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DEVELOPING FATIGUE TEST MACHINE FOR COMPOSITE MATERIAL

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ABSTRACT

This paper's goal is to introduce the third step of the EFOP-3.6.1-16-2016-00014. project on the Faculty of Engineering, University of Szeged. In this period the production technology of composite material was chosen and a fatigue test machine was developed and tested. The paper shortly describes the composite materials and summarizes the theory of fatigue than it presents the process of the development with several prototypes of fatigue test machine, some of which were manufactured and tested. Initially a shaker played the key role in the first two conceptions and finally a crank mechanism became as the best solution. The main solved problems during the development were selection of bearings and solving the partly dynamic balancing of the moving parts.

Keywords: plastic composites, fatigue test machine, dynamic balancing

1. INTRODUCTION (FIRST LEVEL HEADINGS: TNR 10PT, CAPITAL LETTERS, JUSTIFIED)

Only 20% of a presently used cargo aircraft's take-off weight is the payload, approximately half of the rest 80% is the empty weight. Therefore it is evident that the less empty weight, the better for the economy. Due to this fact there is a huge demand in the aircraft industry for light structural materials with advanced parameters. The composite materials are one of the most promising materials from that perspective. Due to the scientific literature the advantages of composites: significant weight reduction, anisotropy, incorrodible, weatherproof, long lifetime, resistance against fatigue, vibration damping, structural severity, formal freedom, aesthetic outfit, retentivity, reasonable heat-expansion, low "tooling" costs, reduced radar wave reflection, low maintenance and operation costs. The disadvantages of the composites: high process costs, low shock and damage resistance (carbon-, graphite-, boron-fibre), tendency to static charge, low lightning-endurance, lower interlayer strength, weak surface press strength, complicated and expensive repair costs, moisture intake, distortion, deformation at overheating, significant differences between tensile and press stress [1-3].

In our opinion the resistance against fatigue needs some more investigations. The composite materials are composed of at least two macroscopically and functionally separable materials in a definite structure. The base materials of the composite technology: reinforcing materials (frame materials, reinforcing fibres), matrix materials (bedding materials), core materials, adhesives, bulking agents, additives. For the purposes of the project the most important ingredients are the reinforcing and the matrix materials. The reinforcing materials are one of the main component parts of the composite materials that ensure the mechanical strength and rigidity of the structure. The matrix materials create the mechanical connection between the separate reinforcing fibres, take the awakening shear stresses, preventing their movement to the others, ensuring the structural integrity. The other parts are not relevant for our investigations [1-3].

In this project the composite material made of woven carbon fibres in epoxy resin produced by vacuum-infusion procession was determined to investigate for fatigue with different number of reinforcing layers. Some of the self-produced specimens were tested only by tensile machine, the others are going to be tested first by fatigue test machine with 10 million loads, and after that comes the tensile test. The difference between the two tensile tests will give details to the resistance against fatigue theory.

Fatigue is one of the most frequent causes for operational failure when the breakage of the material occurs due to fluctuating loads even the occurring stress is far below the yield point. The Wöhler diagram of composite materials is similar to the metal fatigue: it represents the resistance of the tested material to

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cyclic loads. The diagram shows the fact that the material strength reduces with repeated application of load and inversely proportional to the number of cycles applied. The strength value of the previously determined number of cycles is commonly taken as the fatigue limit. In some cases this is a “true” fatigue limit, when the stress value is below the limit, no fatigue mechanism occurs. But in most cases it is only valid till the operation reaches the selected number of cycles. [4-5].

2. DISCUSSION – DEVELOPING THE FATIGUE TEST MACHINE

Initially an extant Brüel & Kjaer LDS V200 shaker had the key role in the ideas of constructions. The axial loading was the obvious solution (Fig. 1.) where the shaker could prepare repeated normal stress in one direction in the centerline of the sections.

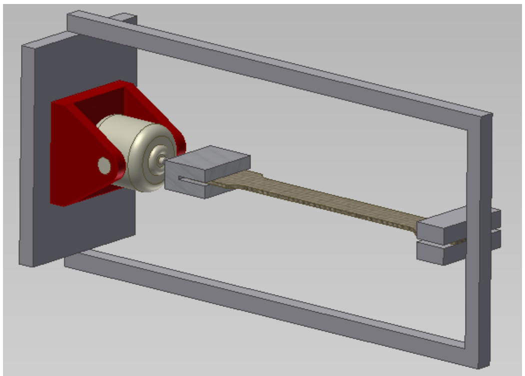

Fig. 1. Model of the tensile fatigue test machine with shaker

As the maximum force of the shaker is only 17,8 N, the tensile stress was not enough high for the specimens which sizes were determined earlier for the tests. So the concept turned to the direction of bending where a smaller force can be compensated with longer arm of force to achieve higher stress. As a specimen with rectangle section with base b and height v is loaded by a force F parallel with v, going through the centroid of the section, with a distance l from the fixed support, than the maximum bending moment at the fixed support [6]:

$$M = lF \tag{1}$$

The second moment of a cross section b x v to the main inertia axis x:

$$I_{xs} = \frac{bv^3}{12} \tag{2}$$

Accordingly the maximum stress in the section of the fixed support:

$$\sigma_{max} = \frac{M}{I_{xs}}\frac{v}{2} = \frac{12lF}{bv^3}\frac{v}{2} = \frac{6lF}{bv^2} \tag{3}$$

The maximum deflection (where E is the Young-modulus):

$$y = \frac{Fl^3}{3I_{xs}E} \tag{4}$$

from which expressing the force F and with the substitution of (2):

$$F = \frac{3I_{xs}Ey}{l^3} = \frac{bv^3Ey}{4l^3} \tag{5}$$

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The maximum stress based on (5) and (3) equations:

$$\sigma_{max} = \frac{6l}{bv^2}\frac{bv^3Ey}{4l^3} = \frac{3}{2}\frac{vEy}{l^2} \tag{6}$$

As the Young-modulus can be expressed from the previously achieved tensile diagrams, the bending of specimen was really encouraging (Fig. 2.).

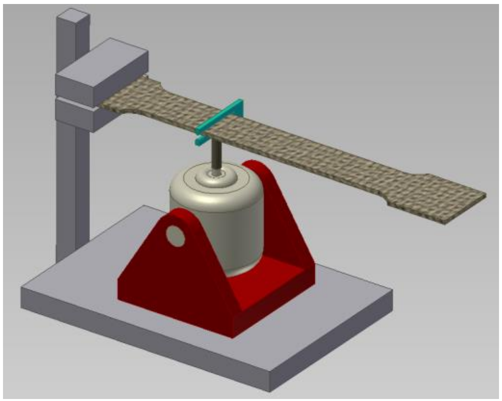

Fig. 2. Model of bending fatigue testing machine

However the maximum amplitude of the shaker is 5 mm that is far than enough to keep the validation of the small deflection equations but not enough to reach the limit stress of fatigue. So the usage of the shaker had to be thrown out and instead of it the rotational movement came into prominence. This was solved easily by a single-phase asynchronous motor controlled by frequency converter. At the first version the horizontally fixed specimen was bended uni-directionally by a pin mounted on a rotating disc (Fig. 3.).

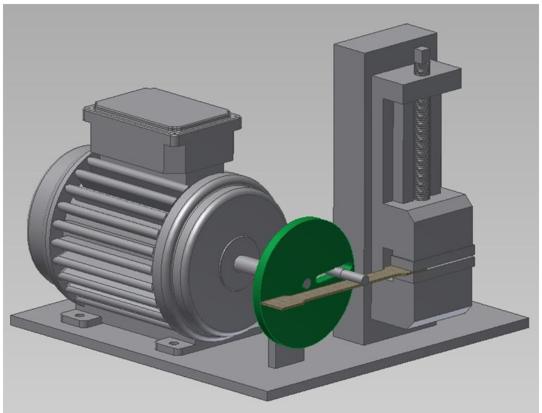

Fig. 3. Uni-directionally bended specimen by rotating disc with pin.

As the construction was extra simple to manufacture the test measures were shortly executed. It was observed that the arrival of the pin to the surface of the specimen is too dynamic, shocking-like, that would probably lead the specimen to go broke earlier than the fatigue would do it. Furthermore the one side bending was not considered effective enough, the symmetric, both directional bending and loading came to the fore.

This was the way how the bending fatigue test machine with crank mechanism appeared of which the first model is shown on Fig. 4.

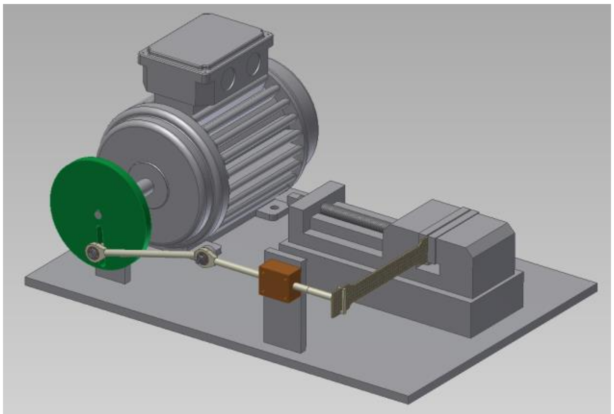

Fig. 4. First model of the bending fatigue test machine with crank mechanism

The main problem in that period of development was choosing the proper, obtainable bearings. Only commercially available bearings, housings and coupling rods were planned to apply. The accelerations due to the relatively high frequency (10 Hz) cause significant bearing forces despite of the small masses. For the self-aligning radial bearings at the rod ends a heavy-duty rod end with integral self-aligning, double row ball bearing, for supporting the horizontally moving coupling rod a sliding bearing with sinterbronze bearing bush was chosen. Due to the lubricant pressed into the porous material during production the maximum sliding speed of this kind of sliding bearing is 10 m/s that is far above our needs. Conversely, two pieces of 08.12.20 (8 mm inner, 12 mm outer diameter, 20 mm length) bush were applied for the horizontal support, and the housing was manufactured at the Faculty. The bushes were controlled against surface pressure by the loading model of Fig.7.

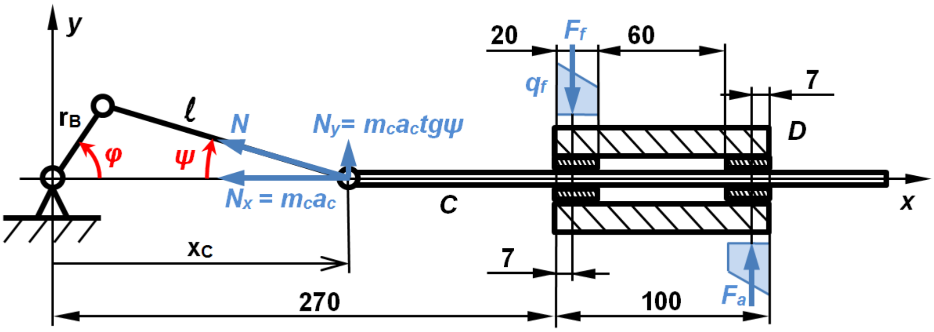

Fig. 7. Loading model of the bushes

At the first model with crank mechanism the previously used disk was taken over, in which there was only one radial slot for setting the bending deflection's size of the specimen. But the accelerations of the crank mechanism were so high that a partly dynamic balancing of the masses [7-8] was neccessary, and the best location of it was on the disc (Fig. 8.).

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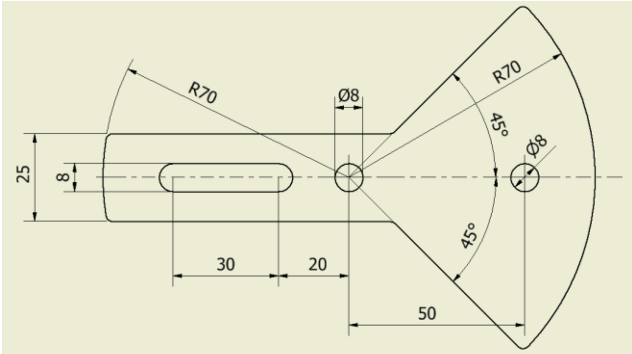

Fig. 8. Disk of the crank mechanism optimized for partly dynamic load [10]

The disk can be divided to two parts (see Fig. 8.): the right one consists of a circular sector with 90° central angle, the other part is the rest of the disc, named stick. The sector's centre of mass measured from the middle point of the circle in case of r=70mm, α=90° (without the hole):

$$r_e = \frac{2}{3}\frac{r\,sin^{\alpha}/_2}{^{\alpha}/_2} = \frac{2}{3}\frac{70mm\,\frac{\sqrt{2}}{2}}{\frac{\pi}{4}} = \frac{4}{3}\frac{70mm\,\sqrt{2}}{\pi} = 42\,mm \tag{7}$$

The mass of the quarter circle if the width of the disc is v=10 mm:

$$m_e = \frac{r^2\pi}{4}v\rho = \frac{(0{,}07m)^2\pi}{4}0{,}01m\;7850\frac{kg}{m^3} = 0{,}30\,kg \tag{8}$$

The kinetic model of the crank mechanism with point-like masses without extension is introduced by Fig. 9.:

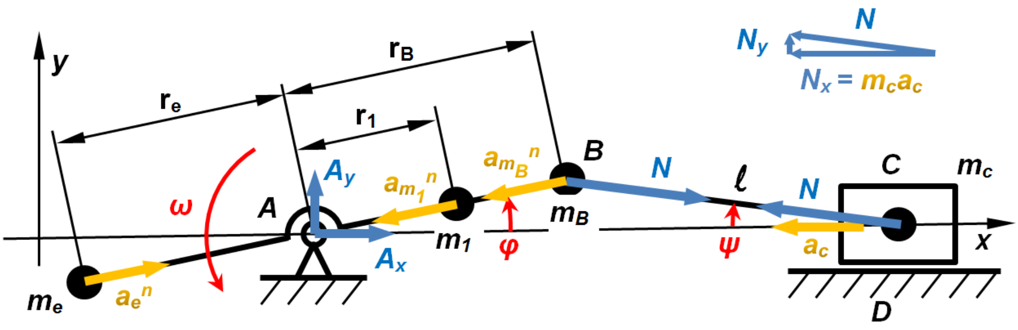

Fig. 9. Kinetic model of crank mechanism [10]

At the kinetic model of crank mechanism the masses are considered without extension: m_e is the mass of the circle section part, m_1 is the mass of the stick, m_B is the mass of pin, bearing and housing connected to the disc, together with the half of the crank, m_C is the mass of the coupling rod with the connected bearing, housing and the other half of the crank. As an engineering approximation the angular velocity of disc ω is considered constant. Accordingly the relationships are the followings:

$$r_B sin\,\varphi = l\,sin\psi \tag{9}$$

$$\psi = arc\,sin\frac{r_B}{l}\,sin\varphi \tag{10}$$

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$$N = \frac{m_C\, a_C}{\cos\psi} \tag{11}$$

The kinetic equations of movement written to *x* and *y* direction according to D'Alambert:

$$-m_e\, r_e \omega^2 cos\varphi + \; m_1\, r_1 \omega^2 cos\varphi + \; m_B\, r_B \omega^2 cos\varphi - \; m_C a_C + \; A_x = 0 \tag{12}$$

$$-m_e\, r_e \omega^2 sin\varphi + \; m_1\, r_1 \omega^2 sin\varphi + \; m_B\, r_B \omega^2 sin\varphi - \; m_C a_C tg\psi + \; A_y = 0 \tag{13}$$

and so the *x* and *y* directional force components awakening in the point *A*:

$$A_x = m_C a_C + \omega^2\, (\, m_e\, r_e cos\varphi - \; m_1\, r_1 cos\varphi - \; m_B\, r_B cos\varphi) \tag{14}$$

$$A_y = m_C a_C + \omega^2\, (\, m_e\, r_e sin\varphi - \; m_1\, r_1 sin\varphi - \; m_B\, r_B sin\varphi) \tag{15}$$

According to (14) and (15) equations a numerical analysis was executed, on which base an optimization was implemented [9]. The goal of the optimization was bringing to the same level the maximum value of the A_x and A_y bearing force components. Fig. 10. presents the changes of the bearing force components during a whole rotation beside optimized parameters. By increasing the mass m_e of the counterweight, the maximum value of the force component A_x decreases, and the maximum value of A_y increases. As the counterweight decreases, the consequence is converse.

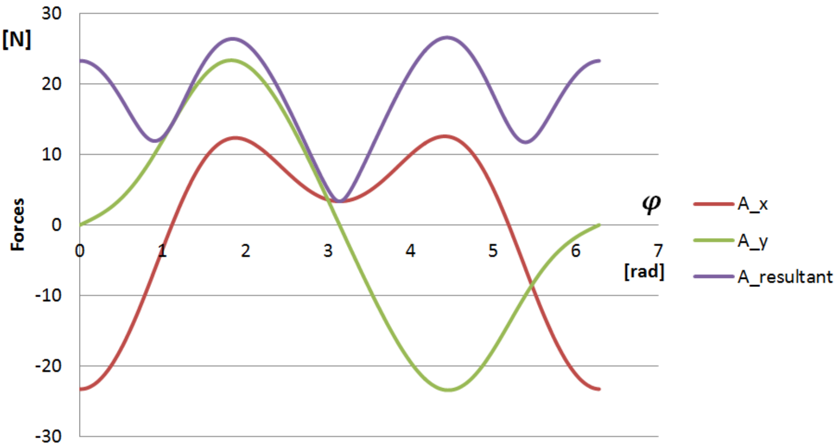

Fig. 10. Changes of force A and x, y components of it during a whole rotation [10]

The test of the fatigue machine with the partly dynamic balanced crank mechanism was successful. For rotation control of the electric motor a frequency converter was applied. For counting the number of bending an optical sensor with a CPU was installed.

3. CONCLUSIONS

One of the main goals of this project was to investigate the fatigue of the previously specialized composite materials and for this purpose a fatigue test machine was developed and manufactured. During the construction initially an extant shaker played the key role, than an electric drive and crank mechanism came to the front. In the course of the development the bearings and the partly dynamic balancing had got special attention. The fatigue machine was completed and was working for more than 20 weeks without any problems. As one specimen's test needed almost two weeks continuous work of the fatigue machine, a new development idea emerged for a multi-gripping and bending system that could be developed in the future.

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