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Date 22, Nov. 82
ABSTRACT

Due to rising lumber costs it has become more and more important to optimize the cutting performance of bandsaws as they are used in the forest products industry. One important aim is to reduce the lateral and torsional deflections of the blade to minimize the thickness of the cut. This study tries to assess the influence of various typical bandsaw parameters on the stress distribution in the bandsaw blade and on the vibrational behaviour of the stationary and the running sawblade. Experiments were performed with a fullsize industrial production bandsaw. The experimental results for the stress-strain measurements of the stationary sawblade are compared with results from theoretical solutions.

Vibration measurements of the stationary and of the running blade are compared to values derived from MOTE'S [9] flexible band solution and KANAUCHI'S solution [21] for lateral deflection and with ALSPAUGH'S [2] solution for torsional vibrations. The results presented show that the static stress measurements agree very well with the analytically predicted results. Major factors influencing the stress distribution in the blade: the axial prestress, stresses due to bending of the blade
over the bandsaw wheels, and stresses due to tilting of the top bandsaw wheel were examined. A comparison of the experimental and analytical results of the vibration measurements for the stationary sawblade showed very good agreement for the lateral natural frequencies, while the torsional natural frequencies were considerably higher than the analytically predicted values. Similar results could be observed for the natural frequencies of the running blade.

The mode shapes of the stationary sawblade at the natural frequencies were measured. It was found that the modeshapes consisted of coupled modes for most tiltstress differences.
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NOMENCLATURE

A = cross section of the blade
B0 = width of the blade from tooth gullet to backside
B = total width of blade
Bw = width of bandsaw wheel
Bs = distance between straingage location SG1 and SG7
Bt = distance center of top wheel to point of
tiltangle rotation

c0 = wave velocity
c = sawblade velocity
D = band flexural rigidity \( (EH^3/12(1-v^2)) \)
fL1 = 1. lateral natural frequency
fL2 = 2. lateral natural frequency
fT1 = 1. torsional natural frequency
fT2 = 2. torsional natural frequency
E = Young's modulus of elasticity
Fvert = vertical cutting force (in x-direction)
Flat = lateral cutting force (in z-direction)
Fhor = horizontal cutting force (in y-direction)
fC = height of crown
G = shear modulus of elasticity
g = gravitational acceleration
H' = band thickness
Ks = wheel support system stiffness
L = band length between guides
\( L_w \) = distance between center of rotation of saw wheels

\( n \) = number of data samples

\( 2R_0 \) = total axial prestressing force

\( R_0^* \) = axial prestress

\( r \) = radius of saw wheels

\( r_c \) = crown radius

\( s \) = standard deviation

\( \text{SIG0}(x) \)

\( = \) axial prestress

\( \text{SIGb}(x) \)

\( = \) bending stress due to \( r \)

\( \text{SIGc}(y) \)

\( = \) bending stress due to crown

\( \text{SIGf}(x) \)

\( = \) stress due to centrifugal forces

\( \text{SIGn}(x) \)

\( = \) stress due to the notch factor at the blade teeth

\( \text{SIGr}(x) \)

\( = \) stress due to pretensioning

\( \text{SIGta}(x) \)

\( = \) stress due to the tiltangle

\( \text{SIG}(x) \)

\( = \) stress in x-direction

\( \text{SIGtemp}(x) = \) stress due to temperature changes
\[ \text{SIG}_w(x) \]

- Stress due to cutting forces

\[ T \]

- Transmissibility

\[ t \]

- Temperature

\[ \text{TA} \]

- Tiltangle

\[ u \]

- Feedspeed

\[ \bar{u} \]

- Mean value of \( n \) data values \( x \)

\[ Z \]

- Tooth spacing

\[ \alpha \]

- Temperature gradient

\[ \nu \]

- Poisson's ratio

\[ \kappa \]

- Wheel support stiffness factor

\[ \rho \]

- Density

\[ \eta \]

- Support stiffness constant \( (\eta = 1 - \kappa) \)
1 INTRODUCTION

1.1 Background

Due to the rising cost of lumber in the forest products industry the need of a thorough understanding of the influence of the governing parameters on the cutting performance of the bandsaw has become a necessity.

To achieve this objective, in 1981 a woodcutting laboratory was set up at the Department of Mechanical Engineering at the University of British Columbia, consisting of a vertical 5 foot production bandsaw together with a specially designed log carriage system and a sophisticated instrumentation and data acquisition system.

The parameters governing the performance of a bandsaw can be separated into three different groups:

a) human influence factors (i.e. The bandsaw operator chooses certain bandsaw parameters by experience)

b) design parameters established by the bandsaw designer

c) stochastic influence factors (e.g. inhomogenities in the lumber,
temperature changes, etc.)

The objective of this study is to examine the influence of parameters characterized under 1.a) and 1.b) on the stress distribution in the sawblade and on the vibrational behaviour of the saw.

1.2 Experimental aims

To achieve this objective the experiments were divided into three major parts:

The first part consists of strain measurements of the stationary sawblade. The parameters which were changed were the axial prestress SIG0(x), the tiltangle TA and the position of the blade around the saw. The tiltangle TA describes the rotation of the top wheel around the z-axis (Figure 3).

The second part consisted of vibration measurements of the stationary sawblade, excited with an electro magnet. The two lowest lateral and torsional natural frequencies were measured for various axial prestresses R0* and different tiltstress differences TSD. In this part a different representation for the axial prestress and for the tiltangle were chosen to allow normalized quantities. The transmissibility of the blade was measured at the previously identified natural
frequencies for different axial prestresses $R_0^*$ and tiltstress differences $TSD$.

The third part of the research program consisted of vibration measurements of a free running sawblade at different axial prestresses $R_0^*$. The wheel support stiffness $K_s$ (MOTE [9]) was measured.

1.3 Previous research

Over the years various researchers have covered some of the experiments done in this study. In 1970 THUNELL [22,23] summarized in a publication various mathematical formulae to calculate stresses in a sawblade due to the geometry of the saw, to idling and to cutting. In 1971 PORTER [19] did stress experiments with a stationary sawblade and compared his results with analytical solutions. In 1972 PAHLITSCH [13,14] analysed the different stresses in bandsaw blades and compared them to experimental results. In 1977 KIRBACH and BONACH [7] measured the effect of wheel tilting and saw tensioning on the natural frequencies of a stationary sawblade. In 1981 TANAKA [21] did an extensive experimental study on stresses in bandsaw blades comparing the behaviour of five different blades with each other. He also examined the vibrational behaviour of these blades for stationary and running conditions.
and compared them to analytical solutions.

In the last 20 years a number of researchers presented analytical solutions to try to predict the natural frequencies of a running bandsaw blade. To compare the experimental data of this study with analytical solutions, MOTE'S flexible band solution [9] and KANAUCHI'S model for lateral vibrations [21] and ALSPAUGH'S analytical solution [2] for torsional vibrations were used.
2 EXPERIMENTAL SET UP

2.1 Band saw facilities

For the experiments presented in this study a vertical 5 foot production bandmill was used (see Figure 1). The saw is equipped with a hydraulic straining system (Figure 2) which can provide an axial prestressing force \( 2R_0 = 45000 \) N to \( 90000 \) N. The top wheel can be tilted by an electro motor (Figure 2) and the whole saw can be positioned towards a carrier which passes a log by the saw. Above and below the cutting region, the bandsaw blade is guided by two pressure guides. During idling and cutting the blade is cooled by waterjets.

The saw was equipped with a pretensioned bandsaw blade with an unknown internal stress distribution. The dimensions and the chosen coordinate system for the saw and the blade are shown in Figures 3 and 4 and are also listed in Table I.

2.2 Instrumentation and data acquisition system

To measure the axial prestressing force \( 2R_0 \), a link in the hydraulic straining system was equipped with a 4-arm temperature compensated strainingage bridge. This transducer will be referred to as "loadcell" or "LC".
FIGURE 1  BANDSAW SET-UP
FIGURE 2 STRAINING SYSTEM AND TILT ANGLE MOTOR
FIGURE 3 DIMENSIONS OF THE BANDSAW
$H = 1.651 \text{ mm}$

$Z = 44.375 \text{ mm}$

$B_o = 242. \text{ mm}$

$B = 260. \text{ mm}$

**FIGURE 4** STRAINGAGE LOCATION ON THE BLADE
A₀ = 398.7 mm = cross-section area of the blade
B₀ = 241.5 mm = width of the blade from backside to gullet
B = 260 mm = total width of blade
Bw = 228.6 mm = width of wheel
Bs = 222 mm = distance between SG1 and SG7
Bt = 222.3 mm = distance between LC and center of wheel

c = 40.7 m/s = blade velocity
D = 86347 Nmm = flexural rigidity
E = 2.1*10⁵ N/mm² = Young's modulus
fc = 0.102 mm = height of crown
G = 80000 N/mm² = modulus of elasticity
H = 1.651 mm = thickness of blade
Ks = 825 N/mm = support stiffness
L = 762 mm = blade length between guides
Lw = 2464 mm = distance between saw wheel axes
r = 762.5 mm = radius of bandsaw wheels
rc = 64 m = radius of bandsaw crown
t = 19 mm = overhang of pressure guides
Z = 44.23 mm = tooth spacing

Table I Parameter list for the saw used in the experiments
The calibration curve for the loadcell LC including the standard deviation \( s \) and the error bounds for a confidence value of 95% are shown in Figure 5.

The tilt angle of the top wheel could be measured with a calibrated digital counter (see Figure 2). The calibration factor of the tilt angle counter is 675 digits /degree tilt, with a standard deviation of 59 digits/degree for \( n=10 \). The measurement chains described below are shown in Figure 9 and the equipment and instrumentation used in the experiments are listed in Table 1a.

To measure static as well as dynamic quantities, two different measurement systems were used. For static and dynamic strain measurements, 9 strain gauges were cemented on the outside of the blade (see Figure 4). For some experiments 3 strain gauges were mounted at position 1, 4 and 7 on the inside of the blade. The data acquisition system for these strain gauges consisted of a NEFF 620/300 signal conditioner which provided constant voltage supply and bridge completion resistors as well as individual bridge balancing for up to 64 channels simultaneously. The conditioned signals were then fed into a NEFF 620/100 A/D converter and amplifier which provided fixed and programmable amplification and analog digital conversion for each channel. The digitalized data values were then multiplexed and sent from the.
FIGURE 5 CALIBRATION FOR LOADCELL LC OF THE WHEEL SUPPORT SYSTEM UNDER A PRESS (MECHANICAL ENGINEERING UBC)

SENSITIVITY $\bar{x} = 19298 \text{ N/mV}$

$n = 11$

$S = 103.4 \text{ N/mV}$

FOR A CONFIDENCE OF 95%:

$19229\text{N/mV} < \bar{x} < 19367\text{N/mV}$
FIGURE 6  CALIBRATION OF LOADCELL FOR THE EXCITATION FORCE OF THE ELECTROMAGNET WITH WEIGHTS

SENSITIVITY $k = 44.5 \text{N/V}$

CHARGE AMPLIFIER DATA:

SENSITIVITY SETTING = 8.975

AMPLIFICATION RANGE = 20

FOR A CONFIDENCE OF 95%:

$n=9$, $s=0.1 \text{N/mV}$,

$44.5 \text{N/mV} < \bar{x} < 44.7 \text{N/mV}$
laboratory through a NEFF 620/500 control processing unit to a PDP 11-34 computer for data processing. The data handling and processing could be controlled through an intelligent Tektronix 4051 terminal in the laboratory which was interfaced to the PDP 11-34. Software graphing facilities allowed one to view the experimental results on the Tektronix screen and to make hardcopies on a connected digital plotter. To run this system, various FORTRAN programs were developed; these are documented in APPENDIX IV. The program NEFF2 scans n channels with a chosen scanning frequency and sampling rate and stores the data in user-specified data files.

The program STRAIN calculates the static two-dimensional stress distribution in the blade from straingage readings at different positions around the saw. It compiles the data from the straingages on the blade and the loadcell, using previously established calibration factors.

The program DYN calculates the stress distribution in the blade from straingage readings during a chosen time duration.

For vibration measurements a different data acquisition system was used. The blade was excited with an electro magnet. The excitation current was generated by a Bruel and Kjaer frequency generator, amplified by a power amplifier and fed into the electro magnet. The
magnet was mounted in a frame which allowed a three-dimensional positioning of the magnet behind the blade between the guides. The excitation force of the magnet was measured with a piezo-electric loadcell and amplified by a charge amplifier. The calibration curve of the loadcell with its standard deviation and the error bounds for a 95% confidence value is shown in Figure 6.

Vibrations of the sawblade were measured with 2 non-contacting eddy-current transducers. Their calibration curves are shown in Figure 7 and 8. Both the excitation and the blade response signals were fed into a 660A Nicolett Dual Channel Frequency Analyzer. The analyser could be programmed to accept the necessary calibration factors and then calculate the transfer function, transmissibility, RMS spectrum and the coherence function between the two channels. Results could be shown on a screen and be plotted out onto a hard copy. A second method of documentation was to enter the data from the frequency analyser into the PDP11-34 and plot them out with the Tektronix terminal and the digital plotter.
SENSITIVITY = $0.344 \text{ mm/V}$

IN THE RANGE FROM 1mm TO 2.75mm

PROBE #1

I = DATA RANGE FOR A CONFIDENCE RANGE OF 95%, n = 3

FIGURE 7 CALIBRATION OF #300 BENTLEY NEVADA EDDY-CURRENT DISPLACEMENT PROBE WITH A FEELER GAGE
FIGURE 8 CALIBRATION OF #300 BENTLEY NEVADA EDDY-CURRENT DISPLACEMENT PROBE WITH A FEELER GAGE

SENSITIVITY = -0.36 mm/V
IN THE RANGE FROM 1mm TO 2.75 TO 2.75mm
PROBE #2
I = DATA RANGE FOR A CONFIDENCE RANGE OF 95%, n = 3
FIGURE 9 MEASUREMENT CHAINS
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<tr>
<td>9</td>
<td>NEFF 620/500 demultiplexer and data storage</td>
</tr>
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<td>10</td>
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Table 1a  LIST OF INSTRUMENTATION AND EQUIPMENT
3 THEORETICAL STRESS EVALUATION FOR A BANDSAW BLADE

3.1 Fundamental theory of stress calculation for a bandsaw blade

The bandsaw blade is subject to a variety of different stresses while it is stationary, during idling and during cutting. To keep the blade positioned on the wheels and to increase the blade stiffness an axial prestress \( \text{SIG}_0 \) is imposed on the blade by forcing the top wheel upwards with a force \( 2R_0 \). This force results in a constant stress distribution across the blade. Additional to this constant stress distribution across the blade, non constant stresses exist due to the crown of the wheels, the tiltangle of the wheels, the amount of pre-tensioning (rolling) done to the blade and stress concentration at the gullet of the teeth.

During idling and cutting the blade furthermore is subject to centrifugal forces, temperature stresses and stresses resulting from the cutting forces. The stresses acting in a bandsaw blade can be classified in the following way:

**static stresses**
- axial stress \( \text{SIG}_0(x) \) due to axial prestressing force \( 2R_0 \)
- bending stress \( \text{SIG}_b(x) \) due to bending of the blade over the wheels
- stress \( \text{SIG}_c(y) \) due to crown of the wheels
- stress \( \text{SIG}_t(a)(x) \) due to the tiltangle of the top wheel
-stress $\Sigma G(x)$ due to the notch factor of the blade teeth
-stress $\Sigma Gr(x)$ due to pre-tensioning of the blade

additional stresses during idling
-stress $\Sigma f(x)$ due to centrifugal forces

additional stresses during cutting
-stress $\Sigma w(x)$ due to the cutting forces
-stress $\Sigma temp(x)$ due to temperature changes

3.2 Static stress for the stationary blade
3.2.1 Axial prestress $\Sigma G_0(x)$ due to the axial prestressing force $2R_0$

The axial prestress $\Sigma G_0(x)$ is caused by the force $2R_0$. Using the cross section area $A_0=B_0*H$ of the blade, where $B_0$ is the width of the blade from the backside to the gullet and $H$ is the thickness of the blade, the axial prestress $\Sigma G_0(x)$ can be calculated from:

$$\Sigma G_0(x) = \frac{2R_0}{2} \cdot \frac{1}{B_0*H}$$

3.2.2 Bending stress $\Sigma b(x)$ due to the radius $r$ of the bandsaw wheels

The stress $\Sigma b(x)$, resulting from bending of the blade over the cylindrical wheels results in a stress which can be calculated from:

$$\Sigma b(x) = \frac{E}{1-v^2} \cdot \frac{H}{2r}$$
With $E=$ Young's modulus, $v =$Poison's ratio and $r =$radius of the bandsaw wheels.

For the saw used in these experiments this stress component amounts to:

$$\text{SIG}_b(x) = 250\text{N/mm}^2$$

### 3.2.3 Stress $\text{SIG}_c(y)$ due to crown of the wheels

Under the assumption that the curvature of the crown follows a circular arc of radius $r_c$, the stress $\text{SIG}_c(y)$ can be calculated from:

$$\text{SIG}_c(y) = \frac{E H}{1-v^2} \frac{1}{2} \left( \frac{1}{r_c} + \frac{1}{r} \right)$$

with $r_c=$radius of crown.

Again for our saw this results in a maximum stress:

$$\text{SIG}_c(y) = \frac{2.1 \times 10^5\text{N/mm}^2}{1-0.3^2} \times \frac{1.675\text{mm}^2}{2} \times \left( \frac{1}{64 \times 10^3\text{mm}} + \frac{0.3}{762.5\text{mm}} \right) = 79\text{ N/mm}^2$$

### 3.2.4 Stress $\text{SIG}_{ta}(x)$ as a function of the tilt

Using the geometric relationship of the top wheel support and the tilt system (see APPENDIX I) the stress
SIGta(x) due to the tiltangle for each straingage position can be calculated from:

\[
SIGta(x) = \frac{\Pi (straingage position) *TA*E*}{180 Lw}
\]

where: TA=tiltangle of the topwheel (degrees) and Lw=distance between wheel axes.

3.2.5 Stress SIGn(x) due to the notch factor of the blade teeth

Photoelastic studies of elliptical shaped gullets (PORTER [19]) show that the stress concentration factor can reach a magnitude of 2.0 to 2.5. Studies by other authors show similar results:

SUGIHARA [22] reports a stress concentration factor of 1.3 to 2.5 while KRILIOV [8] cites a value of 1.35 for bending and 2.0 for tension.

3.2.6 Stress SIGr(x) due to pretensioning of the blade

The direct measurement of the inplane stresses induced into a blade during the pretensioning process is very difficult. Therefore in many sawmills the shape of the deflected blade in x and y-direction is measured (light-gap method) to obtain a measure of the influence of the pretensioning. Nevertheless measurements have been taken by some researchers. KRILIOV [8] reports that
the stress $\Sigma G_r(x)$ reaches a tensile level of 30 to 70N/mm$^2$ while BAJKOWSKJ measures a tensile stress $\Sigma G_r=65\text{N/mm}^2$ with intermittent compressive stress zones of up to 550N/mm$^2$ (PAHLITSCH [13]).

3.3 Additional stresses during idling

3.3.1 Stress $\Sigma G_f(x)$ due to centrifugal forces

The centrifugal forces which act on the blade during running result in a stress:

$$\Sigma G_f(x) = \rho \times c^2$$

with $\rho=$mass density of the blade

During the experiments the bandsaw ran with a velocity of $c=40.7\text{m/s}$, which results in a stress $\Sigma G_f(x)=15.8\text{N/mm}^2$.

3.4 Additional stresses during cutting

3.4.1 Stress $\Sigma G_w(x)$ due to the cutting forces

Up to date very little experimental research has been done concerning the cutting forces. FEOKTISKOV [13] reports that under extreme conditions ($c=45\text{m/s}$, $u=1\text{m/s}$ and depth of cut is 300mm) a cutting force $F_{cut}=900\text{N}$ can
be reached. He calculated that even under such unfavorable conditions the resulting stresses are negligible compared to the stresses resulting from the axial pretensioning force $2R_0$ and from bending of the blade over the wheels. The same can be said for stresses resulting from the cutting forces $F_{lat}$ and $F_{hor}$ (PAHLITSCH [13]). KRILOV [8] determines that the stress $\text{SIG}(x)$ resulting from the cutting forces can reach levels of up to $7 \text{N/mm}^2$.

3.4.2 Stress $\text{SIG}_{\text{temp}}(x)$ due to temperature changes

A study by SAITO and MOVI (1953 [22]) shows an average temperature change in the blade of $45^\circ$ C during cutting moist wood with a feed speed of $u=0.47 \text{m/s}$. Such a temperature difference (assuming rigid wheelsupports) would result in a temperature stress of:

$$\text{SIG}_{\text{temp}} = E*\alpha_t*(t_2-t_1)$$

$$= 2.1*10^5 \text{N/mm}^2*\frac{11*10^{-6}}{\text{deg}}*45\text{deg} = 104 \text{N/mm}^2$$

with $\alpha_t$=linear coefficient of expansion, 
$(t_2-t_1)$=temperature change in the blade

This shows that a temperature change in the blade has a significant influence on the sum of the axial stresses
acting in a bandsaw blade.
4 STATIC STRESS-STRAIN MEASUREMENTS WITH A STATIONARY SAWBLADE

4.1 Experimental procedure

Using the instrumentation set-up described in chapter 2, strain measurements in x and y-directions were taken, while changing the following saw parameters:

i) axial prestress SIGO

ii) tilt angle of the top wheel

iii) position of the strain gages on the blade relative to the bandsaw

The change of the position of the strain gage location around the saw was achieved by rotating the blade by hand around the bandsaw. The chosen strain gage positions are shown in Figure 10 and are labelled from A to N. To calculate the stresses from the strain measurements Hooke’s law was used (TIMOSHENKO, "Theory of Plates" [24])

\[ \sigma_x = \frac{E}{1 - \nu^2} (\varepsilon_x + \nu \varepsilon_y) \]
FIGURE 10 STRAINAGE POSITIONS AROUND THE SAW
All measurements represented here only reflect strains and stresses which were imposed onto the blade after the straingages were applied. It is not possible from these measurements to determine the initial in-plane stress configuration of the blade (due to rolling, welding, straining etc. of the blade) which were present before the experiments were done. To measure strains on the blade accurately, the straingages had to be balanced before each experiment. This was done to minimize the common mode voltage of the straingage signal so that the whole range of the analog/digital converter in the NEFF 100 system could be used for the change of the straingage signal during an experiment. The straingages were balanced at position E (between the guides).

To be able to perform repeatable experiments, the initial conditions of the blade had to be identical for each experiment. It was observed that without any axial preload $2R_0$ the blade at position E would deflect in $y$-direction and the deflected shape would vary from experiment to experiment. This deflection induces bending strains in the straingages during the balancing of the straingages and results in an error of the straingage readings, once the blade is strained. To assure that identical conditions from experiment to experiment exist, the blade was prestrained with an axial preload of $2R_0=2700\text{N/mm}^2$ before balancing. This
procedure results in identical initial conditions during the balancing of the straingages for each experiment. While the average strain level of the blade between experiments is not effected by this prestraining force, it does reduce the slope of the measured strain distribution across the blade due to a tiltangle change. 

During the experiments it was found that the average strain measured on the outside of the blade was a factor of 1.3 higher than the corresponding prestrain level calculated from the loadcell readings from LC. The experiments were repeated with straingages mounted on the inside of the blade and it was found that the measured average strain on the inside of the blade was smaller by a factor of 1.1 than the corresponding prestrain level from LC. Experiments with straingages mounted on the inside of the blade could only be performed at position E (between the guides) and at position L (at the back of the saw). These results show that even with a preload of 2700 N during balancing of the straingages, there are still bending effects in the blade although to an observer the blade appears to be straight.
4.2 Stress-strain measurements with a stationary sawblade

In Figure 11 and 12 the strain levels on the outside of the blade in x and y-direction for different positions around the saw, averaged over the corresponding straingages have been plotted out. For position E and L the average strain levels on the inside of the blade have been added. The strain difference between the average strains at position E and L (straight blade) and the positions, where the blade is in bending over the wheels amounts to 1534 - 409 = 1125 microstrain. The theoretical strain value for bending of the blade over the wheels from chapter 3.2.2 results in a value of 1191 microstrain or a difference of 5.5%. These measurements have been repeated for tiltangles of 0.60° and 1.33°. The strain differences for bending over the wheels are 1168 microstrains and 1187 microstrain respectively and the differences between theoretical and experimental results are 1.9% and 0.3%.

The average stress-distribution in x-direction on the outside of the blade in Figure 13 is calculated from the strain data in Figure 11 and 12 and shows the change of axial stress the blade undergoes, while travelling over the saw wheels. As described in chapter 3 the
**Figure 11**

Average tensile strain variation in the x-direction around for a tilt angle $\theta = 0.06^\circ$. 

- $\Delta$ Blade outside
- $\Box$ Blade inside

Prestress: $\sigma(x) = 67.6 \text{N/mm}^2$

Restrain: $\text{MEPS}(x) = 322 \mu \varepsilon$
FIGURE 12  AVERAGE COMPRESSIVE STRAIN VARIATION IN Y-DIRECTION AROUND THE SAW FOR A TILT ANGLE Tα=0.06°
FIGURE 13
AVERAGE STRESS DISTRIBUTION IN X-DIRECTION AROUND THE SAW
FOR A TILT ANGLE OF TA=±0.06°
stress in the blade while passing over the wheels consists mainly of the sum of the stresses $\text{SIG}_0(x)$ and $\text{SIG}_b(x)$. The stress changes between position E and A amounts to $255\text{N/mm}^2$ which agrees well with the stress values for bending over the wheels from chapter 3.2.2 of $250\text{N/mm}^2$

Figure 14 shows the experimental results of strain changes on the outside of the blade while it travels away from the wheel-blade contacting zone for a tiltangle $TA=0.60^\circ$. Measurements of strains on the inside of the blade could not be performed without damaging the straingages. The geometric location of position C1 to C5 and X1 to X7 are shown in Figure 6. C3 represents the exact contact point between the blade and the bandsaw wheel. It can be observed that at position C5, which is 35mm away from C3, the strain in x-direction has almost reached a constant value which then raises slightly up to position D - the upper guide. Going from position C1 to position C3 the strain in x-direction decreases at a rate of 19 microstrain/mm. At the same time the strain in y-direction increases with a rate of 0.61 microstrain/mm. The stress values in Figure 15 are calculated from the strain data in Figure 14. While the stress in x-direction drops at a rate of $5\text{N/mm}^2$/mm while the blade travels from position C1 to C3, it reaches a minimum of $85\text{N/mm}^2$ at position X4
FIGURE 14

STRAIN VARIATION BETWEEN POSITION C AND D (BETWEEN CONTACT ZONE AND GUIDE), TILT ANGLE $\theta = 0.60^\circ$
FIGURE 15 AVERAGE STRESS VARIATION BETWEEN POSITION C AND D (BETWEEN CONTACT ZONE AND GUIDE), TILT ANGLE TA = 0.60°
compared to an axial prestress of 70N/mm².
The stress in y-direction approaches zero after the blade leaves the contact zone as one would expect to happen from a thin plate supported on two sides in x-direction and having free edges in the y-direction. At position E and L the experiments have been repeated 3 times for 4 different tiltangles at the front of the blade and at the back of the blade. The data from these experiments with their standard deviations and error bounds are shown in Table II.
The standard deviations are calculated for n=12 (three experiments for each tiltangle). Here are only the average strains for n=3 for each tiltangle shown. The error bounds are calculated for a confidence value of 95%.

Table II Average strain data at the outside and the inside of the blade at position E
4.3 Strain measurements across the blade at different locations around the saw for different axial prestresses and tiltangles

Figure 16, 17 and 18 show the stress-distribution across the blade for different tiltangles between position E (between the guides) and position C (contact zone between blade and saw wheel). Position C1 and C5 are 70 mm apart. The results show that in these 70 mm the blade undergoes the total average stress change of 250N.mm² due to bending over the wheel. For a blade velocity of c=40m/s (as used later in the dynamic experiments) this means that this stress change happens in 1.7ms. The change of the slope of the stress-distribution across the blade due to a tiltangle change can not be measured accurately, because by applying an axial preload of 2700N/mm² before balancing of the straingages, local stresses resulting from a tiltangle are reduced.

For a tiltangle of TA=0.06° the above measurements were repeated at position G (Figure 19). The slope of the stress-distribution across the blade at position G5 is 50% less than the slope at position C5 for the same conditions which shows that the influence of the tiltangle on the stress-distribution is greater at the top wheel than at the bottom wheel. This is due to the
FIGURE 16 STRESS IN X-DIRECTION ACROSS THE BLADE AT POSITION C (CONTACT ZONE), TILTANGLE $\alpha = 0.06^\circ$
STRESS IN X-DIRECTION ACROSS THE BLADE AT POSITION C (CONTACT ZONE), TILTANGLE $\alpha=0.6^\circ$
FIGURE 18  STRESS IN X-DIRECTION ACROSS THE BLADE AT POSITION C (CONTACT ZONE), TILT ANGLE TA=1.33°
FIGURE 19

STRESS IN X-DIRECTION ACROSS THE BLADE AT POSITION G (CONTACT ZONE), TILTANGLE TA=0.06°
FIGURE 20
STRESS IN X-DIRECTION ACROSS THE BLADE AT POS. A AND E
FOR INCREASING AX. PRESTRESSES SIG0(x), TILTANGLE TA=0.0°
fact that only the top wheel rotates around the z-axis during a tiltangle change, while the bottom wheel remains stationary.

A different series of experiments are shown in Figure 20. The axial prestress was raised in five steps. The levels of the axial prestresses were measured with the loadcell LC incorporated into the straining system while the stress changes in x-direction in the blade were measured with the straingage set-up on the outside of the blade. The experiments were performed at two different locations, at position E (between the guides) and at position A (on top of the top wheel). The objective of this experimental series is to examine the influence of prestress increases on the stress distribution at position E and A. The results show that the stress increase at position E is equal to the stress increase at position A. While the stress-distribution across the blade at position E is nearly linear, the stress-distribution at position A follows a parabolic curve with a maximum at straingage position 5 (one third across the blade away from the tooth gullet). A similar parabolic stress-distribution has been measured by PAHLITSCH [13] and is due to the fact that the blade is simultaneously bent over the wheels in two directions, in x-direction due to the curvature of the bandsaw wheel diameter and in y-direction due to the crown of the
wheel. The stress maximum across the wheel coincides with the geometric location of the highest peak of the crown of the wheel.

This parabolic stress-distribution at position A results in maximum stress values which are considerably higher than the average stress across the blade. For the tiltangle of $TA=0.00^\circ$ at position A the maximum stress at SG5 averaged over the five experiments is 65.6N/mm$^2$ or 18.5% higher than the axial stress averaged over all SG across the blade. The theoretical stress calculations in chapter 3 do not account for the fact that the stress distribution across the blade, while subject to an axial prestress and to bending follows a parabolic curve and has therefore a maximum value. If fatigue limit calculations for sawblades are performed this fact has to be taken into consideration.
5 DYNAMIC VIBRATION MEASUREMENTS WITH A STATIONARY BLADE

5.1 Experimental set up and theoretical bandsaw models

To measure the natural frequencies of the stationary blade in the cutting region (between the guides) an electro magnet was mounted on the inside of the sawblade at a position between the guides, two thirds of the span length L down from the top guide and away from the center of the blade over to the toothside to simulate the force influence of the cutting process. On the outside of the blade two non contacting displacement probes were situated at the same height as the electro magnet, one on the toothside, the other on the backside of the blade (see Figure 21a). The electro magnet was connected to a frequency generator which supplied a broad excitation to the sawblade, ranging from 0-200 Hz. The excitation force could be measured with a loadcell.

The excitation force as well as the response signals (deflections) of the blade were fed into a FFT dual channel frequency analyser which calculated and displayed the transfer function between input (excitation) into the blade and output (displacement) of the blade. Besides many other functions the RMS values of the input and output signal, the coherence (a measure
FIGURE 21a  POSITIONING OF INSTRUMENTATION FOR VIBRATION MEASUREMENTS

FIGURE 21b  DEFINITION OF THE TILTSTRESS-DIFFERENCE AND RELATED STRESSES
of the linear relationship between input and output data), and the transmissibility (ratio of RMS values between input and output data, in our case this is the inverse of the stiffness of the blade) were of special interest and could be calculated and displayed. These functions will be explained later in more detail by means of an example.

In Figure 22 the RMS values of the excitation force from the electro magnet and the blade response are recorded. The numerical values inside the graph frame (.234, .137, ...) represent the RMS values (log. Scale) for channel A and B over 10 Hz intervals around the natural frequencies. The ratio of the RMS values of channel B divided by channel A give the transmissibility of the blade (see Figure 27).

Figure 23 shows a typical transfer function from the frequency analyser display. A phase shift of 90 degrees and a maximum (peak) in the magnitude of the transfer function indicates a natural frequency. The phase and the magnitude are numerically displayed for a chosen frequency (61 Hz). The same information can be obtained if the real and imaginary parts of the transfer function are plotted. A sign change in the real part and a peak in the imaginary part indicate a natural frequency (Figure 24). Figure 25 shows the coherence function, an indication of the linear cause/effect
RMS SPECTRA FOR EXCITATION AND RESPONSE OF THE STATIONARY SAWBLADE
FIGURE 23: FREQUENCY TRANSFER FUNCTION OF THE STATIONARY SAWBLADE (MAGNETIC EXCITATION)

TRANSFER FUNCTION

PHASE (DEGREES)

TF

MAG (MM/N)

FREQUENCY (Hz)

200
FIGURE 24

REAL AND IMAGINARY PART OF THE TRANSFER FUNCTION

\( R_0^{\text{m}} = 28.0 \)
\( TSD = +15.7 \text{ N/mm}^2 \)
\( 61.00000 \text{ Hz} \)
\( -1.27+00 \text{ E} \)

REAL PART

IMAG. PART

FREQUENCY (Hz)

\( f_{L1}, f_{T1}, f_{L2}, f_{T2} \)
FIGURE 25

COHERENCE OF THE STATIONARY SAWBLADE

COHERENCE

COH

158.0000 HZ  521.03
61.0000 HZ

R₀*=28.0, TSD=15.7N/MM²

FREQUENCY (Hz)

0  200
TILTANGLE = 1.2°  
TSD = +15.7 N/MM²

SHIFT OF NATURAL FREQUENCIES FOR DIFFERENT AXIAL PRESTRESSES $R_0^*$

FIGURE 26
$R_0^* = 28.0$
$TSD = 15.7 \text{ N/mm}^2$

$61.00000 \text{ Hz}$

**FIGURE 27**

*Transmissibility of the sawblade at natural frequencies*
relationship between channel A and channel B. Linear transmission systems excited only by channel A yield a coherence factor of 1. Non linearities or inputs in addition to channel A produce coherent factors between 0 and 1. In Figure 26 the shift of the various lateral and torsional natural frequencies as well as the change of the transfer function magnitude due to an increase of the axial prestressing force $2R_0$ ($R_0^* = f(2R_0)$) can be seen.

To obtain general values independent of geometric factors typical for our bandsaw, the axial prestress $2R_0$ and the tiltangle $T_A$ were normalized.

To describe the influence of the axial prestress the normalization:

$$R_0^* = \sqrt{\frac{2R_0 \cdot L^2}{2 \cdot B \cdot D}}$$

was used (TIMOSHENKO 1959, "Theory of Plates and Shells" [24]). In case of the influence of the tiltangle a variety of possibilities were considered. Different authors on bandsaw vibrations have suggested various formulae to take the effect of changes of the tiltangle $T_A$ into consideration. ULSOY [25] defined the tiltstress ratio $T_S R$:

$$T_S R = (\text{SIG}7 / \text{SIG}1) - 1$$
SIG7 is stress at location of straingage 7 (toothside)
SIG1 is stress at location of straingage 1 (backside)

This yields a value of $TSR = 0$ if $SIG7 = SIG1$. KIRBACH [7] defines the tiltstressratio as ranging from 1.0 (no tilt) to 0, no tension stress at SG1. TANAKA [21] defines:

$$TSR = \frac{SIG7}{SIG1}$$

In contrast to Ulsoy's definition this yields the value of

$TSR = 1$ if $SIG7 = SIG1$.

A problem inherent to all these definitions is the fact that these tiltstress ratios are not only a function of the tiltangle and the geometric situation of the tilting system but also a function of the axial prestress. To find a definition which is only a function of the tiltangle and the individual characteristic of the tilting system the following formulae describing a tiltangle difference $TSD$ was used:

$$TSD = SIG1 - SIG7$$
$$TSD = MSIGO + SIGt - (MSIGO - SIGt) = 2SIGt$$
For a definition of the different stresses see Figure 21b. Because the TSD is a difference of two stress values as compared to TSR, it is not dimension free. The experimental relationship between TSD and TA for the tensioned blade used in these experiments is shown in Figure 28 with the corresponding error bounds for a confidence value of 95%.

The mathematical solution that predict the lateral natural frequencies in a stationary and in a moving bandsaw blade have been derived (as a function of such parameters as bandsaw velocity c, span length L, axial prestressing force 2R0, density and cross section area A of the saw blade) by various researchers. The solutions derived by MOTE [9] and KANAUCHI [21] are used to compare the experimental data obtained with the theoretical data. Mote's and Kanauchi's solutions are derived for a band in transverse vibrations with small amplitudes and simply supported boundary conditions at x=±L.

The solutions are:

Mote's flexible band solution

\[ f_L = \frac{b}{2L} \frac{R_0'}{\rho A} \left(\frac{(1-\kappa) \frac{\rho A c^2}{R_0}}{1+(1-\kappa) \frac{\rho A c^2}{R_0}}\right) \]
FIGURE 28  TILTSTRESS DIFFERENCE OF THE SAWBLADE AS A FUNCTION OF THE TILTANGLE

DATA RANGE FOR A CONFIDENCE RANGE OF 95%, n = 6

TILTSTRESS DIFFERENCE (N/mm²)

TILTANGLE (DEGREES)
with the definition of the wave velocity \( c_0 \)

\[
c_0 = \frac{R_0}{\rho A}
\]

we can write:

\[
f_L = \frac{b}{2L} \frac{c^2}{c_0^2} \left( \frac{1}{1+\left(1-\kappa\right)} \right) \frac{c_0^2}{c^2}
\]

The velocity dependent part of Motes flexible band solution will be examined in chapter 6 while here only the solution for \( c = 0 \) is of interest.

Kanauchi developed the following solution:

\[
f_L = \frac{b}{2L} \frac{R_0^2}{\rho A} \left(1-\frac{\rho A c^2}{R_0^2}\right) = \frac{b}{2L} \left(1-\frac{c^2}{c_0^2}\right) c_0
\]

In comparison to Kanauchi's solution, Mote defines and uses the value \( \kappa \) (see Appendix II) - a measure of the wheel support stiffness \((0<\kappa<1)\). \( \kappa = 0 \) when the wheel support is rigid and \( \kappa = 1 \) when the support is flexible (the wheel support stiffness of all production bandmills equipped with air or hydraulic strain systems is close or equal to 1). For a stationary sawblade \((c = 0)\) Kanauchi's and Mote's solutions are identical:

\[
f_L(c=0) = \frac{b}{2L} \frac{R_0}{\rho A} = \frac{b}{2L} c_0
\]
The experimental data for the torsional natural frequencies were compared with the results from the analytical solution developed by Alspaugh [2]:

\[ f_T = \frac{b}{2L} \left( 1 - \frac{c^2}{c_0^2} \right) c_0 \]

with \( c_0^2 = 4 \frac{H^2 G}{B^2 \rho} + \frac{\text{SIG0}(x)}{\rho} \)

He derived this formula for a thin rectangular strip moving at a constant speed in the x-direction. The strip is assumed to be simply supported torsionally at two lines at \( x=\pm L \) as for example, a band running between fixed rollers supports.

For the stationary sawblade the analytical solution for torsional natural frequencies is:

\[ f_T(c=0) = \frac{b}{2L} \left( 4 \frac{H^2 G \text{SIG}(x)}{B^2 \rho} + \frac{b}{\rho} \right) = \frac{c_0}{2L} \]

Experiments showed that in general coupled modes were observed. However, to the purpose of this work a mode shape in which the toothside and backside of the blade deflected into the same directions along the z-axis were called lateral natural frequencies and a mode shape in which the toothside and the backside of the blade deflected into opposite directions along the z-axis were called torsional natural frequencies.
5.2 Lateral and torsional natural frequencies of the blade between the guides

The two lowest lateral and torsional natural frequencies of the sawblade as a function of the axial prestress $R_0^*$ for different tiltstress differences are shown in Figure 29. These four frequencies have been plotted each by itself with expanded frequency scales for a better resolution in Figure 30 through 33. With increasing axial prestress $R_0^*$ all measured natural frequencies essentially increase linearly but with different slopes.

The theoretical results from Mote's Kanauchi's and Alspaugh's solutions are added to the experimental data in Figures 30 to 33. The absolute and relative errors between the analytical and experimental results for the four different natural frequencies are calculated from the formulae in APPENDIX IIIa and are summarized in Table III for two axial prestresses $R_0^*$ spanning the total range of the experimental axial prestress values and a tiltstress difference $TSD=-40.7\,\text{N/mm}^2$. This tiltstress difference corresponds to the tiltangle for the running sawblade. Examining the results in Table III it becomes evident that the lateral natural frequencies agree very well with the values predicted by theory for the whole measurement range of $R_0^*$. From the Figures 30
FIGURE 29  LOWEST LATERAL AND TORSIONAL NATURAL FREQUENCIES FOR DIFF.
AXIAL PRESTRESSES $R_0^*$ AND TILTSTRESS-DIFFERENCES TSD
FIGURE 30  LOWEST LATERAL NATURAL FREQUENCY $f_{L1}$ FOR DIFFERENT AXIAL PRESTRESSES $R_0^*$ AND TILTSTRESS-DIFFERENCES TSD
FIGURE 31  2nd LOWEST LATERAL NATURAL FREQUENCY $f_{L2}$ FOR DIFFERENT AXIAL PRESTRESSES $R_0^*$ AND TILTSTRESS-DIFFERENCES TSD
FIGURE 32
LOWEST TORSIONAL NATURAL FREQUENCY $f_{T1}$ FOR DIFFERENT AXIAL PRESTRESSES $R0^*$ AND TILTSTRESS-DIFFERENCES TSD

ALSPAAUGH

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FIGURE 33
2nd LOWEST TORSIONAL NATURAL FREQUENCY $f_{T2}$ FOR DIFFERENT AXIAL PRESTRESSES $R0^{*}$ AND TILTSTRESS-DIFFERENCES TSD
Table III Values for theoretical and experimental natural frequencies, and related absolute and relative errors

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<th>RO*</th>
<th>$f_{L1}$ theor. (Hz)</th>
<th>$f_{L1}$ exp. (Hz)</th>
<th>absolute error (Hz)</th>
<th>rel. error (%)</th>
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<th>$f_{T1}$ exp. (Hz)</th>
<th>absolute error (Hz)</th>
<th>rel. error (%)</th>
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<td>144.8</td>
<td>173.2</td>
<td>28.4</td>
<td>19.6</td>
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to 33 it can be seen that the calculated difference between theoretical and experimental values reaches a minimum for the tiltstress difference TSD=-40.7N/mm$^2$. For different tiltstress differences the deviation between theoretical and experimental values increases. Later in this chapter it will be seen that for a tiltstress difference of about $-40.7N/mm^2$ the lateral and torsional modes shapes are uncoupled, while for higher or lower tiltstress differences the modes are coupled.

The torsional natural frequencies of the blade are on the average 29.7% higher than the theoretical predicted values for lower axial prestresses ($R_0^*=23$). This error decreases for higher axial prestresses ($R_0^*=31$) on the average to 18.7%. Similar experiments by TANAKA [21] show that the torsional natural frequencies of a pretensioned blade are 9 to 17% higher than the comparable theoretical values, while the torsional natural frequencies for an untensioned blade show a good agreement with theoretical predicted frequency levels. A theoretical study by ULSOY [25] shows that the torsional natural frequencies rise linearly with the level of pretension in the blade (assuming a parabolic stress distribution in the blade). He calculates that for a parabolic stress distribution with a maximum stress value at the toothside and the backside of the blade of $\text{SIG}(x)^*$ and a minimum stress value at the center of the
blade of SIG(x)*/2, the lowest torsional frequency of the blade is 17Hz higher than for an untensioned blade with a uniform stress distribution across the blade. Because Alspaugh's theory does not consider different stress distribution across the blade, but assumes a uniform stress distribution the theoretical values for the torsional frequencies are lower than the experimental values for the pretensioned blade used in these experiments. A first order polynomial using the least square fitting method was fit to the experimental data and the natural frequencies were plotted as a function of the tiltstress difference TSD for various axial prestresses R0*. These results are shown in the Figures 34 through 38. Both lateral natural frequencies follow a parabolic curve with a maximum at the tiltstress difference TSD = -26.7 N/mm² while the torsional natural frequency fT1 has a minimum for the same tiltstress difference. Table IV shows that the influence of different tiltstress differences on the lateral natural frequencies is of greater magnitude and of opposite sign as its effect on the torsional natural frequencies. The absolute and relative changes of the natural frequencies are calculated according to the formulae in Appendix IIIb.
FIGURE 34  1st AND 2nd LOWEST LATERAL AND TORSIONAL NATURAL FREQUENCIES FOR DIFFERENT TILTSTRESS-DIFF, TSD AND AX. PRESTRESSES R0*
FIGURE 35
LOWEST LATERAL NATURAL FREQUENCY $f_{L1}$ AS A FUNCTION OF THE TILTSTRESS-DIFFERENCE TSD FOR AXIAL PRESTRESSES $R0^*$
FIGURE 36  2nd LOWEST LATERAL NATURAL FREQUENCY $f_{L2}$ AS A FUNCTION OF THE TILTSTRESS-DIFFERENCE TSD FOR AXIAL PRESTRESSES $R0^*$
FIGURE 37  LOWEST TORSIONAL NATURAL FREQUENCY $f_{T1}$ AS A FUNCTION OF THE TILTSTRESS-DIFFERENCE TSD FOR AXIAL PRESTRESSES $R0^*$
FIGURE 38  2nd LOWEST TORSIONAL NATURAL FREQUENCY $f_{T2}$ AS A FUNCTION OF THE TILTSTRESS-DIFFERENCE TSD FOR AXIAL PRESTRESSES $R0^*$
<table>
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<td></td>
<td></td>
<td>Δf₁ (Hz)</td>
<td>Δf₂ (Hz)</td>
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<tr>
<td>29.3</td>
<td>26.7 to 95.6</td>
<td>-4.8</td>
<td>-7.6</td>
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Table IV Change of the natural frequencies as a function of the tiltstress difference TSD
5.3 Transmissibility of the Stationary Sawblade

Beside the evaluation of the natural frequencies the associated mode shapes were of special interest. Therefore the transmissibility $T$ of the blade at the toothside and at the backside of the blade were measured and plotted. The experimental setup is shown in Figure 21a. The blade was excited by the electro magnets, while the blade response was measured with two non-contacting transducers, one located at the toothside of the blade, the other at the backside. The transmissibility represents the ratio of the RMS value of the response signal to the RMS value of the excitation signal. The reciprocal value of the transmissibility is the stiffness of the blade. Experiments measuring these RMS values right at the previously established natural frequencies gave inconsistent results, although data values were averaged from 32 readings and each data value was repeated 3 times. A second approach yielded more promising results. Around each natural frequency, RMS values were averaged over a 10 Hz interval with the natural frequency at the center of the band width and then 32 readings were averaged to yield one data value. This procedure was repeated three times for each data value and was then again averaged. The average error bounds for a confidence value of 95% for the
transmissibility data shown in this chapter were $+0.048\text{mm/N}$.

These experiments then had to be repeated 128 times to arrive at the transmissibility data shown in Figure 39 through to Figure 46. A first order polynomial was fit through the curves from Figure 39 - 46 and then the transmissibilities for $R_0^* = 26.0$ (midpoint of the experimental range) were used in the following discussion.

Figure 39 and 40 show the transmissibility of the stationary blade for $f_{L1}$ at the toothside and at the backside respectively as a function of the axial prestress for different tiltstress differences. While the change of the axial prestress has very little effect on the slope of the curves, an increase in the tiltstress difference greatly increases the transmissibility at the toothside from $0.15\text{mm/N}$ to $0.62\text{mm/N}$ or by 413% while having relatively little effect at the backside. A similar effect can be observed in Figure 41 and 42 for $f_{L2}$, only that the absolute change of the magnitude of the transmissibility at the toothside is smaller- from $0.03\text{mm/N}$ to $0.19\text{mm/N}$ or by 633%. While the transmissibility at $f_L$ increases for an increase of the tiltstress difference, the transmissibility at $f_T$ decreases for the same change of the tiltstress difference. In Figure 43 the torsional
FIGURE 39  TRANSMISSIBILITY OF THE STATIONARY SAWBLADE (MEASURED AT THE TOOTHSIDE FOR $F_{L_1}$) AS A FUNCTION OF $R0^*$ FOR DIFF. TSD
**FIGURE 40**

Transmissibility of the stationary sawblade (measured at the backside for \( f_{L1} \)) as a function of \( R_0^* \) for different TSD.
FIGURE 41  TRANSMISSIBILITY OF THE STATIONARY SAWBLADE (MEASURED AT THE TOOTHSIDE FOR $f_{L2}$) AS A FUNCTION OF $R0*$ FOR DIFF. TSD
FIGURE 42
TRANSMISSIBILITY OF THE STATIONARY SAWBLADE (MEASURED AT THE BACKSIDE FOR $f_{L2}$) AS A FUNCTION OF R0* FOR DIFF. TSD
FIGURE 43 TRANSMISSIBILITY OF THE STATIONARY SAWBLADE (MEASURED AT THE TOOTHSIDE FOR \( f_{T1} \)) AS A FUNCTION OF R0* FOR DIFF. TSD
Figure 44: Transmissibility of the stationary sawblade (measured at the backside, for $f_{T1}$) as a function of $R_0^*$ for diff. TSD.
Figure 45: Transmissibility of the stationary sawblade (measured at the backside for $f_{T_2}$) as a function of $R0^*$ for different TSD.
FIGURE 46  TRANSMISSIBILITY OF THE STATIONARY SAWBLADE (MEASURED AT THE TOOTH SIDE FOR $f_{T_2}$) AS A FUNCTION OF $R_0*$ FOR DIFF. TSD
<table>
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<th>backside TL1 (mm/N)</th>
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<tr>
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<td>R0*</td>
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<tr>
<td>-81.8</td>
<td>.18</td>
<td>.15</td>
</tr>
<tr>
<td>-47.9</td>
<td>.26</td>
<td>.26</td>
</tr>
<tr>
<td>-13.7</td>
<td>.37</td>
<td>.39</td>
</tr>
<tr>
<td>14.0</td>
<td>.58</td>
<td>.62</td>
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<td></td>
<td>TSD(_2) (N/mm(^2))</td>
<td>toothside TL2 (mm/N)</td>
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<td>R0*</td>
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<td>.14</td>
</tr>
<tr>
<td>14.0</td>
<td>.27</td>
<td>.19</td>
</tr>
<tr>
<td></td>
<td>TSD(_2) (N/mm(^2))</td>
<td>toothside TT2 (mm/N)</td>
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<td>R0*</td>
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<td>-13.7</td>
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<td>.08</td>
</tr>
<tr>
<td>14.0</td>
<td>.03</td>
<td>.05</td>
</tr>
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Table V  Transmissibility of the stationary sawblade at the two lowest lateral and torsional natural frequencies for different tiltstress differences and axial prestresses R0*
transmissibility for $f_{T1}$ at the toothside of the blade is lowered (from 0.74mm/N to 0.26mm/N or by 280%) with increasing tiltstress differences, while the transmissibility at the backside of the blade for $f_{T1}$ decreases from 0.14mm/N to 0.05mm/N or by 36% (Figure 44).

In Figure 45 and 46 the transmissibilities for the toothside and the backside at $f_{T2}$ are recorded. Similar to the transmissibilities for the lateral natural frequencies, the change of the amplitude of the transmissibility at $f_{T2}$ is much smaller than for $f_{T1}$. At the toothside the change of the transmissibility as a function of the tiltstress difference ranges from 0.14mm/N to 0.05mm/N or changes by 36%. At the backside the transmissibility for $f_{T2}$ changes from 0.08mm/N to 0.07mm/N or by 9%. While for $f_{L1}$, $f_{L2}$ and $f_{T2}$ an increase of the axial prestress only has a very little effect onto the transmissibility of the blade, at $f_{T1}$ the transmissibility at the toothside as well as at the backside raises considerably for a change of the axial prestress from $R_0^* = 23$ to $R_0^* = 29$. For a better comparison of the transmissibility values as a function of the axial prestress $R_0^*$ these values averaged over the tiltstress differences are shown in Table V.

In Figure 47 the modeshapes of the stationary sawblade at four natural frequencies for $R_0^* = 23$ and
Figure 47 Modeshapes of the stationary sawblade at the two lowest lateral and torsional natural frequencies
R0*=29 and for TSD=-81.8N/mm² and TSD=14.0N/mm² are shown. At the lowest lateral natural frequency the transmissibility at the toothside raises for the stated increase from TSD=-81.8N/mm² to 14.0N/mm² by 0.4mm/N for R0*=23 and by 0.53mm/N for R0*=29, while at the backside the transmissibility is lowered by 0.09mm/N and by 0.04mm/N for the corresponding changes of the axial prestress R0*.

At the second lowest lateral frequency the increase of the transmissibility for the reported change in the tiltstress difference amounts to 0.23mm/N for R0*=23 and to 0.10mm/N for R0*=29 at the toothside of the blade, while at the backside of the blade the transmissibility decreases for the change in the tiltstress difference by 0.01mm/N for R0*=23 and by 0.03mm/N for R0*=29.

These results show that the tiltstress difference has a far greater influence on the modeshapes for the lateral natural frequencies, than a change of the axial prestress.

At fT1 a change in the tiltstress difference from -81.8Nmm² to 14.0N/mm² reduces the transmissibility at the toothside of the blade by 0.56mm/N for R0*=23 and by 0.40mmN for R0*=29. At the backside the transmissibility decreases by 0.25mm/N for R0*=23 and by 0.19mm/N for R0*=29. The equivalent values for the decrease of the transmissibility at fT2 at the toothside amounts to
0.08mm/N for $R_0^*=23$ and to 0.11mm/N for $R_0^*=29$, while at the backside the transmissibility decreases by 0.03mm/N for $R_0^*=23$ and by 0.0mm/N for $R_0^*=29$.

From these results it can be concluded that an increase in the tiltstress difference has mainly an effect on the toothside of the blade. While the transmissibility at the toothside at the lateral natural frequencies increases, it decreases at the torsional natural frequencies. A change of the axial prestress has little effect on the transmissibilities at $f_{L1}$. At $f_{L2}$ a high tiltstress difference and an increase of the axial prestress lowers the transmissibility on the toothside considerably, while for a low tiltstress difference there are only small changes in the transmissibility.

At the torsional natural frequencies an increase of the tiltstress difference lowers the transmissibility at the toothside and at the backside. An increase of the axial prestress raises the transmissibility at the toothside and at the backside for a high tiltstress difference, while there is very little change at a low tiltstress difference.

Figure 47 also shows that the modeshapes for lateral and torsional natural frequencies are coupled. For a special tiltstress difference (between TSD=$-81.8N/mm^2$ and TSD=$14N/mm^2$) the modeshapes become uncoupled. Some other researchers have theoretically and
experimentally examined the lateral and torsional modeshapes of bandsaw blades. PAHLITSCH [16,17] did a study on modeshapes as a function of the blade thickness H, the free span length L, the axial prestress R0* and an edge load Fhor in y-direction acting on the teeth. SOLER [26] did a theoretical study on coupled modes as a function of an edge load Fhor. The author is not aware of any study of modeshapes - theoretical or experimental as a function of the tiltstress difference. Therefore no further analysis of the here presented modeshapes was done.
5.4 Dynamic vibration measurements with a moving blade

To measure the natural frequencies of a running blade the same experiments as in chapter 5.2 were repeated. A different blade was used in these experiments. It had no teeth so that the width \( B_0 = B = 260 \text{ mm} \), but had a similar stress distribution across the blade due to pretensioning compared to the blade used in the stationary experiments. The blade velocity \( c = 40.7 \text{ m/s} \) could not be varied. To compare the experimental data for the lateral natural frequencies with Mote's flexible band solution, the value of the support system stiffness \( K_s \) had to be evaluated (see APPENDIX II). The analytical results of Mote's, Kanauchi's and Alspaugh's solutions are added to the data plots in Figure 49 to 51, while the relevant data and error bounds are shown in Table V. For the range of the axial prestress \( R_0^* = 26 \) to \( R_0^* = 31 \) the analytical results from Mote and Kanauchi differ only by 0.13 Hz to 0.10 Hz or by 0.3% to 0.18%. In Figures 49 to 51 they are therefore represented only by one curve. The lateral natural frequencies are on the average lower than the theoretically predicted frequencies. For \( R_0^* = 26 \) the deviation amounts to 9.9% and for \( R_0^* = 31 \) it is 11.2%.

Similar to the stationary sawblade the torsional natural frequencies are considerably higher than the
FIGURE 48  NATURAL FREQUENCIES $f_L$ AND $f_T$ FOR THE RUNNING BLADE FOR DIFFERENT AXIAL PRESTRESSES $R_0^*$, $c = 40.7$ m/s
FIGURE 49  NATURAL LATERAL FREQUENCY $f_{L1}$ OF THE RUNNING BLADE FOR DIFFERENT AXIAL PRESTRESSES $R_0^*$, $\alpha = 40.7 \, m/s$
FIGURE 50  NATURAL LATERAL FREQUENCY $f_{L2}$ OF THE RUNNING SAWBLADE FOR DIFFERENT AXIAL PRESTRESSES $R0^*$, $c = 40.7$ m/s
FIGURE 51  NATURAL TORSIONAL FREQUENCY $f_{T1}$ OF THE RUNNING BLADE FOR DIFFERENT AXIAL PRESTRESSES $\varrho_0^*$, $c = 40.7 \text{ m/s}$
FIGURE 5.2 NATURAL TORSIONAL FREQUENCY $f_{T2}$ OF THE RUNNING BLADE FOR DIFFERENT AXIAL PRESTRESSES $R_0^*$, $c = 40.7 \text{ m/s}$
theoretically predicted frequencies. For $R_0^*=26$ the average difference amounts to 32.5% and for $R_0^*31$ it is 17.9%. These differences are due to the fact that -like for the stationary sawblade- Mote, Kanauchi and Alspaugh assume a constant stress distribution $R_0/A$ across the blade in their analytical solutions, while the blade used in these experiments was pretensioned which results in a parabolic stress distribution across the blade.
<table>
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<th>fT1 theor. (Hz)</th>
<th>exp. (Hz)</th>
<th>absolute error (Hz)</th>
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<th>exp. (Hz)</th>
<th>absolute error (Hz)</th>
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Table VI Values for theoretical and experimental natural frequencies and related errors for the running blade
CONCLUSIONS

Strain measurements for the stationary saw blade show that in the free span length stresses in x-direction increase proportionally with an increase of the axial prestressing force $R_0^*$. During bending over the wheels the measured stress in x-direction consists of the sum of the axial prestress and the stresses due to bending of the blade in x and y-direction (due to the wheel radius $r$ and the crown of the wheel). The combination of bending of the blade in two directions results in higher local stresses than a mere addition of the theoretical stresses suggests. This fact has to be considered in fatigue limit calculations.

An incremental increase of the axial preload $2R_0$ resulted in an equivalent increase of the stress in x-direction across the blade, independent of the position of the straingage location around the saw. The stress in the y-direction at the free span length, while the blade travels away from the wheels approaches zero.

Vibration measurements of the blade between the guides showed that the lateral and torsional natural frequencies increase with the axial prestress $R_0^*$. A change of the tiltstress difference resulted in a quadratic relationship for the natural frequencies with a maximum for $f_{L1}$ and $f_{L2}$ at $TSD = -26.7 \text{N/mm}^2$ and a
minimum at the same tiltstress difference for the lowest torsional natural frequencies. The slopes for both the lateral and torsional natural frequencies as a function of the tiltstress difference were similar so that no conclusion for an optimum tiltstress difference can be drawn from this.

A comparison of the experimental data with theoretical solutions derived by Mote [9], Kanauchi [21] and Alspaugh [2] showed that the theoretical predictions for the lateral natural frequencies agreed very well with these experimental results. The torsional natural frequencies were considerably higher than the predicted frequencies from Alspaugh's solution. This is due to the fact that Alspaugh assumes a constant stress distribution across the blade, while the blade which was used in these experiments was pretensioned. The pretensioning results in a parabolic stress distribution with higher tensile stresses at the edges of the blade.

Measurements of the transverse mode shapes of the stationary blade between the guides showed that a change of the axial prestress \( R_0^* \) has very little influence on the normalized deflections of the blade while for a change in the tiltstress difference between \(-81.8\,\text{N/mm}^2\) and \(14\,\text{N/mm}^2\), the maximum deflections at the lowest lateral natural frequency could be lowered by 413% and at the lowest torsional natural frequency by 280%. This
shows that at the natural frequencies a raise of the axial prestress R0* has very little effect, while trying to achieve an optimum stress distribution in the blade through a change of the stress distribution across the blade promises far better results.

Another interesting fact was that the mode shapes at the lateral and torsional natural frequencies consisted of coupled modes for most tiltstress differences.

The results for the running sawblade showed similar relationships and errors between the experimental and theoretical data values as observed for the stationary sawblade.
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APPENDIX I

Stress variation in x-direction across the blade as a function of the tiltangle $TA$

Due to the geometry of the tiltangle system a tilting of the top wheel results in a non uniform stress-distribution across the blade (see Figure 53a). This stress distribution can be divided into the sum of two stress-distributions, the uniform stress-distribution $SIGt0$ and the triangular stress-distribution $SIGta$. The loadcell $LC$, incorporated into the straining system measures the stress due to the axial prestressing force $2R0$ plus the stresses $SIGt0+SIGt$ due to the tiltangle. The stress difference between the straingage locations 1 and 7 due to any tiltangle is called $2*SIGt$. From Figure 53b we can derive the formula for the stress $SIGta$ for a tiltangle $TA$

$$\tan TA = \Delta L/Bs = TA/180^\circ, \quad \Delta L = \pi * TA * Bs/180^\circ$$

with $TA$ in degrees

and

$$SIGta = 2*SIGt = \varepsilon * E = \Delta L/Lw * E = \pi * yt^* E^* TA/(180^\circ * Lw)$$
The uniform part $\sigma_{t0}$ of the stress distribution for any tiltangle $TA$ can be calculated from:

$$\tan TA = \Delta L/(Bt-Bs/2) = \pi * TA/180^\circ$$

or

$$\Delta L = \pi*(Bt-Bs/2)*TA/180^\circ$$

and

$$\sigma_{t0} = \varepsilon E = E\Delta L/L = \pi*(Bt-Bs/2)*E*TA/(Lw*180^\circ)$$

The additional stress $\sigma_{tta}$ which is measured by the loadcell LC in the straining system due to a tiltangle consists of the sum:

$$\sigma_{tta} = \sigma_{t0} + \sigma_{t}/2 = \sigma_{t0} + \sigma_{t}$$
FIGURE 53a  NON UNIFORM STRESS DISTRIBUTION ACROSS THE BLADE AS A FUNCTION OF THE TILTANGLE TA

\[ \sigma g_t = \sigma g_{t0} + (B_t - B_s) \rho \]

FIGURE 53b  CHANGE OF STRAIN AS A FUNCTION OF THE TILTANGLE TA

\[ l_w' = l_w + \frac{(B_t - B_s)}{2} \]
APPENDIX II

Calculation of the wheel support stiffness $K_s$

In the publication "Some Dynamic Characteristics of Band Saws" C.D. MOTE [9] develops a mathematical model to describe the influence of the wheel support system stiffness $K_s$ on the lateral natural frequencies. He introduces a nondimensional factor which can be calculated from:

$$\eta = 1 - \kappa = \frac{1}{L_w K_s} \frac{1}{1 + \frac{2A^2 E}{2 E}}$$

Experiments with static loading of the top wheel support system showed that the wheel support stiffness $K_s = 825\text{N/mm}$. Substituting this value into the above formula yields:

$$\eta = 1 - \kappa = \frac{1}{2464\text{mm} \times 825\text{N/mm}} = 0.012$$

or $\kappa = 0.988$
APPENDIX III

a) Error calculation for stress data

absolute error:
\[ r_{abs} = \text{exp. Data value} - \text{theor. Data value} \]

relative error:
\[ r_{rel} = \frac{(\text{exp. Data value} - \text{theor. Data value}) \times 100}{\text{theor. Data value}} \]

b) Formula for calculation of the absolute and relative change of the natural frequencies as a function of the tiltstress-difference TSD:

\[ f_{abs} = f(TSD=95.6\text{N/mm}^2) - f(TSD=26.7\text{N/mm}^2) \]

\[ f_{rel} = \frac{f(TSD=95.6\text{N/mm}^2) - f(TSD=26.7\text{N/mm}^2)}{\frac{f(TSD=26.7\text{N/mm}^2) + f(TSD=95.6\text{N/mm}^2)}{2}} \]

\( f = \text{frequency value (Hz)} \)

c) Formula for the calculation of the relative error for straingage and loadcell data:

\[ r_{rel} = \frac{(SG - LC) \times 100}{LC} \]

\[ r_{rel} = \text{relative error} \]

SG = change of average stress across the blade from seven straingages SG

LC = change of axial prestress from loadcell LC

LC = absolute maximum axial prestress from LC
PROGRAM NEFF

THIS PROGRAM SCANS NEFF CHANNELS IN HANDSHAKE MODE. THE USER SELECTS THE NUMBER OF SAMPLES, THE SAMPLING RATE, THE NUMBER OF CHANNELS AND THEIR ADDRESSES AND GAINS. THESE PARAMETERS ARE CHANGED BY EDITING THE PARAMETER FILE 'PAR.DAT'.

NOTE: MAXIMUM CLOCK RATE IS 40000 HZ. FOR MULTI-CHANNEL SCANS, THE CLOCK RATE SHOULD BE LIMITED TO 22000 HZ.

THE MAXIMUM SAMPLE SIZE IS 4096.

THIS PROGRAM INCLUDES EXTERNAL SCAN START.

SYSTEM LIBRARY Routines

EXTERNAL WTQIO
EXTERNAL GETADR
EXTERNAL ASNLUN

DATA ARRAYS

DIMENSION LIST(4096)
DIMENSION IDAT(4096)
DIMENSION IBUFF(1)

QIO PARAMETER ARRAY

DIMENSION IPARM(6)

QIO STATUS ARRAY

DIMENSION ISTAT(2)

DINP STATUS ARRAY

DIMENSION ISOB(2)

BYTE NO, ANS, ANS2, YES
DATA NO/78/, YES/89/

ASSIGN NIO: TO LU 3

CALL ASNLUN(3,'NI',0)

GET STARTING AND ENDING LIST INDEX
IF NOT GOOD DO AGAIN

THE FOLLOWING ROUTINE TO ALLOW EXTERNAL SCAN INITIATION CAN BE INSERTED INTO THE PROGRAM BY REMOVING THE FOLLOWING 'GO TO' INSTRUCTION.

GO TO 595

DECIDE IF SCAN WILL BE STARTED EXTERNALLY.

WRITE(5,111)
111 FORMAT(1X,'WILL SCAN BE INITIATED EXTERNALLY? Y/N ',$)
READ(5,222) ANS

C CONTINUE
C
CALL ASSIGN(1,'PAR.DAT')
C
READ PARAMETERS FROM FILE 'PAR.DAT'.
C IWCT=SAMPLE SIZE PER CHANNEL.
C NCHAN=NUMBER OF CHANNELS.
C CLOCK=NEFF SAMPLING RATE PER CHANNEL.
C
READ(1,901) IWCT,NCHAN
901 FORMAT(2I5)
READ(1,551) CLOCK
551 FORMAT(F16.5)
READ(1,552) LIST(1), LIST(2)
552 FORMAT(26)
IF(NCHAN.EQ.1) GO TO 590
C FOR TWO OR MORE CHANNELS, USE SAMPLE/HOLD MODE.
C CLOCK RATE AND SCAN LIST SIZE ARE MULTIPLIED
C TO GIVE DESIRED SAMPLING RATE AND SAMPLE SIZE
C FOR EACH CHANNEL.
C
IWCT=IWCT*(NCHAN+1)
IF((IWCT.GT.4096).OR.(IUCT.LE.0)) GO TO 800
CLOCK=CLOCK*(NCHAN+1)
IF ((CLOCK.GT.40000.).OR.(CLOCK.LT.0)) GO TO 800
READ(1,552) (LIST(I), I=3,NCHAN+2)

DO 560 I=2,(IWCT-NCHAN+1)
   LIST(I+NCHAN+1)=LIST(I)
560 CONTINUE
GO TO 580
C FOR SINGLE CHANNEL, USE SAMPLE MODE ONLY.
C
DO 570 I=2,IWCT-1
   LIST(I+1)=LIST(I)
570 CONTINUE
580 CONTINUE
C SET PROGRAMMABLE CLOCK.
C
DWWL=1./CLOCK
HERTZ=1./XRATE(DWELL,IRATE,IPRSET,1)
WRITE(5,321) CLOCK, HERTZ
321 FORMAT(IX,' CLOCK= ',G15.5, ' HERTZ= 'G12.5)
CALL CLOCKB(IRATE,IPRSET,1,IND,1)
WRITE(5,1234) IND
1234 FORMAT(IX,'IND CODE= ',I3)
C
C RESET SERIES 500 BUS
C BYTE COUNT = 2
C IPARM 1 = IDATA ADDRESS
C FUNCTION = 1002 OCTAL
C
IPARM(2)=2
CALL DETADR(IPARM(1),IDAT)
CALL WTO(IO('1002,3,10,,ISTAT,IPARM,IDS)
C
C PRINT COMPLETION MESSAGE
WRITE(5,905)
FORMAT(1X,*1X,'SERIES 500 BUS RESET!'/)
WRITE(5,906) ISTAT(1),ISTAT(2),IDS
FORMAT(1X,*DRIVER COMPLETION CODE = ',06,' (OCTAL)'*/
* 1X,'LAST RESPONSE = ',06,' (OCTAL)'*/
* 1X,'DIRECTIVE STATUS = ',06,' (OCTAL)'*/

WRITE DATA TO RAM, READ BACK AND CHECK

CONVERT WORDS TO BYTES
IPARM(3)=RAM STARTING ADDRESS
IPARM(1)=LIST ADDRESS
FUNCTION CODE=400 OCTAL

IPARM(2)=IWCT*2
IPARM(3)=1
CALL GETADR(IPARM(1),LIST(1))
CALL WTC10("400,3,10*,ISTAT,IPARM,IDS)

READ DATA BACK FROM RAM INTO CORRESPONDING LOCATIONS IN ARRAY IDAT
IPARM(1)=IDAT ADDRESS
IPARM(2) AND IPARM(3) UNCHANGED FROM ABOVE
FUNCTION CODE=1000 OCTAL

CALL GETADR(IPARM(1),IDAT(1))
CALL WTC10("1000,3,10*,ISTAT,IPARM,IDS)

PRINT ANY ERRORS
IERR=0
DO 400 I=1,IWCT
IF(IDAT(I),EQ.LIST(I)) GO TO 400
IERR=IERR+1
WRITE(5,920) LIST(I),IDAT(I)
920 FORMAT(1X,'RAM ERROR - OUTPUT = ',05,' ; READ BACK = ',05,/) 400 CONTINUE

PRINT ERROR COUNT

WRITE(5,921) IERR
FORMAT(1X,'WRITE TO RAM AND READ BACK COMPLETE',14,' ERRORS ',/) WRITE(5,906) ISTAT(1),ISTAT(2),IDS

CONTINUE

TO ACTIVATE THE EXTERNAL SCAN START OPTION,
REMOVE THE FOLLOWING 'GO TO'.

GO TO 345

IF (ANS.EQ.NO) GO TO 345

EXTERNAL START

WAIT FOR SIGNAL TO START SCAN.

CALL DINP(0,0,105B,INPUT)
IF (INPUT.EQ.0) GO TO 10
GO TO 346

CONTINUE
MANUAL START

WRITE(5,654)
654 FORMAT(1X,'TO START SCAN, ENTER RETURN',$/)
READ(5,222) CR
346 CONTINUE

EXECUTE FROM RAM HANDSHAKE

FUNCTION CODE=3001 OCTAL
IPARM(1)=IDAT ADDRESS
IPARM(2)=BYTE COUNT
IPARM(3)=RAM STARTING ADDRESS

CALL GETADR(IPARM(1),IDAT(1))
IPARM(2)=IWCT*2
IPARM(3)=1
CALL WTQIO('30011-3,10,,ISTAT,IPARM,IDS)

REPEAT LAST FEW DATA POINTS UNTIL THE TOTAL NUMBER OF DATA POINTS EQUALS 'IWCT', A POWER OF 2.

DO 888 I=IWCT,IWCT+NCHAN+1
   IDAT(I)=IDAT(I-NCHAN-1)
888 CONTINUE

WRITE(5,922)
922 FORMAT(IX,'EXECUTE FROM RAM IN HANDSHAKE MODE',$/)

STORE DATA IN USER SELECTED FILES.

DO 450 J=3,NCHAN+2
   CALL FILES(J)
   IF(NCHAN.EQ.1) GO TO 444
   WRITE(2,930) INT(FLOAT(IWCT)/FLOAT(NCHAN+1)),HERTZ/FLOAT(NCHAN+1)
   WRITE(2,935) (FLOAT(IDAT(I))/32768.,I=J,IWCT+NCHAN+1)
444 WRITE(2,930) IWCT,HERTZ
   WRITE(2,935) (FLOAT(IDAT(I))/32768.,I=J,IWCT+NCHAN+1)
445 CONTINUE

CALL CLOSE(2)
450 CONTINUE

WRITE(5,906) ISTAT(1),ISTAT(2),IDS
WRITE(5,924)
924 FORMAT(IX,'FINISH EXECUTING FROM RAM',$/)
GO TO 802

THE FOLLOWING OPTION ALLOWS SCAN TO BE REPEATED USING THE SCAN LIST STORED IN NEFF RAM.

802 WRITE(5,112)
112 FORMAT(1X,'DO YOU WISH TO REPEAT SCAN? ',$/)
READ(5,211) ANS2
211 FORMAT(A5)
   IF(ANS2.EQ.YES) GO TO 354
   GO TO 804
800 WRITE(5,902)
902 FORMAT(1X,'***** INVALID DATA *****',$/)
804 STOP
END
SUBROUTINE FILES(J) ALLOWS USER TO CHOOSE FILE NAMES FOR DATA STORAGE.

SUBROUTINE FILES(J)
BYTE BUF(80)

WRITE(5,140) J-2
FORMAT(1X,'ENTER CHANNEL(I,) FILE NAME ',I)
READ(5,150) (BUF(I),I=1,80)

150 FORMAT(80A1)
L=LENGTH(BUF,80)
BUF(L+1)=0
CALL ASSIGN(2,BUF)
RETURN

END

FUNCTION LENGTH FINDS LENGTH OF ALPHANUMERIC DATA STRING. THE FIRST 'BLANK' CHARACTER INDICATES THE END OF THE STRING.

INTEGER FUNCTION LENGTH(BUF,N)
BYTE BUF(1),BL
INTEGER N
DATA BL/32/
DO 100 I=N,1,-1
IF(BUF(I),NE.BL) GO TO 200
CONTINUE
200 LENGTH=I
RETURN
END
"STRAIN" reads strain gage data for one loadcell and 9 strain gages from files. It calculates the axial preload, the strains and stresses due to loadcell readings, and strain and stress mean values due to 9 single strain gage readings.

'SG' means strain gage, 'LC' means loadcell, 'SIG' means stress.

INTEGER I, J, EXCT, EXPNO, N, M, K
REAL CALLC, CALSGX, CALSGY, TWORO(5:1:10), A, YOUNG, HERTZ, TILTRA
REAL POS, EPS(5:1:10), SIG(5:1:10), DATA(5:1:10)
REAL EPSX(5:1:10), SIGY(5:1:10), SIGX(5:1:10)
REAL MSIGSG(10), MSIGLC(10), MEPSLC(10), MEPSSB(10), M2RO(10), NUM

BYTE IBUF(80)

ASSIGN PRINTER TTS TO LABEL '1'
CALL ASSIGN(1, 'TTS')
CALLC = 1.9264E5
CALSGX = 166.32
CALSGY = 415.8
YOUNG = 210000.0
A = 398.7

EXPNO IS AMOUNT OF EXPERIMENTS DONE
WRITE (5, 71)
71 FORMAT (IX, 'ENTER AMOUNT OF EXPERIMENTS DONE: ', I4)
READ (5, 72) EXPNO
72 FORMAT (I4)
M IS #OF EXPERIMENTS, N IS #OF STRAINGAGES USED, K IS NUMBER OF SAMPLES BEEN TAKEN
DO 81 M = 1, EXPNO
81 WRITE (5, 37) M
37 FORMAT (IX, 'EXPERIMENT NUMBER: ', I4)
DO 91 N = 1, 10
91 READ (2, 212) DATA(M, N, K)
90 READ DATA FROM FILE SG(M, N), #0 OF SCANS IWCT AND SCANNING FREQUENCY HERTZ
WRITE (5, 700)
700 FORMAT (IX, 'ENTER FILENAME, WHERE STRAINGAGE DATA ARE STORED: ', *'')
READ (5, 701) IBUF(J), J = 1, 80
701 FORMAT (80A1)
L = LENGTH (IBUF, 80)
DO 91 N = 1, 10
91 IBUF(L+1) = 59
IF (N.GE.8) GOTO 150
IBUF(L+2) = N+48
IBUF(L+3) = 0
GOTO 111
150 IBUF(L+2) = 49
IBUF(L+3) = N-8448
111 CALL ASSIGN (2, IBUF)
CALL ASSIGN (2, IBUF)
READ (2, 112) IWCT, HERTZ
112 FORMAT (I5, F16.5)
READ DATA FROM FILE SG(M, N) SIGNALVALUES
DO 211 K = 1, (IWCT-1)
211 FORMAT (6E13.5)
DO 213 K = 1, (IWCT-1)
213 CONTINUE
CALL CLOSE(2)
CONTINUE
DO 214 K = 1, (IWCT-1)
214 CONTINUE
NOW ALL DATA FOR M EXPERIMENTS ARE READ INTO IWCT, HERTZ AND DATA(M, N, K)
DO 215 K = 1, (IWCT-1)
215 CONTINUE
CALCULATE ABS. VALUES: TWORO, STRAIN AND STRESS FOR LOADCELL-READINGS

DO 58 N=1,10
   DATA(1,N,K)=(DATA(1,N,K)+DATA(2,N,K))/2
58 CONTINUE

DO 311 M=3,EXPNO
   DATA(M,1,K)=DATA(M,1,K)-DATA(1,1,K)
   TWORO(M,1,K)=DATA(M,1,K)*(CALLC/0.835+8588.7)*1.19
   EPS(M,1,K)=TWORO(M,1,K)/(2*A*YOUNG)
   SIG(M,1,K)=EPS(M,1,K)*YOUNG
311 CONTINUE

CALCULATE VALUES FOR SGX: ABS., STRAINSGX

DO 511 M=3,EXPNO
   DO 411 N=2,8
      DATA(M,N,K)=DATA(M,N,K)-DATA(1,N,K)
      EPS(M,N,K)=DATA(M,N,K)/CALSGX
411 CONTINUE
511 CONTINUE

CALCULATE STRESS SGX

DO 811 N=2,8
   DO 911 M=3,EXPNO
      SIG(M,N,K)=230769.2*(EPS(M,N,K)+0.15*(EPS(M,9,K)+EPS(M,10,K))
911 CONTINUE
811 CONTINUE

CALCULATE STRESS SGY

DO 912 M=3,EXPNO
   DO 913 N=2,8
      EPSX(M,1,K)=EPSX(M,1,K)+EPS(M,N,K)
      EPSY(M,1,K)=EPSY(M,1,K)+EPS(M,N,K)*0.3/7.0+0.5*(EPS(M,9,K)+EPS(M,10,K))
      SIGY(M,1,K)=EPSX(M,1,K)*0.3/7.0+0.5*(EPS(M,9,K)+EPS(M,10,K))
912 CONTINUE
913 CONTINUE

CALCULATE STRESS SGY

DO 921 M=3,EXPNO
   DO 922 N=2,8
      SIGX(M,1,K)=SIGX(M,1,K)+SIG(M,N,K)
      SIGY(M,1,K)=SIGY(M,1,K)+SIG(M,N,K)
922 CONTINUE
921 CONTINUE

CALCULATE AVERAGE LOADCELL-STRESS "MSTRESSLC FOR ALL EXP.

DO 924 M=3,EXPNO
   MSIGLC(K)=MSIGLC(K)+SIG(M,1,K)
924 CONTINUE

CALCULATE AVERAGE STRESS DISTRIBUTION FROM STRAINAGE 2-8 ACROSS THE BLADE

DO 925 M=3,EXPNO
   SIGX(M,1,K)=SIGX(M,1,K)+SIG(M,N,K)
   SIGY(M,1,K)=SIGY(M,1,K)+SIG(M,N,K)
925 CONTINUE
MEPSLC(K) = MEPSLC(K) + EPS(M, 1, K)
MEPSSG(K) = MEPSSG(K) + EPSX(M, 1, K) / 7.0

CONTINUE
MEPSLC(K) = MEPSLC(K) / (EXPNO - 2) * 1.0E6
MEPSSG(K) = MEPSSG(K) / (EXPNO - 2) * 1.0E6

CONTINUE

DO 928 M = 3, EXPNO
   TILTRA = TILTRA + SIG(M, 2, 1) / SIG(M, 8, 1)
928 CONTINUE
TILTRA = TILTRA / (EXPNO - 2)

C ************* ONLY VALUES FOR K=1 ARE BEING PRINTED, BUT ALL VALUES FOR K=1 TO
   (IWCT-1) ARE STORE IN ARRAYS DATA, EPS, SIG
C
K=1
C WRITE ALL DATA ON SCREEN AND PRINTER
C FIRST MULTIPLY ALL STRAINS BY 1.0E6 TO GET MICROSTRAIN UNITS
DO 51 M = 3, EXPNO
   DO 52 N = 1, 10
      EPS(M, N, K) = EPS(M, N, K) * 1.0E6
   52 CONTINUE
51 CONTINUE
WRITE (1, 9)
WRITE (5, 9)
9 FORMAT ('1')
C WRITE ON SCREEN: BLADEPOSITION AND PRINT ON PAPER
WRITE (5, 11)
11 FORMAT (1X, 'ENTER BLADEPOSITION (A TO X):', 6X, $)
   READ (5, 12) POS
12 FORMAT (A3)
C WRITE AVER. AXIAL PRELOAD FROM LC FOR M EXP.
   WRITE (5, 14) M2R0(K)
14 FORMAT (1X, 'AVERAGE AXIAL PRELOAD FROM LOADCELL FOR M EXPERIMENT
   M2R0=', F12.2, ' CMICROSTRAIN')
WRITE (1, 14) M2R0(K)
C WRITE AVER. AXIAL PRESTRESS FROM LOADCELL
   WRITE (5, 15) MSIGLC(K)
15 FORMAT (1X, 'AVERAGE AXIAL PRESTRESS FROM LOADCELL
   MSIGLC=', F12.2, ' CN/MM²')
WRITE (1, 15) MSIGLC(K)
C WRITE AVER. AXIAL STRESS FROM SG2-SG10 FOR M EXPERIMENTS
   WRITE (5, 16) MSIGSG(K)
16 FORMAT (1X, 'AVERAGE AXIAL STRESS FROM SG2-SG10 FOR M EXPERIMENTS
   MSIGSG=', F12.2, ' CN/MM²')
WRITE (1, 16) MSIGSG(K)
C WRITE AVER. AXIAL PRESTRAIN FROM LOADCELL FOR M EXPERIMENTS
   WRITE (5, 19) MEPSLC(K)
19 FORMAT (1X, 'AVERAGE AXIAL PRESTRAIN FROM LOADCELL', 18X,
   'MEPSLC=', F12.2, ' CMICR0STRAIN')
WRITE (1, 19) MEPSLC(K)
C WRITE AVER. AXIAL STRAIN FROM SG2-SG8
   WRITE (5, 22) MEPSSG(K)
22 FORMAT (1X, 'AVERAGE AXIAL STRAIN FROM SG2-SG8', 23X,
   'MEPSSG=', F12.2, ' CMICR0STRAIN')
WRITE (1, 22) MEPSSG(K)
C WRITE TILTSTRESSRATIO FOR M EXPERIMENTS
   WRITE (5, 24) TILTRA
24 FORMAT (1X, 'TILTSTRESSRATIO FOR M EXPERIMENTS =', F4.2)
WRITE (1, 24) TILTRA
WRITE(1,17)
WRITE(1,17)
WRITE(1,17)
WRITE(5,17)
WRITE(5,17)
WRITE(5,17)
WRITE(5,17)
WRITE(18)

17 FORMAT("
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*DYN* READS STRAINAGE-DATA FOR ONE LOADCELL AND 9 STRAINAGES FROM FILES. IT CALCULATES THE AXIAL PRELOAD TWO RO(N), THE STRAINS AND STRESSES DUE TO LOADCELL READINGS; STRESSES AND MEAN VALUES DUE TO 9 SINGLE STRAINAGE READINGS. *SG* MEANS STRAINAGE, *LC* MEANS LOADCELL, *SIG* MEANS STRESS EPS MEANS STRAIN; STRESS VALUES FOR LC

AN DS G2-SG8 CAN BE STORED AT FILE LOCATIONS.

MAXIMUM VALUE FOR IWCT (NO OF SAMPLES) =512!!

INTEGER I,J,EXPNO,M,K,POS,NCHAN
REAL CALLC,CALSGX,CALSGY,TO RO(1,1:513),YOUNG,HERTZ,TILTRA
VIRTUAL EPS(1:9,513), SIG(1,8,513),DATA(3,9,513)
VIRTUAL EPSX(1:1,513),SIGXY(1,1,513),SIGX(1,1,513)
REAL MSIG(513),MSIGLC(1),MEPSLC(513),MEPSSG(513),M2RO(1),NUM_BYTE
IBUF(80)

CALL ASSIGN TT5 TO LABEL *1"

CALL ASSIGN(IF' TT5')
CALLC=1.9264E5
CALSGX=166.32
CALSGY = 415.8
YOUNG=210000.0
A=398.7

M IS #OF EXPERIMENTS, N IS #OF STRAINAGES USED, K IS NUMBER OF SAMPLES BEEN TAKEN
NCHAN IS NUMBER OF CHANNELS BEING SCANNED
WRITE (5,71)
71 FORMAT (1X,'ENTER #OF CHANNELS BEING SCANNED:',$)
READ (5,72) NCHAN
72 FORMAT (14)
DO 81 M=1,NCHAN
81 WRITE (5,37) M
37 FORMAT (IX.'EXPERIMENT NUMBER',I4)
READ DATA FROM FILE SG(M,N), # OF SCANNS IWCT AND SCANNING FREQUENCY HERTZ
WRITE (5,700)
700 FORMAT (1X, 'ENTER FILENAME, WHERE STRAINAGE DATA ARE STORED:',$)
701 READ (5,701) (IBUF(J),J=1,80)
701 FORMAT (80A1)
L=LENGTH (IBUF,80)
DO 91 N=1,NCHAN
91 IBUF (L+1)=59
IF (N.GE.8) GOTO 150
IBUF(L+2)=N+48
IBUF(L+3)=0
GOTO 111
150 IBUF(L+2)=49
IBUF(L+3)=N-8+48
IBUF(L+4)=0
111 CALL ASSIGN (2,IBUF)
112 CALL ASSIGN (2,IBUF)
READ (2,112) IWCT,HERTZ
112 FORMAT (I5,F16.5)
READ DATA FROM FILE SG(M,N) SIGNALVALUES
READ (2,212) (DATA(M,N,K),K=1,IWCT)
212 FORMAT (6E13.5)
CALL CLOSE(2)
91 CONTINUE
81 CONTINUE
NOW ALL DATA FOR M EXPERIMENTS ARE READ INTO IWCT, HERTZ AND DATA(M,N,K)
DO 213 K=1,IWCT
213 CALL ASSIGN (2,IBUF)
C
C NOW ALL DATA FOR M EXPERIMENTS ARE READ INTO IWCT, HERTZ AND DATA(M,N,K)
DO 213 K=1,IWCT
C
C C
C C NOW ALL DATA FOR M EXPERIMENTS ARE READ INTO IWCT, HERTZ AND DATA(M,N,K)
DO 213 K=1,IWCT
C
C
DATA(1,N,K) = (DATA(1,N,K) + DATA(2,N,K)) / 2

CONTINUE

DATA(3,1,K) = DATA(3,1,K) - DATA(1,1,K)
TWO(1,1,K) = DATA(3,1,K) * CALLC/0.835 + 0588.7 * 1.19
EPS(1,1,K) = TWO(1,1,K) / (2*A*Y0UNG)
SIG(1,1,K) = EPS(1,1,K) * Y0UNG

CONTINUE

IF (NCHAN, EQ. 1) GOTO 900

C CALCULATE VALUES FOR SGX: ABS STRAIN SGX
DO 411 N = 2, (NCHAN-1)
   DATA(3,N,K) = DATA(3,N,K) - DATA(1,N,K)
   EPS(1,N,K) = DATA(3,N,K) / CALSGX
CONTINUE

C CALCULATE
DATA(3,NCHAN,K) = DATA(3,NCHAN,K) - DATA(1,NCHAN,K)
EPS(1,NCHAN,K) = DATA(3,NCHAN,K) / CALSGY

CONTINUE

C CALCULATE STRESS SGX
DO 411 N = 2, (NCHAN-1)
   SIG(1,N,K) = DATA(3,N,K) / CALSGX
CONTINUE

C CALCULATE STRESS SGY
DO 811 N = 2, (NCHAN-1)
   SIGY(1,1,K) = 230769.2 * (EPS(1,1,K) + 0.30 * (EPS(1,NCHAN,K)))
CONTINUE

C CALCULATE THE AVERAGE FOR TWO(0) FOR ALL M EXP.
C CALCULATE THE AVERAGE STRESS DISTRIBUTION FROM STRAINAGE 2-8
C * ACROSS THE BLADE
DO 913 N = 2, (NCHAN-1)
   SIG(1,1,K) = SIG(1,1,K) + SIG(1,N,K)
CONTINUE

C CALCULATE AVERAGE STRAIN FOR LC AND SGX ACROSS THE BLADE FOR ALL
C * M EXPERIMENTS IN MICROSTRAIN
MEPSLC(K) = MEPSLC(K) + EPS(1,1,K)
MEPSGG(K) = MEPSGG(K) + EPSX(1,1,K) / (NCHAN-2)
MEPSLC(K) = MEPSLC(K) / 1.0E6
MEPSGG(K) = MEPSGG(K) / 1.0E6

CONTINUE

C TILTRA = TILTRA + SIG(1,2,1) / SIG(1,9,1)

C ***********************************************
C ONLY VALUES FOR K=1 ARE BEING PRINTED, BUT ALL VALUES FOR K=1 TO
C * (IWCT-1) ARE STORE IN ARRAYS DATA, EPS, SIG
C WRITE ALL DATA ON SCREEN AND PRINTER
WRITE(1,9)
WRITE(5,9)
C FORMAT('1')
C WRITE ON SCREEN: BLADEPOSITION AND PRINT ON PAPER
900 WRITE (5,11)
11 FORMAT(1X,'ENTER BLADEPOSITION (A TO X):',A2)
READ(5,12) POS
12 FORMAT(A3)
WRITE(1,13) POS
WRITE(5,13) POS
13 FORMAT(1X,'BLADE POSITION1',A2)
WRITE(5,17)
WRITE (1, 17)
C  WRITE AVERAGE AXIAL PRELOAD FROM LOADCELL FOR M EXPERIMENT
DO 16 K = 1, IWCT
   M2R0(1) = M2R0(1) + TW0R0(1, 1, K)
   MSIGLC(1) = MSIGLC(1) + SIG(1, 1, K)
16  CONTINUE
   M2R0(1) = M2R0(1) / IWCT
   MSIGLC(1) = MSIGLC(1) / IWCT
WRITE (5, 14) M2R0(1)
WRITE (1, 14) M2R0(1)
14  FORMAT (1X, 'AVERAGE AXIAL PRELOAD FROM LOADCELL FOR M EXPERIMENT
         *S, M2R0 = ', F12.2, ' [N]')
WRITE (5, 15) MSIGLC(1)
WRITE (1, 15) MSIGLC(1)
15  FORMAT (1X, 'AVERAGE AXIAL PRESTRESS FROM LOADCELL
         * MSIGLC = ', F12.2, ' [N/MM^2]')
IF (NCHAN .EQ. 1) GOTO 901
WRITE (5, 17)
WRITE (1, 17)
WRITE (5, 24) TILTRA
WRITE (1, 24) TILTRA
24  FORMAT (1X, 'TILTSTRESSRATIO FOR M EXPERIMENTS = ', F4.2)
C  ASSIGN FILENAME TO OUTPUT FILE FOR STRESS DATA LC
901  WRITE (5, 57)
WRITE (1, 57)
57  FORMAT (1X, 'ENTER FILENAME, WHERE LC STRESS VALUE
         *SHALL BE STORED: ', *)
CALL FILES
WRITE (4, 58) (SIG(1, 1, K), K = 1, IWCT)
58  FORMAT (6F9.2)
CALL CLOSE (4)
IF (NCHAN .EQ. 1) GOTO 902
C  ASSIGN FILENAME TO OUTPUT FILE FOR STRESS DATA SG2-SG8
C  WRITE SG STRESS VALUES FROM SG1 - SG(NCHAN-1) TO FILE
DO 50 N = 2, (NCHAN-1)
   WRITE (5, 51) N
51  FORMAT (1X, 'ENTER FILENAME, WHERE STRESS VALUE FROM SG', I1, '
         *SHALL BE STORED: ', *)
   CALL FILES
   WRITE (4, 53) (SIG(1, N, K), K = 1, IWCT)
53  FORMAT (6F9.2)
   CALL CLOSE (4)
50  CONTINUE
C  WRITE AVERAGE STRESS FOR ALL (NCHAN-1) SG INTO FILE
C  WRITE (5, 56)
56  FORMAT (1X, 'ENTER FILENAME, WHERE AVERAGE SG STRESS- VALUES
         *SHALL BE STORED: ', *)
   CALL FILES
   WRITE (4, 54) (MSIGSG(K), K = 1, IWCT)
54  FORMAT (6F9.2)
   CALL CLOSE (4)
STOP
END
SUBROUTINE FILES
BYTE IBUF (80)
READ (5, 52) (IBUF(J), J = 1, 80)
52  FORMAT (80A1)
   L = LENGTH(IBUF, 80)
   IBUF(L+1) = 46
   IBUF(L+2) = 68
   IBUF(L+3) = 65
   IBUF(L+4) = 84
   IBUF(L+5) = 0
   CALL ASSIGN (4, IBUF)
INTEGER FUNCTION LENGTH(BUF, N)
BYTE BUF(1)
INTEGER N
BYTE BL
DATA BL/32/

DO 10 I=N,1,-1
   IF(BUF(I) .NE. BL ) GO TO 20
10 CONTINUE
LEN=
RETURN
END
"FREQ" ENTERS DATA FOR A TIME SCALE INTO A FILE
CALLED "FREQ.DAT". THE DATA REPRESENT THE TIME AXIS IN A X-Y PLOT
AND ARE NEEDED TO PLOT DATA ACQUISITIONED WITH THE "NEFF"
PROGRAMM. THE TIME AXIS DATA ARE INCREMENTS OF THE RATIO
IWCT/HERTZ = (SAMPLES TAKEN/SCANNING FREQUENCY). IWCT AND Hertz
DEPEN ON THE EQUIVALENT VALUES IN "PAR.DAT".

REAL HERTZ, TIME(512)
INTEGER IWCT,N
WRITE (5,'(1X,'ENTER #OF SAMPLES BEING TAKEN (IWCT FROM "PAR.DAT"):
*','$)')
READ (5) IWCT
WRITE (5,'(1X,'ENTER SCANNING FREQUENCY (HERTZ FROM "PAR.DAT")
* DO NOT FORGET THE !!DECIMAL-POINT!! :',*)
READ (5) HERTZ
DO 15 N=1,IWCT
  TIME(N)=TIME(N-1)+1/HERTZ
15 CONTINUE
CALL ASSIGN (2,'FREQ.DAT')
WRITE (2,'(6F12.8)') (TIME(N),N=1,IWCT)
CALL CLOSE (2)
STOP
END

"bye"

HAVE A GOOD AFTERNOON
27-SEP-82 12:43 TT5: LOGGED OFF

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