THE ELASTOHYDRODYNAMIC (EHD) LUBRICATION OF A MECHANICAL FACE SEAL

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ABSTRACT

There are various causes for face seal leaks. Leakage normally takes place thru the radial seal gap formed by the two sliding surfaces. Calculations are based on the assumption the a hydrodynamic film exists in face seals and that the leakage can be calculated in accordance with the known equations for laminar flow thru a radial annular gap. The power consumption can be calculated also from the Newton relation. Normally mechanical seals have radial rigid plain faces and only in special cases do spherical sealing surfaces occur. The form of the surfaces can, however, be altered by heating and wear, for instance.

KEYWORDS: elastohydrodynamics, lubrication, mechanical face seal.

1. INTRODUCTION

There are various causes for face seal leaks. Leakage normally takes place trough the radial seal gap formed by the two sliding surfaces. Only the primary leakage trough the seal gap between the faces of the seals will be considered. Since in practice the liquid film thickness *h* is seldom constant and the actual gap form can considerably deviate from the assumed parallel gap because of temperature differences in the ring, deviations from the theoretical calculations are to be expected.

In addition to the mechanical forces different temperatures and their gradients also influence the geometry of the seal gap. In the case of elastic distortion the magnitude of the elastic modules and the dimensions of the rings are the determining factors, and in thermal distortions it is the values of the thermal properties of the materials and the heat transfer factors in conjunction with the construction of the rings which influence the temperature gradients and, in turn, the gap shape. The temperature gradient in both axial and radial directions influences the geometry of the seal gap.

In mechanical seals there are often several heat sources which strongly influence the radial temperature distribution in the rings. In addition to the friction heat from the sliding interface, the medium to be sealed may be a source of heat, as well as a hot shaft or housing, and heat from liquid turbulence.

Depending on the direction of the temperature gradient, the temperature distribution in the sealing rings can considerably differ. Those areas of the rings which lie furthest from the heat sink or are nearest to the source of heat show the highest temperatures. They expand to a grater extend then other sectional areas and alter the shape of the original parallel film gap.

In face seals the contact points of the asperities on the two seals surfaces pressed against one another with an average pressure, could distort both plastically and elastically. Thus, the most highly stressed bearing points will be distorted plastically or worn away while the neighboring areas will be elastically distorted. In the usual combinations of materials for face seals, a carbon graphite ring is usually run against a metal, metal oxide, or carbide ring with different elastic modules.

Since the surfaces is always highly finished, the low modules ring will always take a considerably greater proportion of distortion. For these reason the sliding surfaces under load will look more like an aerial photograph of a group of lakes then of a group of islands, the hollow spaces between the surfaces are seldom connected to one another. With the rotation of one ring it is possible to imagine that the liquid is transported from one hollow to another, as in the case of a revolving door, until the liquid particles emerge at the far side of the gap. Thus, in this range there will be no detectable influence of viscosity. As long as liquid inflow and outflow counterbalance each other no pressure builds up in the interface. Where roughness of the sliding surfaces is uniform, the leakage losses with an exchange flow are independent of the seal width, but strongly dependent on the value of the roughness, the closing pressure, the rubbing speed, and the size and direction of the centrifugal pressure. Further more, the inflow and outflow

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Letter to Editor

MODEL AND ANALYSIS OF A MECHANICAL SEAL BY FINITE ELEMENT METHOD. INTERFACE TENSION DISTRIBUTION

UDC 539.3+519.6

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Abstract. *Modeling to access solutions is the goal of predictive engineering. The paper presents boundary element analysis or the numerical simulation of the behavior of a mechanical face seal. The present boundary element analysis is a particularly one for it contains the nonlinear effect due to changes in boundary conditions resulting from the contact of the static ring and the sealing head of the face seal. These all have significant influence on the behavior of the system. The results can be used for optimizing designs, predicting limits or investigating failures.*

INTRODUCTION

In a face seal (Fig. 1), an axial force presses a rotating floating ring 5 against a fixed counter face 6. The axial leakage path between the floating ring and the shaft is closed by a static seal such as an O-ring 7. The static and sliding surfaces of the traditional stuffing box are effectively interchanged, with the advantage that the geometry of the sliding sealing surfaces can now be produced more accurately and less expansively and there is no longer any wear on the shaft to shaft sleeve.

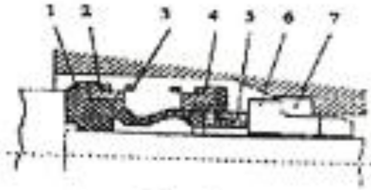

Fig. 1.

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INTERFACE CONTACT PRESSURE DISTRIBUTION IN DYNAMIC CONTACT FACE SEALS, ANALYZED BY FEM

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Summary: *A great diversity of contact seal designs, materials, operating conditions and factors that affect their performance have not yet allowed the general conclusions on friction and wear of this seals to be drawn. At the same time, the results of studies of particular cases may often lead the design engineer to an erroneous decision if the seal develops is different in some way from that he has taken as a prototype. In some instances, these factors are interdependent. The paper presents the service conditions of sliding contact seals in machinery, determined by combinations of the above factors analyzed by FEM (finite element method).*
Keywords: *mechanical face seal, dynamic seal, tribology, FEM (finite element method), leakage, wear, friction*

1. INTRODUCTION

The performance of seals is characterized by the degree of tightness, service life, power losses, by the extent of damage to the contacting surfaces in operation, etc. the degree of tightness, wear life t_w, and performance factor i are the most important characteristics of seal performance. In addition to the above factors, temperature, whose level is determined by their joint action, also effects the performance of dynamic seals. The service conditions of sliding contact seals in machinery, determined by combinations of the above factors, are very diverse. The temperature, pressure, flow rate and properties of the fluid are chosen depending on the seal application.
Dynamic contact seals such as face (axial) seals operate with external continuous friction.

2. FRICTION AND WEAR IN DYNAMIC CONTACT SEALS

A great diversity of contact-seal designs, materials, operating conditions, and factors that affect their performance have not yet allowed the general conclusions on friction and wear of these seals to be drawn. At the same time, the results of studies of particular cases may often lead the design engineer to an erroneous decision if the seal he develops is different in some way from that he has taken as a prototype.
For the face seals design calculation methods have been devised for the assessment of fluid pressure (with regard to out-of-squareness of the faces and to pressure in the clearance), behavior of the fluid sealed in face clearances, hydrodynamic effect for the sealing rings, deformations of the rings due to pressure and temperature, and also temperature fields in the rings of the rubbing pair.

3. DESIGN AND OPERATION OF A MECHANIC FACE SEAL.

In a face seal, an axial force presses a rotating floating ring against a fixed counterface or vice versa. The axial leakage path between the floating ring and the shaft is closed by a static seal such as an O-ring, elastomeric sleeve, or U-seal, etc. Figure 1 shows a simple form of face seal. The static and sliding sealing surfaces of the traditional stuffing box are effectively interchanged, with the advantage that the geometry of the of the sliding sealing surfaces can now be produced more accurately and less expensively, and there is no longer any wear on the shaft or shaft sleeve. To compensate for any lack of alignment of the seal faces and for longitudinal thermal

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Overall tribology

AUTOMOTIVE MECHANICAL FACE SEALS – TRIBOLOGICAL SIMULATION

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ABSTRACT

The behaviour of a face seal is determined by the complex interaction of a number of factors. Advantages are usually attained at the price of disadvantages in other directions. For example if the roughness is constant, an increase of the contact pressure reduces leakage, but the wear and frictional heat increase. As against this, increasing leakage losses can reduce the friction and heat production, but the effectiveness of the unit as a seal is reduced. Again, a high friction may not only lead to increased wear but also, due to thermal distortion, to considerable leakage losses, or it may cause the seal to break down because of a thermal stress cracks. By appropriate seal design, choice of materials and type of seal arrangement, individual requirements such as minimum leakage, maximum life or minimum friction can be met. The paper presents the finite element method (FEM) and experimental analysis of the leakage and friction rates of a mechanical face seal for the chemical industry.

Keywords: mechanical face seal, simulation, finite element method, tribology.

AIMS AND BACKGROUND

The behaviour of a mechanical face seal in run can be simulated and illustrated on the computer.

For any of the type dimensions taken into consideration the basic structure consists of primary sealing, formed of:

- the pressure ring/floating ring, elastic and/or in movement of rotation;
- the fixed counter face friction ring.

The ensemble of the sealing shows axial symmetry, both from the point of view of geometry and mechanical loading[1].

* For correspondence.

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Overall tribology

CALCULATION BY FINITE ELEMENT METHOD (FEM) OF TEMPERATURE DISTRIBUTION IN THE COMPONENTS OF A MECHANICAL FACE SEAL

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ABSTRACT

The paper makes analysis of temperature distribution in mechanical face seal components, which are used in household equipment and passenger cars and trucks water pumps.

Keywords: mechanical face seal, simulation, finite element method, tribology, temperature distribution.

AIMS AND BACKGROUND

The temperature in seal interface due to frictional heat has a considerable influence on the behaviour of a face seal, affecting wear and thermal distortion. If the temperature is too high vaporisation of the lubricant film occurs, causing increased friction and wear[1]. The safe working temperature of the face materials can also be exceeded and the seal will then fail due to welding or thermal stress cracks. The friction heat produced in the seal interface can be removed by conduction, convection and radiation processes. The heat produced in the seal interface is transported away by conduction in the seal rings and from the seal rings it passed onto surrounding fluids by convection. Some also is lost to the surroundings by radiation.

The maximum temperature in the interface is of particular interest.

Face seals are used over a temperature range from –200 to +1000°C. The proprieties of many materials are only known at room temperatures so that exact calculation of the maximum permissible temperature difference and thermal resistance factors at the working temperatures is not possible.

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Make Harmony Between Technology and Nature, and Your Mind will Fly Free as a Bird

CONSIDERATIONS UPON THE CIRCULAR SECTION CIRCLIPS/RETAINING RINGS AXIAL LOAD-CARRYING CAPACITY

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Abstract: *The reference standards as well as the specialty literature offer very many useful data to the designer. But, in specific applications, requiring small axial limits, they are insufficient to provide a satisfying safety without additional calculation. By means of FEM (finite element method), the authors are trying to point out the real behavior of the assembly in such cases*

Key words: *circlips, axial load-carrying capacity,FEM*

1. INTRODUCTION

Circlips / retaining rings are designed to position and secure component in bores and houses. Simultaneously they provide rigid end – play take – up in the assembly to compensate for manufacturing tolerances or wear in the retained parts. For the technical designer, who uses standardized and/or in a list manufactured rings on shafts or in housings with nominal diameter, a computation is not necessary. It is of crucial importance however with special applications of the normal rings and particularly with the construction of special rings.

The reasoning for the fundamentals of the bending calculus is presented in detail (Mesaroş-Anghel et al., 2006) for circlips with rectangular section, and FEM analysis has been done.

The authors propose to analyze circular section circlips. The conclusions of the paper can be directly applied in technical design. In the future, if experimental research results are added, also the reconsideration of the present standards regarding shape, dimensions, and materials can be made

2. FUNDAMENTALS

The strength calculation is based on the consideration that a circlip -for the shaft or for the housing – is a curved bended bar.(Argeşanu, 1999); (***. 1973); (Voinea, 1989) The ideal solution is a curved bar of same firmness (Hübener, 1970).

The circular section is particularized in fig. 1 and the following equations:

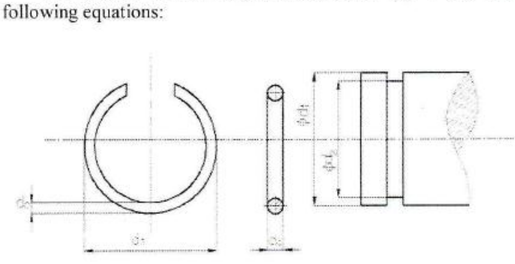

Fig.1. Circular sectionof the circlip

$$\frac{1}{r}=\frac{M_b}{EI} \quad (1)$$

If a bar with a neutral radius of curvature r, already curved, is deformed on a radius ρ, the equation is:

$$\frac{1}{r}-\frac{1}{\rho}=\pm\frac{M_b}{EI} \quad (2)$$

Using the names for the neutral diameters, usual with circlips

$$r=\frac{1}{2}\cdot D_3\ ;\quad \rho=\frac{1}{2}\cdot D1\ ;\quad \frac{2}{D_3}-\frac{2}{D_1}=\pm\frac{M_b}{EI} \quad (3)$$

$$\frac{1}{D_1}-\frac{1}{D_3}=-\frac{\sigma_b}{E\cdot d_c} \qquad \frac{1}{D_3}-\frac{1}{D_1}=-\frac{\sigma_b}{E\cdot d_c} \quad (4)$$

$$\sigma_b=\frac{(D_1-D_3)\cdot E\cdot d_c}{D_1\cdot D_3} \qquad \sigma_b=\frac{(D_3-D_1)\cdot E\cdot d_c}{D_1\cdot D_3} \quad (5)$$

$$\sigma_b=\frac{1{,}15(d_1-d_3)\cdot E\cdot d_c}{(d_1+d_c)(d_3+d_c)} \qquad \sigma_b=\frac{1{,}15(d_3-d_1)\cdot E\cdot d_c}{(d_1-d_c)(d_3-d_c)} \quad (6)$$

$$\sigma_b=\frac{M_b}{W}=\frac{P\cdot l\cdot 6}{d_c^2\cdot s}\ becomes\ P=\frac{\sigma_b\cdot d_c^2\cdot s}{6\cdot l} \quad (7)$$

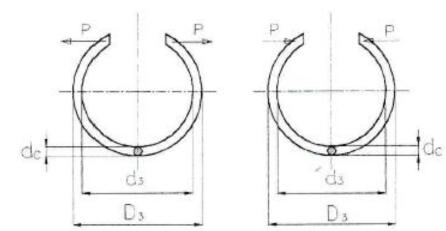

Fig.2. Forces acting on a circlip

3. COMPUTATION OF THE CIRCLIP

For the special cases mentioned above it is of interest the axial loading behavior (and axial load-carrying capacity) of the ring as well as its stability in the reserved groove (in the shift or housing)

Fig.2 presents the situation in which a machine part presses a circlip with an axial force. At first sight, shearing seems to condition for the drawing out of use of the ring, so that, at the beginning of the use of these machine elements it was very much insisted on this kind of calculation. It was observed that, because of the relationship between the depth of the groove and the thickness of the ring, the shear never takes place because at loadings under the maximum stress there takes place a loss of stability by deformation (as in fig.3a,b,c)

It is said that the ring is "inverted" The deformation is determined by the occurrence of a lever arm that modifies by a bending moment the shape of the ring that becomes conical. For a better understanding of the situation, the deformation (the characteristic angle ψ) is exaggeratedly enlarged. As to be seen, between the chamfer g and the level of the elastic deformation i

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Make Harmony Between Technology and Nature, and Your Mind will Fly Free as a Bird
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DETERMINATION OF THE OPTIMUM VARIANT OF SHAFT-HUB JOINT FOR GEARS

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Abstract: *The evolution of the constructive solutions of the joint that form the cylindrical fitting specific to the gears determined the occurrence of some typified families whose carrying capacity tend to equal the performances obtained by the joint by shrinking joints. From figure 3 can be observed the advantage shown by the shrink joints with the double conics intermediate elements fig.1 c, which allows the obtaining of high performance solutions.*
The choice of one of the joint solution (fig.1) according to the analyzed parameters (G/M_t); $(G/M_t)/(G/M_t)$ and $(G/M_t)/(G/M_t)_e = /(G/M_t)/ (G/M_t)_e /$ (d)- (fig.3)- give the best accomplishment of the formulated requirements.
Key words: *shaft-hub, gears, key joints, elastic hub-hydropath*

1. INTRODUCTION

The principle of relatively immobilization (joint) of some parts that form the specific cylindrical fits for gears may be of the type: with or without intermediate elements (Manea, 1970). (Gheorghiu, 1986; Sheng, 2008)

From the category of the ones with intermediate elements are mentioned:

-Shrink joints with cylindrical smooth surfaces with intermediate conic elements -rings- annular keys (fig. 1.a);

-Shrink joints with cylindrical smooth surfaces with intermediate elements elastic hub-hydropath (fig.1.b);

-Shrink joints with double conic intermediate elements (fig. 1.c);

-Key joints (fig.1.d).

Of the category of those without intermediate elements are mentioned:

-Shrink joints pushed or fretted (fig. 1.e).

According to the figure 1 it is observed the existence of a large number of modalities of relatively fixing the shaft and the hub, but the main request of choice of one of the alternatives is constituted by the ratio between the weight and the transmissible torque, that is that of a maximum carrying capacity.

In the choice of the fixing solution of the gear on the shaft, it always must be made a compromise between the economic requirement for a compact construction, of some low costs of fabrication and the technical condition of a great carrying capacity (Gheorghiu & Madaras, 1986).

The best accomplishment of some requirement like: high reliability, safety in exploitation, the minimum of tension concentrations, high qualities of the fabrication of joint surfaces etc., depends on the consideration of the following conditions:

a) The choice of a solution that best accomplishes the working requirements.

b) The compact construction and the right dimensioning of all the joint elements.

c) The use of proper materials.

d) Mechanical working and thermal treatments as cheap as possible.

e) The easy assembling and disassembling.

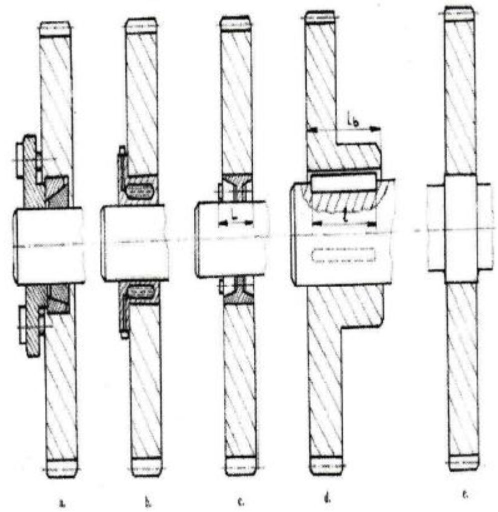

Fig. 1.

f) Optimum servicing and working conditions for on all the load steps.

Each condition in part has a more or less influence on the carrying capacity, so at the question regarding "THE MOST FAVORABLE" choice of the joint it cannot be found an universal right/valid answer without a comparative analysis. (Zografos, 2008; Korakianitis, 2008)

2. COMPARATIVE ANALYSIS

Among the options of fixing the gear on the shaft (fig.1) are analyzed the alternatives c) and d) which are considered to be the most used in the machine construction.

The choice derives also by the fact that at an inadequate process of the joint surfaces, but also a inadequate thermal treatment or a defective assembling drives at unfavorable displacement of the contact spot which determine the occurrence of some extremely high normal forces (fig.2).

The consequences can be failures of the surface, bending breaking or displacements.

In such of these cases, errors of execution and assembling determine a real carrying capacity, which can be really different from the calculated one. (Guang, 2007; Gheorghiu, 1986)

The calculations were made for:

- $d \in (25;400)$mm- the shaft diameter;
- βBG- the fatigue coefficient of concentration [v.2];
- $\varepsilon\delta$- dimensional coefficient [v.2];
- $\gamma\delta$- the surface quality coefficient [v.2];
- Shaft material OL 50 STAS 500/2-80;
- The torsion of the shaft is after a pulsate cycle;
- c=1,5 the safety coefficient at the fatigue resistance;

Analytical and experimental modeling of the drivers spine

Veronica Argesanu, Raul Miklos Kulcsar, Ion Silviu Borozan, Mihaela Jula, Saša Ćuković, Eugen Bota

Abstract—The aim of this study is to determine a analytical expression in the coronal plane of the drivers spine while driving along curved roads and also to determine ergonomic parameters for the car seat design. To determine the analytical expression and the ergonomic parameters, an experiment was developed to monitor the position variation in time of the vertebras in the coronal plane. The result lead to three sinusoidal equations. The amplitude values of the sinusoidal functions describing the variation in time of angles between the vertebras gives an image regarding the deformation degree of the intervertebral discs.

Keywords—Spine, ergonomics, vehicle, musculoskeletal affections.

I. Introduction

THE possibility to drive in complete healthy and safety conditions not only for the professional drivers but also for the rest of the population which uses vehicles as frequent transportation means leads to efficiency by improving the quality of life.

In this context it is noted the following objectives and research directions: the development of modern mathematical models and principles to be included in a design or control algorithm.

The present study is based on the egronomical research regarding the spine's behavior while driving along curved roads.

II. Analytical expression of the spine in the coronal plane

The optimal ergonomic body posture of the driver sitting in the car seat is influenced by the structural characteristics of the seat. The body has to be constrained to the seat such way so that the spine's form is an ideal anatomical or ergonomic optimal shape. Therefore to design and construct the car seat, it is proposed to start from the ideal anatomical shape of the spine in the coronal plane (Fig. 1). [2, 3]

To determine the design parameters of the car seat is necessary to know the analytical form of the spine's shape in the coronal plane.

In the coronal plane, the shape of the spine can be expressed mathematically by the equation of a straight vertical line. Vertebrae centers are collinear. Considering a reference system as in figure 2, the vertical line's equation containing vertebras centers is considered to be $x = 0$.

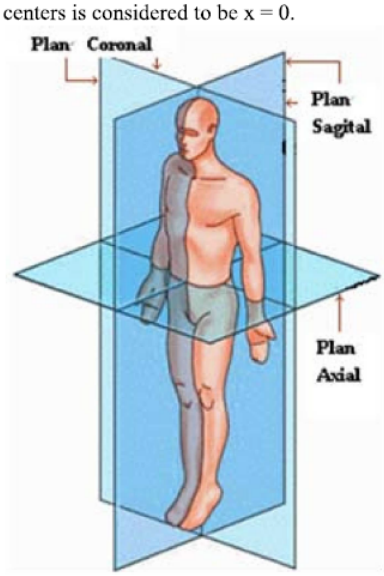

Fig. 1- Anatomical planes.

Point O, the origin of the coordinate system coincides with the lowest point of the coccyx.

The analytical expression $x = 0$ of the spine's shape in the coronal plane is only valid if the vehicle is at rest, or the vehicle travels on a rectilinear continuous road (unreal case).

Due to the centrifugal force acting on the human body while the vehicle is traveling along a, the human body changes its posture in the coronal plane in the opposite direction of the centrifugal force, to maintain the balance in the car seat. Thus the spine's shape changes depending on the vehicle's traveling speed and the curved path's radius, causing the spine shape mathematical expression in the coronal plane to be a motion law.

The spine shape is the line containing the centers of the vertebras. Anatomically, the shape and movement of the spinal column are shown by the relative rotational movement between the vertebras. According to anatomy and kinematic studies of the human spine, it is concluded that the center of rotation between two vertebras is the center of the intervertebral disc that connects the two vertebras. Thus

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Experimental determination of the intervertebral stress

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Abstract— The aim of this study is to determine the intervertebral stress that leads to spine musculoskeletal affections. To determine the intervertebral stress, the L4 and L5 vertebras were made by using rapid prototyping. Between the vertebras the intervertebral disc was made by using silicon material. In the intervertebral disc five force sensors were inserted. A STEWART platform was used to remake the vertebra's relative movements during that time with the aid of the five force sensor, the intervertebral stress was recorded. It was clearly shown at what load and movements during the drive, spine musculoskeletal affections can appear.

Keywords—spine, ergonomics, vehicle, musculoskeletal affections.

I. INTRODUCTION

To determine the musculoskeletal affections it is necessary to perform a stress and deformation analysis of the spinal column by modelling, simulation and experimental validation.

The continuous contraction of the muscles from the spinal column determines a supplemental load on vertical direction increasing the equivalent stress from the vertebrae and especially in the intervertebral discs determining deformations which in some cases cross over the point that musculoskeletal disease can be treated or recovered by physiotherapy. [9, 10, 11]

The form of the spinal column in sagittal plane and the amplitudes of the inclination in the coronal plane are determined directly from the driver body reactions to the forces that occur from the car running on different routes. In the case of a non-ergonomic position of the spine in sagittal plane, the amplitudes of the inclination in the coronal plane lead to deformation of the intervertebral discs, which exceed the aforementioned anatomical limits. [1, 2]

The driver's body optimal ergonomic posture while seated on the car seat is influenced by construction characteristics of the seat in order to constrain the body as its spine shape to follow the ideal anatomic spine shape or ergonomic optimal. As this is known, in order to design and manufacture of auto vehicles car seats, it is proposed to start from the ideal anatomical shape of the human spine. [3, 4, 5]

The analysis aims to determine the equivalent stresses of the two vertebrae and intervertebral disc special deformations.

In order to determine the design parameters of the car seat it is necessary to know the analytical shape of the spine in sagittal plane and coronal plane.

The analytical expression of the spine in sagittal plane:

$$y=\frac{1}{L}\left(\frac{m}{6}-\frac{A}{3}\right)\cdot x^3+A\cdot x^2-L\cdot\left(\frac{m}{6}-\frac{2\cdot A}{3}\right)\cdot x+\frac{R}{L}\cdot x+\left(\frac{m}{6}-\frac{2\cdot A}{3}\right)\cdot\frac{L^2}{\pi}\cdot sin\frac{\pi\cdot x}{L}-\frac{R}{L}\cdot sin\frac{\pi\cdot x}{L} \tag{1}$$

L and R are x, y coordinates of the point L5-S1 from figure 1.

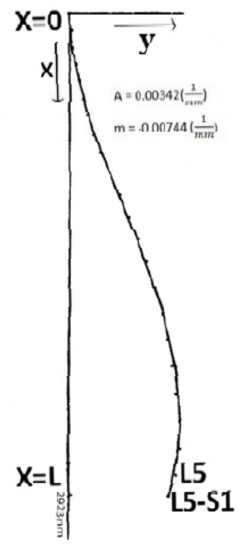

Fig. 1. The spine analytical shape in the sagittal plane.

From the analysis of 30 X-Rays have been obtained values for A parameter between $0.00003mm^{-1}$ and $0.00005mm^{-1}$ and for the m parameter values between $0.00005mm^{-1}$ and $0.0015mm^{-1}$. These values are for the erect position of the spine.

For the seated position the values are: $A=0.00004mm^{-1}$ and $m=0.0016mm^{-1}$. [6, 7, 8]

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Human Body Posture before and after Maxillofacial Surgery

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Abstract

The aim of this study is to observe how the posture is affected by maxillofacial fractures and how it is improved after maxillofacial surgery. The procedure for all tests is non-invasive and it's formed by a Multi Sensor Electronic Baropodometric Platform, a PodoScanalyzer (2D), the D.B.I.S Software witch calculates the values of the B.P.I. Index , Body Analysis Kaptur System and Dynamic Image System. Using the Baropodometer we evaluate the balance of the human body, oscillations of the pressure centre, visual information of poor posture. a group of 10 patients with different maxilla fractures were investigated before surgery and after. For each patient there were made two analyses. A dynamic analysis which determined how the maxilla fracture affects the walking, and how it is improved after the surgery, and a stabilometry analysis which determined how the maxilla fracture affects the standing posture of the body, and how it is improved after the surgery. As conclusions, the body posture is affected by the maxillofacial fractures and improves after the surgery.

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Keywords: posture; maxillofacial surgery; baropodometer

1. Introduction

The posture is the human body behavior in relation with the environment in which he lives, and in relation with the laws that governs these environment, first of all the force of gravity. To do this, man has developed a specialized structure to overcome gravity, called the tonic postural system of vertical stability. [4, 5]

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