measurement was used. Lower limit of the temperature area was defined by the fact that temperature rose quickly over 24 ºC, and on lower temperatures the charts could not be measured completely. Upper limit was defined in the same way. Higher temperatures could not be maintained during measurements. The temperature was generated by inner friction of soft roll cover. It would also have been possible to warm the soft roll with backwater circulation. However, this would have turned the temperature gradient inside the cover, creating lowest temperatures on roll surface. This would have an effect on the measurements and their reliability (Chinn 1999).

Table 3. Process parameters and their varying ranges in 1^{st} model measurements.

Parameter	Varying range	Accuracy step	Number of elements
Line speed (m/min)	300 … 600	5	60
Load (kN/m)	15 … 20	1	6
Temperature (°C)	24 … 27	1	4
Running way	upward	upward	1
Total number of measurements			1440

Table 4. Process parameters and their varying ranges in 2^{nd} model measurements.

Parameter	Varying range	Accuracy step	Number of elements
Line speed (m/min)	300 … 600	1	300
Load (kN/m)	15	1	1
Temperature (°C)	25 … 27	1	3
Running way	upward/downward	upward/downward	2
Total number of measurements			1800

Nip load varied between 15 … 20 kN/m in the first model. Nip load 15 kN/m is a nominal nip load, which creates the most even load in nip cross direction (CD), see Figure 34. The second model consists only the nominal nip load 15 kN/m. The pressure profile is measured with nip film Fujifilm Prescale type Super low. Material consists of two plastic stripes coated with micro balls containing reactive chemicals.

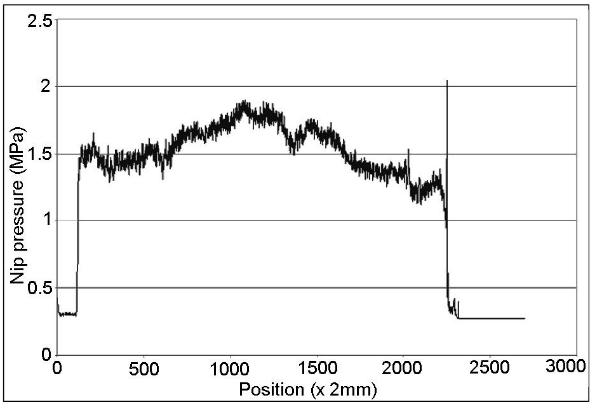


Figure 34. Nip pressure distribution in CD with nominal nip load 15 kN/m. Tender end on right side. Results from TEKES project Polyroll - Dynamics and operation monitoring of polymer covered rolls.

Line speed varied between 300 … 600 m/min during vibration chart measurements, corresponding roughly to 3 … 6 Hz. Speeds below 300 m/min were not used, because resonance doesn't evolve on such low speeds. On the other hand, the maximum speed of test machine is 600 m/min.

Vibration charts presented in Figures 35 and 36 have a correspondence with analytical model results presented in Figure 14. Figure 35 represents a vibration RMS result in a downward running measurement in temperature of 27 ºC. Figure 36 represents resonance frequency in the same conditions. It can be seen that highest amplitudes in vibration RMS are situated in frequency near 122 Hz and as speed changes, frequency will change as a multiple of rotational speed until it reaches its limits and changes its rotational multiple. The same phenomenon can be seen in analytical model results, see Figure 14. The resonance occurs only in a narrow frequency band, which on current load and temperature is approximately 121 – 125 Hz.

Speed 530 m/min in Figure 35 corresponds to estimated rotational frequency of 5.08 Hz, according to data in Table 2. Figure 36 shows the resonance frequency of 121.7 Hz on the same speed. From these values it can be calculated that the number of waves on roll surface is in this case 24, when resonance frequency is divided by soft roll rotational frequency. A small error between exact value is easily explained by accuracy of speed control or measurement results as well as possible small sliding between the rolls in the nip.

The vibration charts are quite different in upward and downward running measurements. The characteristics are clearer in downward measurements, but high amplitude peaks appear in upward measurements also, see Figures 35 and 37 respectively. Especially highest speed peaks are very close each other, like speeds of 526 m/min and 552 m/min.

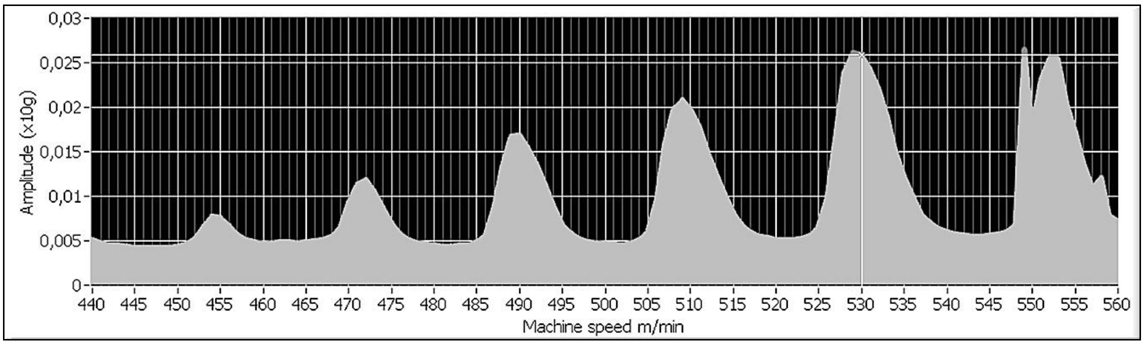


Figure 35. Vibration RMS result in a downward running measurement in temperature of 27 ºC, load 15 kN/m.

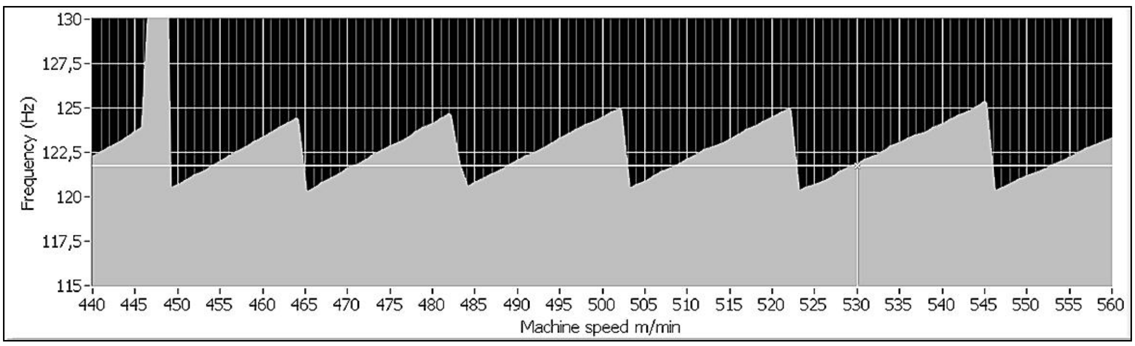


Figure 36. Resonance frequency result in a downward running measurement in temperature of 27 ºC, load 15 kN/m.

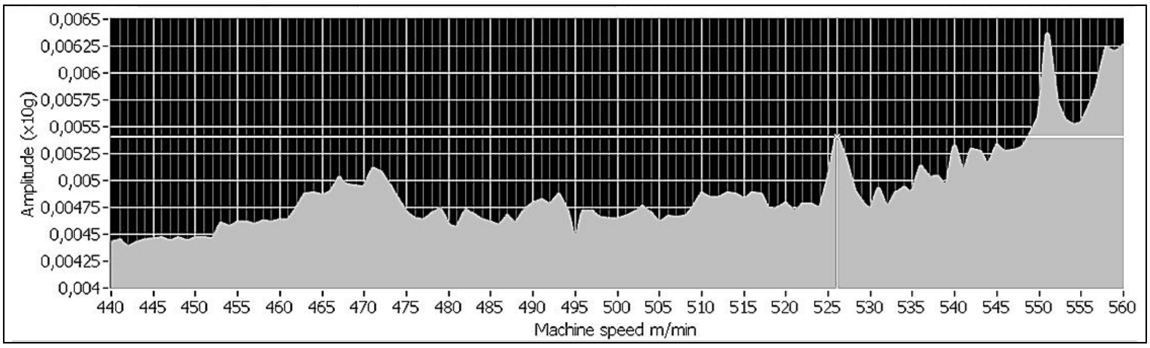


Figure 37. Vibration RMS result in an upward running measurement in temperature of 27 ºC, load 15 kN/m.

The vibration charts were measured with running upward- and downward separately in the second model. The shape of resonance areas, or ‘peaks’ and ‘valleys’ were clearer in running downward measurements. Measurements were verified with additional stable speed measurements to make sure the delay phenomenon did not corrugate the results in original measurements, where speed was slid upwards or downwards. Additional verification measurements were performed on certain speeds separately to make sure the vibration history of speed sliding does not have negative effect on results, see chapter 5.3 Stable speed method.

Vibration charts pointed out, that on higher nip loads natural vibration is dominating and delay resonance is lower in amplitude. This can be seen from dominating frequency on higher nip loads and higher line speeds. As can be seen in Figure 38, dominating frequency is not affected by nip speed on high line speeds. This also indicates that speed change methods might not be as effective on higher nip loads. However, the roll crowning in the test roll is designed for nip load of 15 kN/m, and on this load the delay resonance is dominating and thus method is applicable.

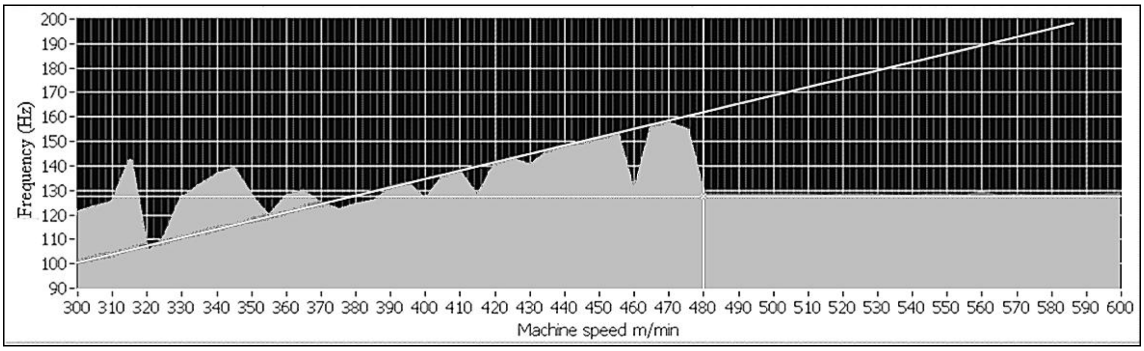


Figure 38. Frequency map of nip vibration shows static vibration frequency on higher nip speeds and loads, temperature 25 ^oC and load 20 kN/m.

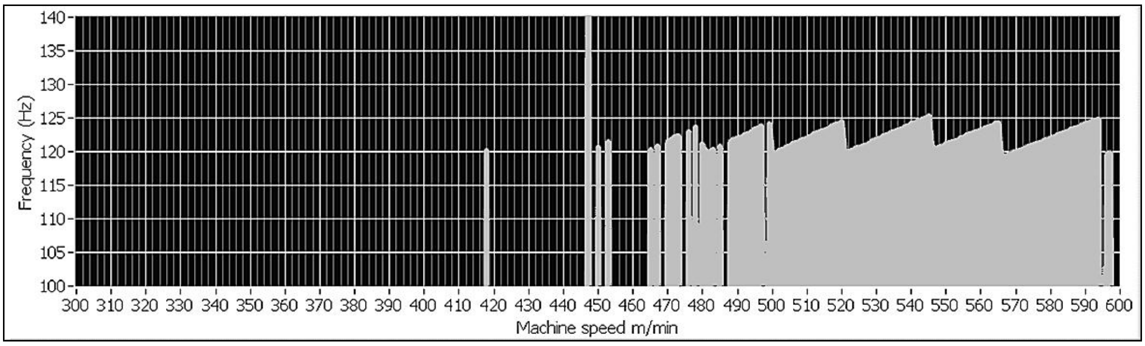


Figure 39. Frequency map of nip vibration shows variable vibration frequencies which are multiples of rotational frequencies.

From the charts it can be seen, that the test machine has two different kinds of vibration states. On lower line loads the vibration frequency is a multiple of rotational frequency, as can be seen in Figure 39. In these cases vibration amplitude usually stays lower than on higher line loads. Missing measurement results in Figure 39 on low speeds are a result of low line speed and low line load, which is not generating enough heat to complete measurements. In other words, temperature quickly dropped below 25 ºC, which was the minimum temperature in current vibration charts. On higher line speed and line load, the natural frequency of the roll nip contact dominates rotational frequency multiples so the frequency is the same on wide rotational speed band. The amplitudes are usually higher on this vibration mode.

5.5 Reliability of results

In the course of the study the author became convinced of that phenomenon delay resonance vibration is an exceptional phenomenon. Sometimes the resonance awakes very rapidly and sometimes not with the same process parameter settings or speed, nip load and soft roll surface temperature. This is because there are also other parameters affecting to the vibration behavior like oil temperature of support bearings of the roll, temperature of the joints of the links and very important is the temperature inside the soft cover.

Influence of temperature:

For the reasons mentioned it is very important to run the machine for such a long time that a stabilized running state is achieved. In the measurement procedure before every documented measurement the machine was running six hours and after that the measurements were carried out. Measurements in stable running state were repeated minimum three times or more times in the temperature window of ±0.5 ^{o}C of the soft cover. In order to create similar temperature gradient inside the cover between measurements, backwater heating of soft roll was not used during this study.

Influence of nip load:

In the design of the pilot roll press the nip load is measured from both ends of the roll with force sensors so the frictions of hydraulic cylinders or link joints do not affect on the nip load. Nip load is in close loop control and feedback comes from force sensors. Experimental results from pilot roll nip loading mechanism show excellent line load controlling properties (Kivinen, 2001).

Influence of roll rotational frequency:

The rotational frequency of the rolls is a very important process parameter. All speed change measurement results presented in this study are carried out with the direct drive installation. These permanent magnet AC motors are unique prototype motors and the electric drives were tuned by a specialist of the motor manufacturer.

Referring to the Table 2 “Measurement results of machine set line speed and corresponding real rotational speed and frequency and calculated compressed cover line speed of the soft roll” there are very small differences in rotational frequency of the soft roll between nip loads 15 kN/m and 20 kN/m. The results show that the rotational frequency of the soft roll is lower with higher nip load. With higher nip load somewhat higher torque from the motors are needed. A test were carried out to be sure that the rotational frequency of the hard master roll stays stable with both of the loads. In Figure 40 is a measurement result of the rotational frequencies of hard and soft roll with nip loads 15 kN/m and 20 kN/m. The results show that the rotational frequency of the hard roll stays stable in 9.138 seconds measured over 37 pulses that corresponds to 4.049 Hz, see Figure 40 a and of the soft roll lowers from 9.111 seconds to 9.130 second measured over 35 pulses that corresponds to from 3.841 Hz to 3.834 Hz, see Figure 40 b.

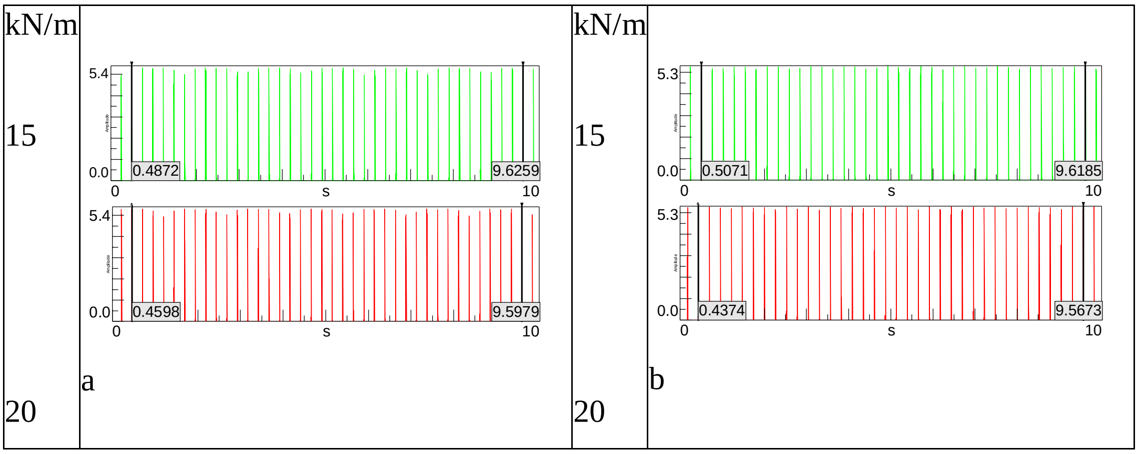


Figure 40. a) Pulse signal of 37 rotations of the hard roll with nip loads of 15 kN/m and 20 kN/m. b) Pulse signal of 35 rotations of the soft roll with nip load of 15 kN/m and 20 kN/m.

6. Application of speed variation methods for delay-resonance control

6.1 Introduction

As introduced in chapter 4, resonance control by speed variations is based on three different approaches and corresponding drive speed setting techniques.

Speed setting to fixed anti-resonance speeds:

This is based on experimentally defined map of resonance speeds, at which barring is initialized. The principle is just to set the active running speed between two successive known resonance speeds preferring the middle point as the most favor guess.

Speed scheduling for safe passing of the resonance speeds:

This can be done by sliding back and fort over larger speed intervals or by oscillating the speed near the local resonance so that in all cases the wave formation mechanism is disturbed.

Transient speed change for phase-shift control of the barring excitation:

This can be done near the resonance by a periodical set of rapid speed pulses, after which the speed remains the original one, but the phase of barring excitation has been changed opposite to the current vibration response.

This chapter describes in details how these speed variation principles have been implemented and realized in the pilot roll press at TUT.

6.2 Fixed speed method

Measurements on stable speeds were performed in order to verify usability of vibration charts. Soft roll cover temperature was maintained in 25 ºC during measurements, so the chart in Figure 41 is valid. Downwards running vibration charts were used, because the shape of resonance peaks is clearer than in running upwards measurements. Line load was the nominal 15 kN/m and line speeds were 520 m/min and 544 m/min representing low vibration areas and 526 m/min and 550 m/min representing nip resonance situations, so these speeds are located respectively in valleys and peaks in the vibration chart. Therefore it is presumable that speeds of 520 m/min and 544 m/min would result in a stable vibration on low level, while the other speeds would derive in resonance state.

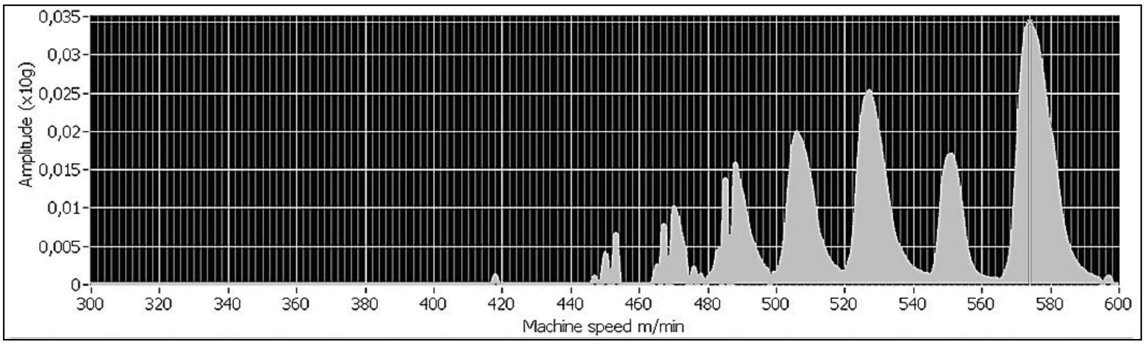


Figure 41. Vibration chart of resonance peaks and anti-resonance valleys running downward measurement with temperature of 25 ºC and line load of 15 kN/m and multiple factor 23.

The vibration chart verifying measurements were carried out during time period of over eight minutes or 500 seconds, to see if the vibration level stays stable or if it will rise. In Figure 42 the vibration behavior in resonance region and in Figure 43 the vibration behavior in anti-resonance region is shown.

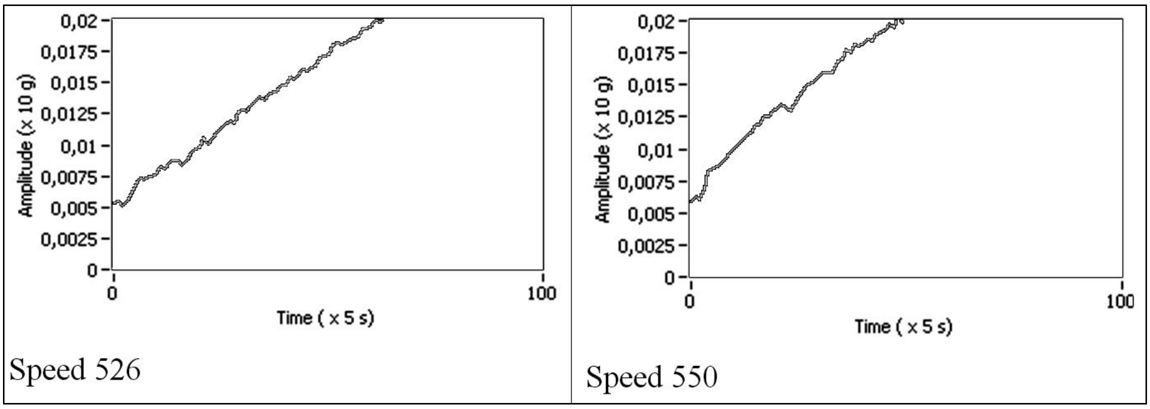


Figure 42. Vibration levels in resonance speed regions during long time measurement (8.3 min).

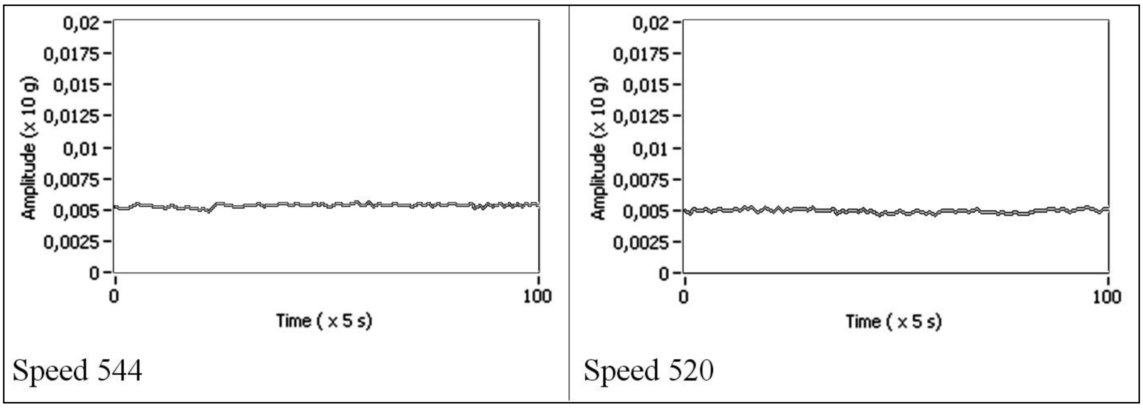


Figure 43. Vibration levels in anti-resonance speed region during long time measurement (8.3 min).

The results show that assumption of resonance states and low vibration states was correct. Vibration charts seem to be a valid way to select a line speed with low vibration. Low vibration speed areas exist at the bottom of valleys roughly within an interval of 20 m/min. As there are many low vibration speed areas in the high speed region, it should be possible to select a low vibration speed in case of industrial production.

6.3 Speed scheduling method

The collected vibration charts contain information about non-resonating line speeds on different running parameters. Although there are stable line speeds windows between resonances, nip vibration can crawl up with time even if the line speed is steady. Therefore it might not be preferable to use a single non-resonating speed because through speed changes lower level of vibration can be achieved.

Figures 44, 45 and 46 describe the influence of the speed change method on the vibration of the nip contact. Temperature is 25 ºC, so vibration chart in Figure 41 is valid. Used speeds don’t match exactly to either peaks or valleys in vibration charts. Reason for not selecting valleys in the charts is simply because resonance does not easily evolve on such speeds. On the other hand, using peaks in the vibration charts can easily lead to very strong resonance states. In Figure 44 and in Figure 45 vibration with stable line speeds of 570 m/min and 596 m/min respectively are shown. In a few minutes of time, vibration exceeds 0.2 *g* in both cases. Figure 46 represents nip vibration when speed change method is in use so that the line speed is changed between these two line speeds in every 10 seconds. During the measurement time of 16 min, vibration level stays below 0.2 *g*.

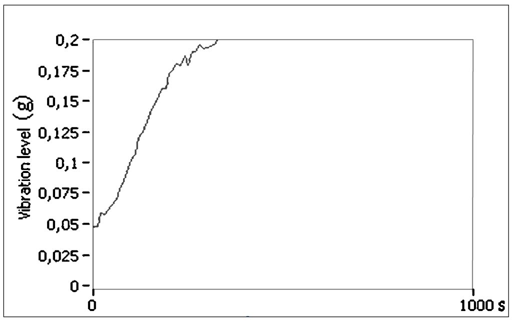


Figure 44. Vibration on fixed line speed of 570 m/min.

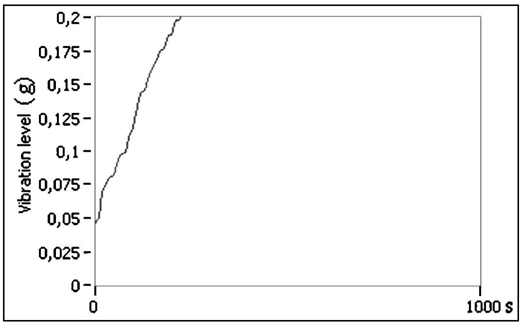


Figure 45. Vibration on fixed line speed of 596 m/min.

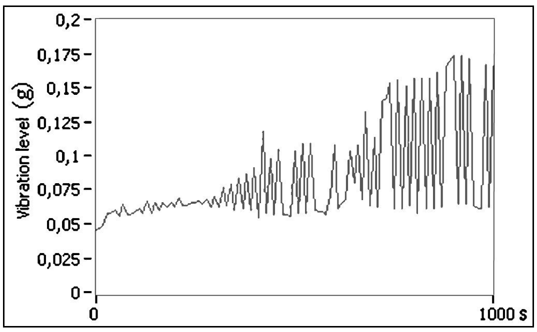


Figure 46. Vibration when utilising line speed scheduling between 570 m/min and 596 m/min with intervals of 10 seconds.

The measurement results indicate that when the running speed stays stable the deformation starts gradually to grow up and the level of the vibration rises. When the line speed is changed in every 10 second the deformation profile has not enough time to develop and the vibration stays in low level. The speed change method works in delay type vibration cases also in situations when the delay is crawling. The running of the machine is always a little bit in different phase of the excitation of the vibration. The speed change method is not so dependent on the limit values of the speed range if they are in low or in high vibration position in the vibration cart. In this example the low limit was, according to the Figure 41, near resonance and the other limit was closer to a valley area despite both speeds don’t match either peaks or valleys. The speed change can be implemented step by step or by sliding the line speed continuously.

6.4 Phase change method

Vibration in soft roll contact with integer number of waves in the cover creates surface deformation on soft roll covers. In conditions when deformation is not recovered before next nip contact, the delay vibration may start and act as an amplified excitation for vibration. This occurs if the deformations arrive nearly in phase with the vibration (Bradford, Emmanuel 1988). Rolls usually have different diameters. In the case of contact of two soft rolls, deformations in principle enter nip in different phases and act as two separate excitations. If one roll is hard, the deformation of the soft roll is a single excitation for delay resonance vibration. Delay resonance vibration has a frequency band, which is approximately 20 Hz wide in the surrounding of 120 Hz and is affected by machine operation parameters as explained earlier.

In hard-soft roll contact, shift of the phase of roll vibration compared to roll barring can be implemented through speed changes. This is due to change in time when wave profile arrives in the nip when compared to vibration phase of the rolls, which does not change in the speed change. In resonance, the two rolls in a nip contact vibrate with same frequency but opposite phase. This causes soft roll cover to deform, which will increase vibration in form of barring phenomenon. The barring waves and responding contact vibration will then amplify each others, in other words the contact is in resonance. Under certain conditions, deformation has been observed to increase vibration exponentially over time (Howe, Coscrowe 1963). If the roll rotational speed is changed, barring excitation will move out of phase with roll vibration. This will decrease vibration amplitude. The idea of phase change is to speed up or slow down roll speed for a short period of time so that the line is delayed in optimum case just half wavelengths of the wave profile of soft roll cover. In that sense phase shift is a rapid speed change method that prevents the delay mechanism to accumulate. The phase shift method may be easier to be implemented in production line cases in the viewpoint of for example web buffering than other speed change methods presented.

In order to perform phase shift for vibration of rolls, it must be considered what kind of speed step would be the most suitable. It seems sensible to use a quick step so the wave formation has less time to renew itself. The phase should be changed before the profile has recovered, to ensure attenuating counter-vibration on the roll cover. In preliminary measurements it was found that in the pilot roll press a short period speed step of +10 m/min is suitable for quick and effective phase change. Reason for such relatively large speed step is the PI speed controller of the rolls, which reacts slowly to changes. When the step is selected to a high value, a shorter time period for the step is sufficient. The phase shift experiments were carried out with direct drive configuration of the pilot roll press, Figure 47.

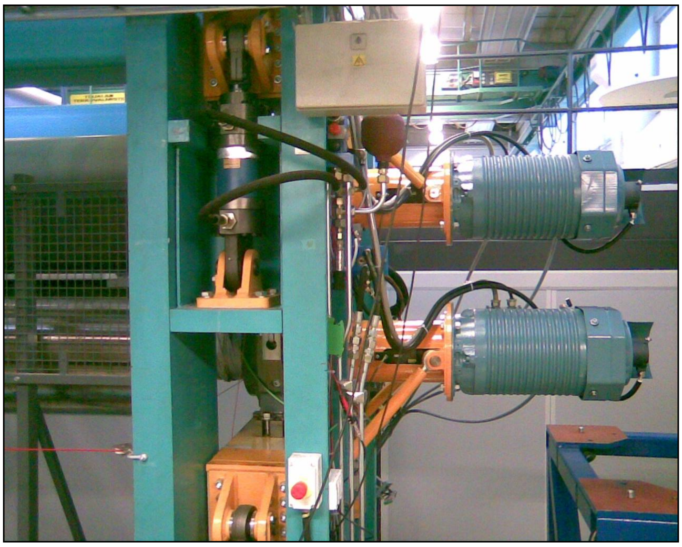


Figure 47. Speed controlled hard roll (lower) is the master drive roll and torque controlled soft roll is the slave roll.

As usually, hard roll is the master drive roll and controlled with speed control and the soft roll is the slave roll with torque control. In speed control the nip surface speed is calculated based on the measured RPM and on the assumed roll diameter. Therefore speed control would be better termed RPM control (Roisum 1998). Process control parameters are not accurate or sensible enough to detect a short-period speed step response. The soft roll surface position is especially hard to detect by process control variables. Therefore, a set of measurements was needed to determine accurate delay of step for phase shift. Measurements were performed by giving speed step commands for process control and monitoring speed changes directly from roll drive RPM control. The drive RPM control contained a pulse encoder with 1024 pulses per one revolution on the shaft of the electric motor. Speed commands were given through LAN from vibration control PC to process control PC (Salmenperä et al. 2002). The diagram of phase shift experiment arrangement is shown in Figure 48.

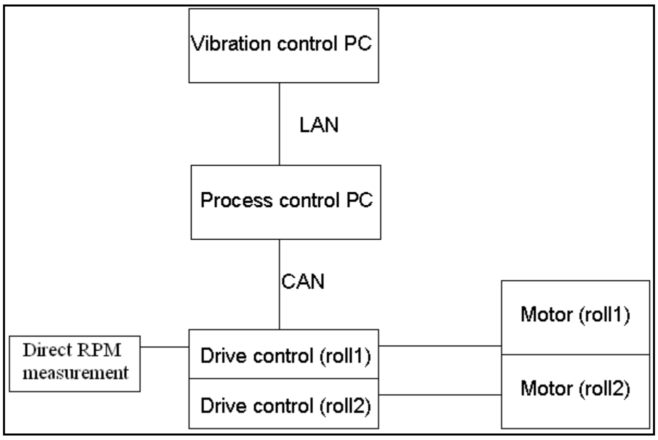


Figure 48. Diagram of phase shift experiment arrangement. Roll 1 is the soft roll.

With experiments it was found out that the needed speed step command could be performed in less than half second interval, and the roll cover would be delayed enough in few seconds. In Figure 49 it can be seen that the change time of the rotational speed is about 0.5 seconds and the length of the stable rotational speed is about five seconds. PI drive controller has a noticeably long effect for step response.

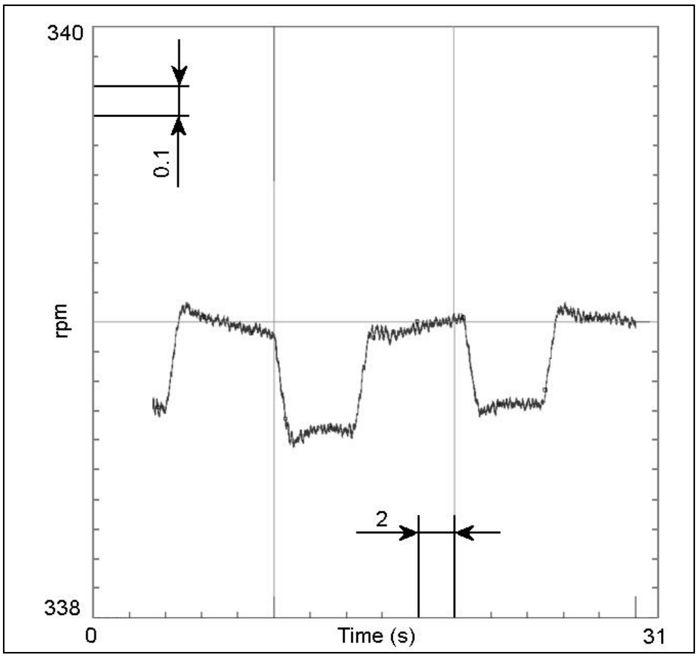


Figure 49. An example of phase change measurement with line speed of 590 m/min and step command of + 10 m/min for span of 350 ms.

Preliminary experiments with the phase shift method showed that the generation of the nip resonance can be avoided in some quantity. In order to test the method wider a phase shift control program was developed. The idea in the program is that it measures resonance frequency and calibrated rotational speed and calculates the needed delay time and executes it. The operation of the program is based on an experimentally constructed curve between delay time and delay length on roll surface. The curve is shown in Figure 50. The curve shows needed time delay to delay deformation for certain distance.

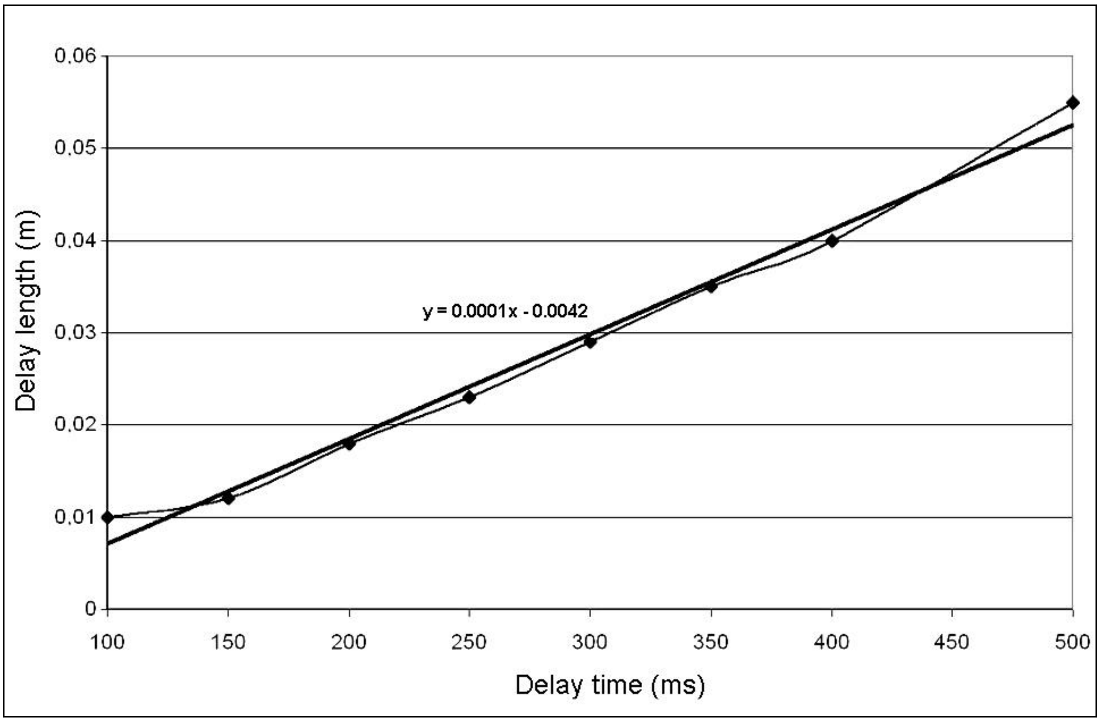


Figure 50. Experimentally constructed curve and linear regression between delay time and delay length on roll surface.

The phase shift control program, named Create phase shift was implemented with LabView software. The program consists of three parts. First part performs the measurement of the vibration signal and also visualizes it. The second part calculates the calibrated rotational speed, speed step length and starts the speed step. The third part performs the ending of the step. The loop time of the program is 10 seconds. Block diagram presentation of the phase shift program is shown in Figure 51 and in Figure 52 the part 2 is expanded.

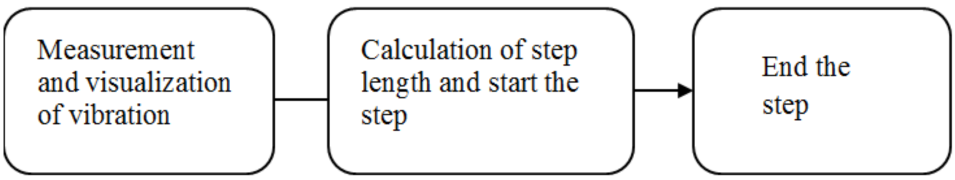


Figure 51. Block diagram of Create phase shift program.

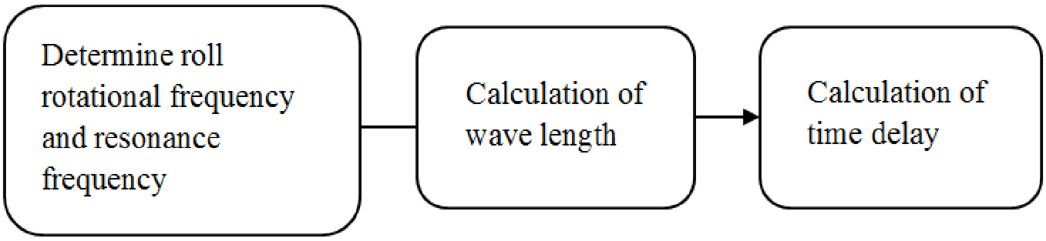


Figure 52. Block diagram of the calculation procedure in the second block of Create phase shift program.

Examples of the results of the phase shift program testing are shown in Figures 53 and 54. These figures show the development of the nip vibration during time. One step for x-axis corresponds to 10 seconds. From the left side, the resonance frequency and calculated delay to perform phase shift can be read. The phase shift step is applied once in 10 seconds. Figure 54 shows that phase shift mode on the vibration stays in lower level compared to the phase shift off mode. With a longer period, the effect of phase shift is smaller and with shorter period, the influence of phase shift does not have enough time to set to the opposite phase as a new phase shift is already applied. Therefore, 10 seconds was selected as maximum delay between phase shift steps.

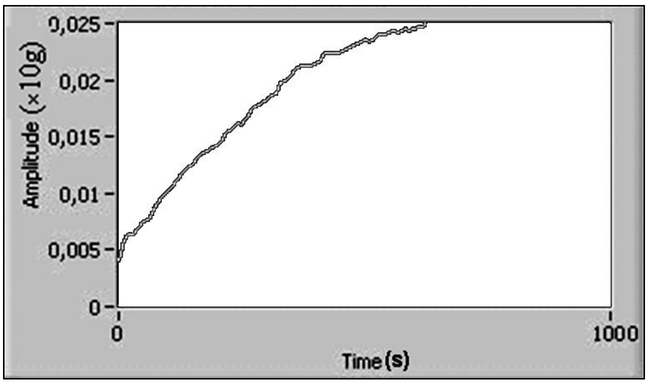


Figure 53. Vibration acceleration RMS curve with phase shift mode off. Line speed is 540 m/min.

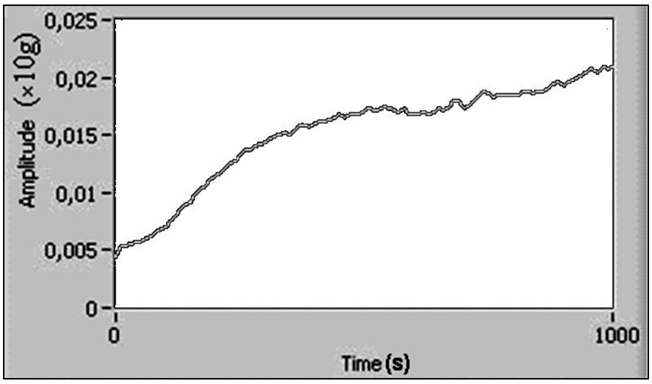


Figure 54. Vibration acceleration RMS curve with phase shift mode on. Line speed is 540 m/min, delay to use for phase change 168.9 ms and detected frequency is 124.2 Hz.

6.5 Vibration control by fuzzy speed setting or by using vibration charts

This section presents the idea of vibration control diagnostic systems and especially fuzziness in diagnostics, despite it has not been implemented in this work. Reason for this is that fuzzy systems suit well to imitate expert-like decision making. Diagnostic systems have evolved during last years and spread over all kinds of applications. Modern industrial systems usually also include remote monitoring and -diagnostics. Diagnosis results can be viewed and decisions can be made in real-time, on-line. The development in wireless communications cuts costs of remote diagnostic systems, especially because of the easy installation in mill environment.

Usually a diagnostic system utilises the sensor- and control signals of the target system, but it may as well have special diagnostic sensors. A diagnostic system often demands software, which may be located into the process control system for some applications. It can also be situated to a specific diagnostic processing unit. If robustness is an issue, the latter choice is better. If the diagnostic software utilises the same processing unit and memory than the target control system, the diagnostic system may disturb the control system if not special attention is paid (FDIR 2002).

When a diagnostic system has detected a fault situation, the next step is fault isolation and diagnosis. This means clarifying the cause, type, position and occurrence time of the fault. This information is needed for the diagnostic system to start necessary operations to retain optimal machine state (Chen 2001). In some systems the optimal machine state is reached through automatic procedures. More commonly this means an alarm, which is sent to control room or operator.

Many approaches can been taken for automatic fault diagnosis, including statistical, polynomial, neural network, fuzzy and neuro-fuzzy classification methods (Leonhardt et al. 1997). Vibration data is often or can easily become available from rotating machinery, but in many cases there is little provision for taking full advantage of its potential for improving reliability in machine condition diagnostics. There is also ample domain knowledge about machinery malfunction, but fault diagnosis is usually far too complex to be reliably provided by simple expert rules. There is a clear need for methods capable of simultaneously handling numerical data and human expert knowledge. Fuzzy systems methodologies are well suited for such problems, as they can naturally process both numerical data and linguistic information.

Fuzzy control systems are widely adapted to process industry. Usual applications are sequential processes. The first common notice of fuzzy control is the smoothness of sequence changes. The fuzzy diagnostic control for roll rotational speed in the presented system supervises the functionality of the production and prevents breakdowns. Remote fuzzy diagnostic control module receives process variables through process control LAN. Inference results are transferred to process control, which interprets them as diagnostic control commands. Initial simulation results are presented.

Fuzzy set theory was first introduced in 1965 by Lofti A. Zadeh (Zadeh 1965). It may be regarded both as a generalization of classical set theory and as a generalization of dual logic. For a classical set of objects which is to be analyzed by fuzzy means, then

$$\tilde{A} := \{(x, \mu_{\tilde{A}}(x)); x \in X\} \tag{79}$$

represents a fuzzy set of X, where μ_A represents the membership function and $\mu_A(x)$ represents the degree of membership of element x to the fuzzy set $\tilde{A}$. A fuzzy number is a convex, fuzzy set, normalized over the interval [0,1] of the set of real numbers. Precisely one element with a degree of membership 1 exists. The membership function of this fuzzy set is piecewise continuous. A linguistic variable is a variable whose values are not numbers, but rather linguistic constructs. The contents of these terms are defined by fuzzy sets over a base variable.

Fuzzy rule base –technique is based on fuzzy reasoning of rules, which represent wanted actions in certain situations. It is especially applied to fuzzy control in industry. The calculation proceeds in three stages: 1. Fuzzification of input variables; 2. Fuzzy reasoning; 3. Clarification of output variable. The flow chart of the calculation is presented in Figure 55.

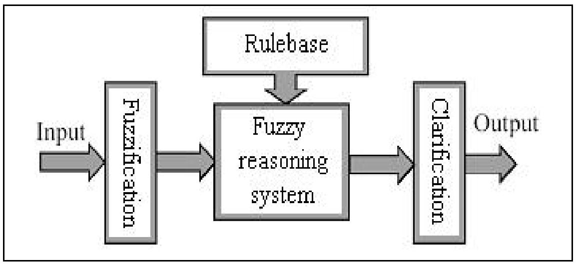


Figure 55. Fuzzy rule-based calculation flow chart.

Linguistic terms are defined for the input –and output variables. They can be based on expert knowledge, through fuzzy models or through learning (Driankov et al. 1993, Karray et al. 2004).

In vibration control the fuzzy logic can serve the purpose to find a stable enough state between two resonance speeds. The *rms*-value of the barring vibration speed v_{rms} be classified according to ISO 10816-1 standard and the linguistic set evaluating the *magnitude* of the vibration could be then

$$\{stable, small, acceptable, disturbing, marking, damaging\}$$

The rule basis could then propose according to the *magnitude* the following actions

If the previous speed increase is changing the situation x steps better/worse, change speed y steps higher/lower

when the speed setting follows the following quantified portions of the set value

$\{0.1\%, 0.2\%, 0.4\%, 0.8\%, 1.6\%\}$

Vibration charts can be used also as a source data for a vibration control system following simple but well working engineering practice. However, the charts alone do not contain a way how to utilise them. For automate the use of vibration charts it is possible to develop several kinds of algorithms avoiding resonance based on vibration charts. The next algorithm is presented as an example of a suitable way for resonance avoidance:

1. A vibration limit accepted for roll nip oscillation is selected.
2. Suitable speeds for current soft roll temperature are offered. Some of them may be chosen.
3. Highest chosen speed is set.
4. Vibration trend is monitored.
5. If vibration limit is exceeded, a new speed, highest available is chosen. New speed cannot be the speed selected in section 3.
6. If vibration limit is instantly exceeded, a new speed is selected. New speed cannot be the speed selected in section 3 or 5.
7. Move to section 4.

The algorithm described is designed to avoid constant switching between speeds that induce too high vibration levels. Other ways to design such a system could contain a history of attempted speeds. The history information could enable artificial intelligence to neglect speeds leading to undesirable vibration levels.

6.6 Feasibility analysis of speed variation approach for delay-resonance control

The supervised speed control methods for avoidance of delay resonance presented in this study are focused on soft line contact of two rolls. This kind of contact situations can be found in coating and soft calendering units in paper machines, in printing machines and for example, the contact between car tyre and road surface is a soft nip contact. Although in this study the soft nip contact has been on view, the barring resonance vibration exists also in hard rolling contact. In the case of the hard contact, the resonance frequency can be considerable higher than in soft nip contact and no delay phenomenon occurs. Hard nip contacts can be found in hard calendering units like calender stacks in paper machines, in nip contact of grinding stone and cylinder and in contact of a train wheel against the rail.

The essential technical preconditions for applying presented vibration avoidance methods especially in phase shift method are the following ones:

- a backlash-free connection between drive mechanism and the driven component
- an accurate enough pulse encoder of the roll shaft
- data acquisition unit with high sampling rate
- fast responding control algorithm
- drive unit with high enough torque performance

The backlash-free connection can be achieved for example with direct drive applications where the drive unit is rotating the controlled shaft without too flexible coupling. Another application can be the use of backlash-free gear transmissions. The operation environment of the component, machine or process line unit must be such that rotational frequency or line speed changes are possible to carry out without disturbing the operation preconditions. In autonomous operating machines, it is possible to implement the speed change methods for vibration control without any special constructional changes. In the case of process lines like paper machines, the change in line speed of one unit can create problems in line tension or in operation of other units. In these cases, a continuous upward and downward speed change method or continuous phase change method with buffers for the product could be a possible solution.

Surfaces in the line contact are exposed to wear and vibration can create the run out to be non-symmetrical. Abrasion affects the parameters of the surfaces and vibration models may become invalid in time. It may be necessary in these cases to re-establish vibration charts with new measurement sets. This situation holds, when material removal processes take place by means of grinding or wearing in roll manufacturing and in train wheel case. Paper machines on the other hand are very complex machines and probably the same model wouldn’t be valid in different machines. In these cases the phase shift method could still be valid since as long as periodical accumulation of vibration is disturbed, it will help to decrease vibration.

7. Conclusions

A fresh research line has been opened to control vibrations of roll nip systems by intelligent speed variation methods. When the main stream in the research has offered passive, semi-active and active vibration damping methods and related devices, this approach utilises the characteristic nature of delay-resonance states of polymer covered rolls for finding speed windows with acceptable vibration amplitude. The research work toward this goal has been shared to two main tasks:

1. Development of knowledge basis for prediction of delay-resonances based on theoretical models and experimental vibration monitoring.
2. Generation, implementation and evaluation of three different speed variation approaches to control delay-resonances.

The research work performed has led to new scientific knowledge of nip oscillations and a family of algorithm-based vibration control approaches to be offered for the industry.

7.1 Factors controlling the delay-resonance

Existence of delay-resonances can be theoretically shown by analyzing the distribution of the poles of the system equations describing the self-excited vibration case including the delay-effect of the polymer cover. This knowledge has now been empirically verified for different speed changing programs. The most important findings are the ones:

1. As predicted, there is a discrete spectrum of resonance speeds and corresponding running frequencies f_{rot} which are distributed according to a hyperbolic law $f_{rot} = n^{-1} f_{nat}$, in which n is the number of waves at the roll cover and f_{nat} is the natural frequency of the nip oscillation. This spectrum has been

experimentally found by sliding the speed upward and downward showing more organized behaviour in the latter case.

2. The barring vibration appears after a delay of several hours from the start-up, when the cover temperature has stabilized during steady running conditions.
3. The resonance spectrum appears below certain value *n* of wave numbers and beyond corresponding threshold speed.
4. The resonance behaviour for similar running parameters is always the same if similar tests have been repeated after a brake of several hours, several days or several weeks.
5. If the barring vibration is fully developed and the roll speed is slightly shifted, a strong beating phenomenon appears, but dies out with time.

Moreover, a systematic work has been done to identify the important factors controlling the vibrations. As already known, based on the use of different traditional damping methodologies, the design parameters of rolls and other parts of the system have great influence on the response amplitude. As they are fixed ones and studied elsewhere, the main interest in this research has been in the adjustable running parameters. This is a set of three parameters: running speed level, line speed level and cover temperature. The effect of these parameters has been experimentally verified during extensive set of test runs, the output of which is the following qualitative response behaviour:

1. Cover temperature is the most important running parameter. High enough temperature of cover is a necessary condition for the barring phenomenon to start. For a cool cover, which is the case after start-up, no vibration appears. Such temperatures close to room temperature are, however, difficult to maintain because the web handling process requires higher temperature and the polymer hysteresis makes a rise of the cover temperature.
2. When using an active heating/cooling circuit for the covered roll, cover temperature can be considered as an independent running parameter. The role of this circuit is to preheat the cover to make faster start-ups and during start to limit overheating of the cover. During normal run the cover temperature is elevated and this clearly changes the elastic and viscous behaviour of the cover making it softer and slower to recover. This effect increases the dominating

recovery factor γ thus gaining the feedback mechanism and leading to a clear generation of discrete delay-resonance speeds.

3. For a roll cover, passively heated as a balance of heat input from the rolling hysteresis and heat output via forced convection to room air, the response of roll cover is time-dependent and can be considered more as a process condition than a real running parameter. When knowing that the heat input is proportional to the rolling resistance $M = \eta E^{-1} vN$, the cover temperature follows the speed and line speed level making the temperature a dependent parameter.
4. When applying a longer run at higher speeds and higher line loads, the increase of the cover temperature completely changes the polymer response showing response, in which the regular location of delay-resonances vanishes as the cover behaves dominantly like a plastic material.
5. The resonance speeds depend on the roll temperature and line load making the whole spectrum to travel.

The conclusion of this knowledge is that the cover temperature has to be precisely controlled to keep the system behaviour constant and better predictable. The regular enough location of delay-resonances forms also a promising basis to develop algorithm-based resonance avoiding approaches.

7.2 Performance of the speed variation methodology

Three methods for resonance avoidance based on speed variation approach were developed during this work:

1. Setting speed to safe windows at the valleys between two successive resonance peaks.
2. Changing speed according to some schedule to systematically disturb the cumulative wave-formation effect at the roll cover.
3. Applying angular changes to the roll position making the barring excitation in opposite phase with respect to the natural oscillation.

The experiences of the proposed algorithm-family can be concluded to the following observations:

1. The method to fix the speed to the average speed of two successive peak speeds needs precise information on the resonance chart. If this is not available, it must be created by means of a separate system identification run. When the system is in steady condition, this method works well. If the line load and cover temperature are changing, the resonance chart has to be updated to avoid risk of sliding to resonance. This method does not work sufficiently in simplest form and requires some intelligent rule basis to be added on the top.
2. A systematic speed oscillation or some other sliding schedule brings acceptable response levels in average although peaks change almost random in time. Such wave mixing approach can be realized in many different ways.
3. A third way to mix wave formation is to apply transient speed changes to make phase shifts to the roll. It also limits the vibration level effectively but brings a risk of fatigue to the components in the power train and can thus be recommended to the direct drive cases only.

As a summary, fixed speed methodology can be updated to be a flexible one by adding a fuzzy logic based ruling basis for processes, where the speed is not set to a certain precise value for other reasons. Speed scheduling method is effective for cases, where the speed can be modified continuously as is the case in reeling or winding processes. In contrast to the other methods, phase shift method can be used for all speeds as it does not modify the speed. By arranging every second phase to be driven by acceleration and every second by deceleration, the need of buffering the web is minimized.

7.3 Future work

Major part of the open research questions were solved during this work. Some findings needing deeper analysis have been left to the subject of future work.

A challenging problem related to the role of the roll cover is the thermo-mechanical response analysis of the polymer cover. This requires theoretical/empirical modelling of the polymer in elevated temperatures at time scales typical for rolling contact. The elastic foundation model should be also developed to include other deformation effects than the compressive ones only.

Another area worth of investigation is the development of fuzzy logic or some other soft computing based algorithms to manage the overall problem of learning the system behaviour and constructing a way to avoid resonances based on this information. Rule-based fuzzy control can straightforwardly be implemented by using the knowledge basis about the vibration control task. This work has been left to the next industry-driven part of the research.

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