CONTROL AND ASSESSMENT FOR LINEAR CONTACT UNDER THERMO-MECHANICAL LOAD

Fl. Tudose- Sandu-Ville, L.C. Hanganu, E. Murăraşu
Technical University „Gh.Asachi" of Iaşi

INTRODUCTION

The increase of the industrial products quality is possible through reliability rising.

For machine parts located in enclosure, where premature deteriorations are difficult in recording, for non-stop program of work, under important thermo-mechanical contact loads and machine parts realized with high prices through modern (high) technologies, reliability of such machine parts is very important to be at optimum size.

Global thermo-mechanical wear is present at the contact of two surfaces under normal mechanical load and thermal tide, with poor lubrication and water cooling, with a relative motion of rolling (combine or not) with small slidering.

This complex phenomenon is an atypical combination of four fundamental types of wear (abrasive, adhesive, corrosive wear and contact fatigue), coexisting and interacting during the process, having a predominant position one besides others determined by the specific conditions of contact.

Global thermo-mechanical wear is like a competition between different factors, specific to contact mechanics, having a conjugate result in the destruction of contact layer. To achieve an evaluating method for the lifetime of linear contacts working under mechano-thermal solicitations, before exploitation and to find conditions for such contacts to realize all their potential lasting in exploitation, appears to become important for competitive industrial products.

1. THEORETICAL BACKGROUND

In the matter of evaluating lifetime prediction for a contact under mechanical load, the Ioannides-Harris method [4] is important.

Based on their investigations, Ioannides and Harris (1985) published the following equation:

$$\ln\frac{1}{\Delta S}\approx\frac{N^{e}(\tau-\tau_{l})^{c}\cdot\Delta V}{z^{h}} \qquad (1)$$

S = probability of survival for the stressed volume V,
N = number of stress cycles endured,
e = Weibull slope of life distribution,
z = depth at which stress τ occurs,
τ_l = endurance limit shear stress,
c, h = exponents depend upon exponents e and p (type of contact) which are determined empirically.

If $\tau < \tau_l$ in any incremental volume, that volume will not experience fatigue failure. Only those incremental volumes where $\tau < \tau_l$ can fail in fatigue, not the entire volume V. This means that for sufficiently low loads, such contacts will not fail in fatigue. For a mechanical load under this threshold the contact fatigue failure (pitting) doesn't appear [4] - figure 1.

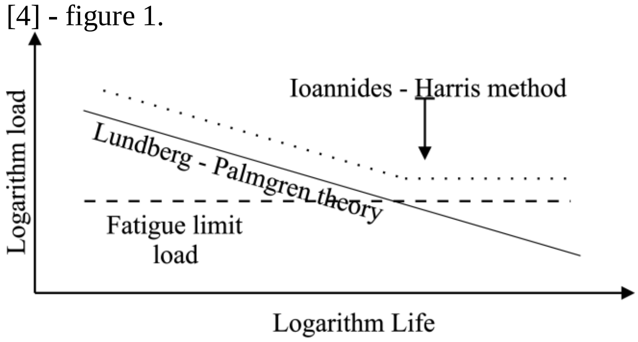


Figure 1.

Harris et al (1992) subsequently showed that the stress term in equation (1) could be expanded as follows:

$$\tau-\tau_{l}=\tau+\tau_{ech}-(\tau_{l}-\tau_{ech.r}) \qquad (2)$$

In equation (2) τ_{ech} is the equivalent shear stress caused by hoop stresses due to press fitting and high speeding rotation, and $\tau_{ech\cdot r}$ is the equivalent shear stress caused by residual stress due to processing (carburizing for example) [3].

Taking this observation into account, to achieve an evaluating method for the lifetime of linear contacts working under thermo-mechanical solicitations an equivalent stress is to be considered [6].

The decisive stress calculation after the equivalent stress from Huber-Mises-Henckey theory was taken into account [6]:

$$\sigma_{EMT}(\lambda)=\frac{1}{\sqrt{2}}\left(\sigma_{xMT}-\sigma_{yMT}\right)^2+$$
$$+\frac{1}{\sqrt{2}}\left(\sigma_{yMT}-\sigma_{zMT}\right)^2+\frac{1}{\sqrt{2}}\left(\sigma_{zMT}-\sigma_{xMT}\right)^2+$$
$$+\frac{1}{\sqrt{2}}6\lambda^2\left(\tau^2_{xyMT}+\tau^2_{yzMT}\right)+\frac{1}{\sqrt{2}}6\tau^2_{xzMT} \quad (3)$$

Where EMT means equivalent mechanical, thermal, λ is a coefficient defined as:

$$\lambda=\frac{(\tau_{ON})_f}{(\tau_{-1N})_f}=\frac{(\tau_{ON})_t}{(\tau_{-1N})_t} \quad (4)$$

Where $(\tau_{ON})_f$ and $(\tau_{ON})_t$ are shear and respectively torsion fatigue limit stresses for N pulsating cycles, $(\tau\text{-}1N)_f$ and $(\tau\text{-}1N)_t$ are shear and respectively torsion alternating fatigue limit stresses for N symmetrically alternating cycles.

The mechanical components in $\sigma_{EMT}(\lambda)$, equation (3), are the same with the mechanical stresses accepted in mechanical contacts for a linear contact between two cylindrical bodies on generation line, under a radial load [2].

When the cooling of the test rollers is made at the inner diameter, thermo elastic stresses appear for the theoretical case of a cylinder placed in a field of variable temperature [5]. These stresses in polar coordination are:

$$\sigma_{rr}=\frac{E\alpha}{2(1-\mu)}\left(1-\frac{R_1^2}{r^2}\right)\left[T(R_2,t)-T(r,t)\right] \quad (5)$$

$$\sigma_{\varphi\varphi}=\frac{E\alpha}{2(1-\mu)}\left[\left(1+\frac{R_1^2}{r^2}\right)\cdot T(R_2,t)+\left(1-\frac{R_1^2}{r^2}\right)T(r,t)\right]-$$
$$-2T(r,t) \quad (6)$$

In equation (6):
R_1 – inner roller radius
R_2 – outer roller radius
T (r, t) variable temperature, depending on γ value, $R_1<r<R_2$ and measurement time t,α, E, μ rollers material constants [5].

Returning to equation (1), the new expression is:

$$\ln\frac{1}{\Delta S}\approx\frac{N^e\left(\sigma_{EMT}(\lambda)-\sigma_{\lim}\right)^c\cdot\Delta V}{z^h} \quad (7)$$

,with the same semnifications for components.

The result of thermo-mechanical contact fatigue towards the contact layer is pitting and cracking. The surfaces with pitting and cracking fields determine an important level of noise and vibration in exploitation. The set up of a vibration level above an standard one (due top mechanical load only) and to a number of cycles N under the classic cycles number N_0 ($N<N_0$), shows the influence of variable thermo elastic stresses in the failure of the contact.

2. TEST RIG

The macrogeometrical mechano-thermical contact fatigue is experimental researched in the case of linear, almost dry contact with radial mechanical loading. Two samples are tested in the same time, named cooled rollers. These rollers are in permanent rolling contact with a warmed roller [8].

The test rig is presented in figure 2 and is composed by the following subassemblies:

A. Mechanical motion subassemblies (1, 2, 3, 4, 5, 6, 7) (electro engine, reduction rotation, couplings and bearings) which give an equal and constant rotation for the three rollers.

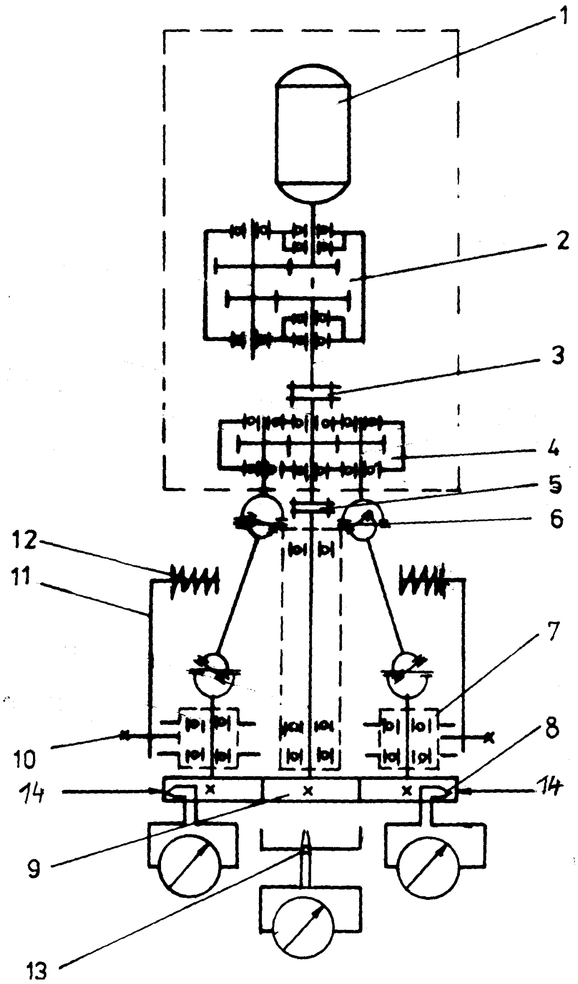
B. Mechanical loading subassemblies (10, 11, 12) (screw, screw-nut, calibrated spring). The mechanical loading is made upon the whole bearing, which is mechanically guided. The rolling bearings are not supplementary loaded.

C. Thermal measurement device (8).

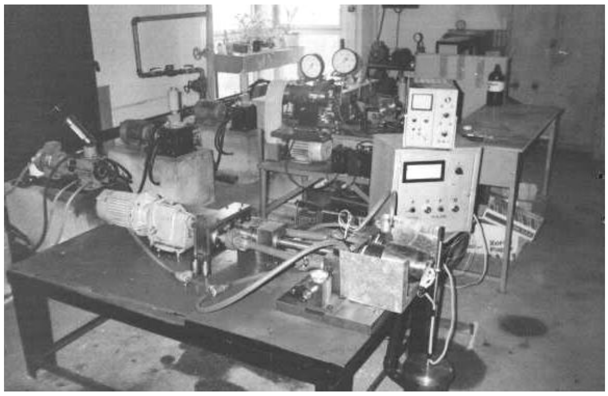
D. Warming device (13). The thermal tide is concentrated only on the warmed roller.

E. Cooling device (water) with poor flow (9) on the frontal/cylindrical surfaces of the cooled rollers.

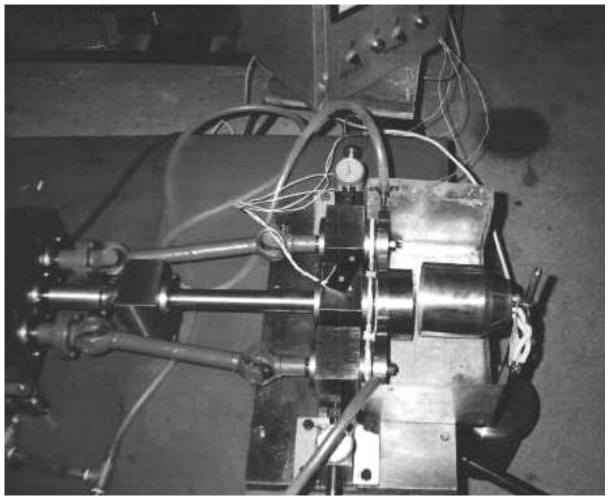
F. Vibration measurement level device.



Test rig general schema



General view of test rig



Driving supply of test rig

Figure 2.

4. VIRTUAL MEASUREMENT INSTRUMENT FOR CONTROL AND ASSESEMENT

In LabView program a virtual measurement instrument was performed and is presented in figure 3.

This instrument is used to measure temperature, speed and vibration level of cooled rollers.

The vibration level indicates the destroyed contact layer stage.

CONCLUSIONS

The variable thermo elastic stresses induced by a thermal tide determine the faster occurrence of a cracking and pitting field in a contact layer.

The surfaces with thermo-mechanical contact failure present an important level of noise and vibration in exploitation.

The control and measurement of vibration and noise levels indicate with accuracy the moment of thermo-mechanical contact fatigue set up.

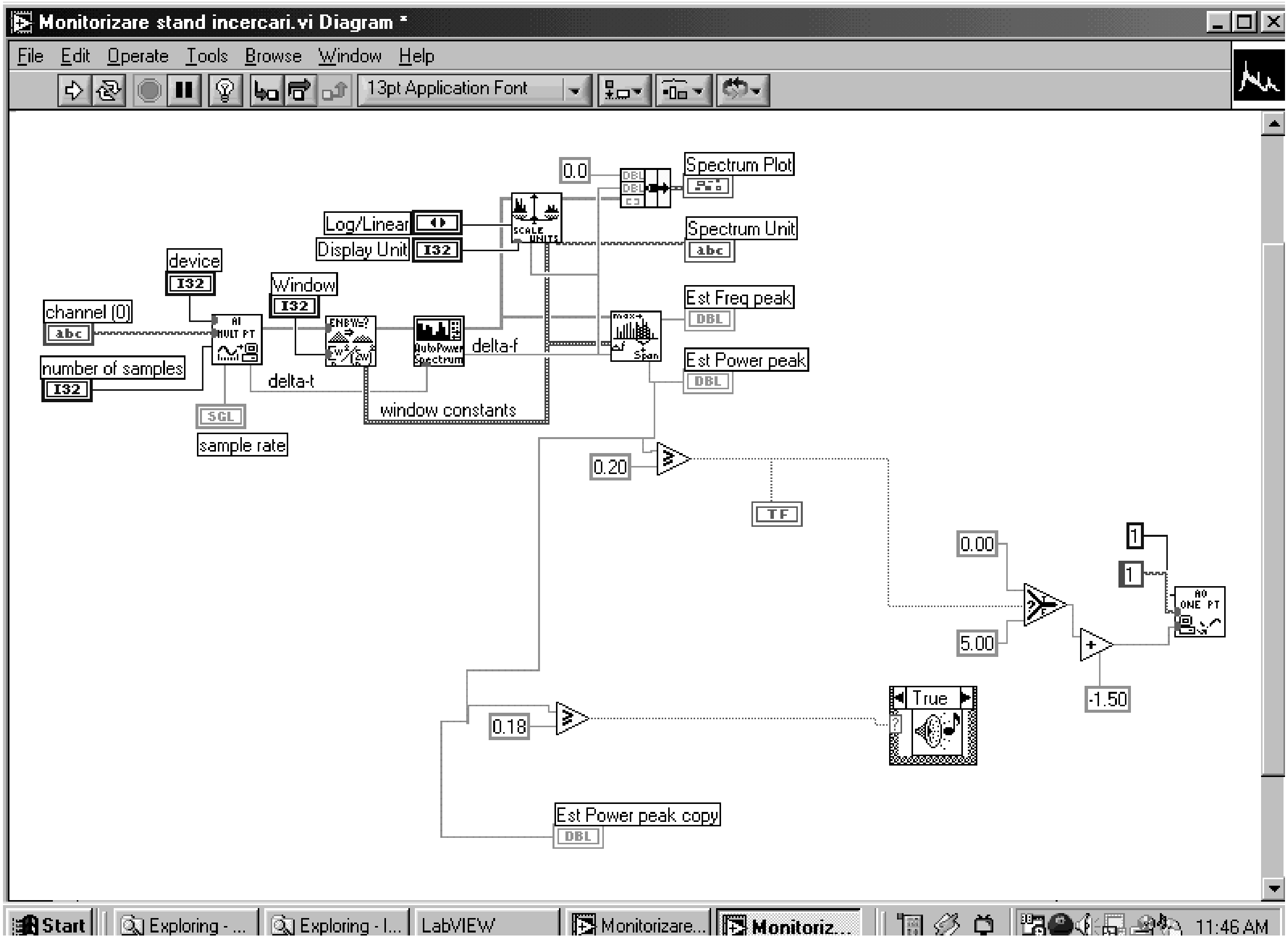


Figure 3.

The control for linear contact under thermo-mechanical load by noise and vibration device offers results for a statistical evaluation of thermo-mechanical contact running time.

The established of an evaluation method for a preexploitation reliability of thermo-mechanically-loaded contacts is an efficient technical and economical solution.

Bibliography

1. ***Cottet, F., Ciobanu, O.*** Bazele programării in LAB VIEW" *1998, Edit. Matrix, Bucureşti, România.*

2. ***Creţu, S. S.*** *Mecanica contactului, vol. I, 2002, Editura "Gh. Asachi" Iaşi.*

3. *Harris, T. A., McCool, J. I.* *On the Accuracy of Rolling Bearing Fatigue Life Prediction. April 1996, Journal of Tribology, vol. 118/297.*

4. ***Ioannides, E., Harris T. A.*** *A New Fatigue Life Model For Rolling Bearings", Trans. of ASME, Journal of Tribology, 107, p. 367-378, 1985.*

5. Popa B., Madarasan T., Bataga N., Adameşteanu I. *Solicitări termice în construcţia de maşini, Editura Tehnică, Bucureşti, 1978.*

6. Popinceanu, N. G., Diaconescu E., Creţu S. S. *"Critical stresses in Rolling Contact Fatigue", Wear, 71, p. 265-282, 1981.*

7. ***Racocea C., Tudose-Sandu-Ville O. F.****, "Statistical Evaluation of Rolling Mille Rollers Life Time", Technical University "Gh. Asachi", Journal, Iaşi, tom XL, fasc. 1-4, section 5, 1994.*

8. Tudose-Sandu-Ville O. F., Tărăboanţă G. F*. "Experimental Research on Thermo-Mechanical Contact Fatigue", The Annals of the "Dunărea de Jos" University of Galaţi, fasc. VIII, Tribology p. 104-107, 2002.*

9. **** "National Instruments. Users Handbook".*