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MACHINE DESIGN 2009

THE EDITOR IN CHIEF: PROF. PHD. SINIŠA KUZMANOVIĆ

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PROGRAM MODULE FOR STRENGTH CHECK OF THE SHAFTS AND AXLES ACCORDING TO THE DIN 743

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Abstract: *Faculty of Mechanical Engineering in Nis, Design department is working several years on the program system for **P**ower **T**ransmitter **D**esign – PTD. Standard DIN 743 gives new approach in calculations of the shafts and axles. Therefore has developed program module of program systems PTD, which eases complicate calculations of the shafts and axles according to the DIN 743.*

Key words*: DIN 743, Shaft, Design, Software*

1. INTRODUCTION

Global market gives more and more complex challenges to the manufacturers in areas of productivity, quality and rapid product development. Intensive growth of industrial needs delivers increase of project – designing tasks with more and more complexities included into the project realization. Engineering praxis, as imperative, demands application of computer sciences into all phases of product development process.

Application of computer sciences and computers is possible in the following phases/tasks of product development:

- Representation and modeling,
- Processing and data management,
- Documenting,
- Analysis and deduction processes,
- Calculations and simulations,
- Data acquisition,
- Optimization,
- Diagnostics,
- Processing and knowledge management,
- Product concepts generation.

Effects of computer's application into the product development process are:

1. Shorter cycle of design and time reduction needed for product selling,
2. Costs decrease,
3. Quality improvements,
4. Product complexity increase,
5. Possible variant solution's number increase,
6. Dislocated design, manufacture and maintenance.

These effects are possible because of:

1. computer's processing power increase, from the aspects of hardware and software, informational technologies,
2. increase of software's capabilities,
3. increased knowledge of engineers about computer science,
4. methods capable to integrate CAx tools into the design process,
5. virtual product design process.

Shorter design cycle and product development time decrease come from:

- automatic generation of work documentation from virtual models,
- faster final technical documentation generation,
- automatism of repeating tasks,
- simulations,
- automatic controlling and product validation,
- integrated product design,
- decreased number of demands for design changes,
- shorter time of design changing and changes implementation.

Design costs can be decreased from the following:

- designer's costs decreases,
- cost decreases in prototyping and testing,
- manufacturing costs decreases,
- guarantee cost decreases.

Based on the all fact given above, Faculty of Mechanical Engineering in Nis, Design department is working several years on the program system for power transmitter design – PTD.

Program system for gear power transmitter's design PTD is complex and heterogenic. System is based on modular principle and enables computer based definition and design engineer's task solving. This system is part of the intelligent integrated system for gear power transmitters design software developed at Faculty of Mechanical Engineering in Nis. Basic task of this system is to enable integrated application of various program modules and systems developed by various firms and authors into the one complex and functional system. Because of the universality and variant number of the authors, system relies on the application of data exchange, communication and sharing.

Integrated program system for power transmitter's design PTD (Fig. 1) has three different program modules:

1. program modules for power transmission calculations,
2. program modules for rotating elements calculations,
3. program modules for mechanical connections calculations.

First module of PTD, which includes power transmitters design, can be used for calculation of the spur, conical and worm gear pairs, friction, chain and belt pairs, as well.

Second module covers calculation of the shafts, roller and sliding bearings, and third module is used for calculations of the pins, pressed connections, bolted connections and notched connections.

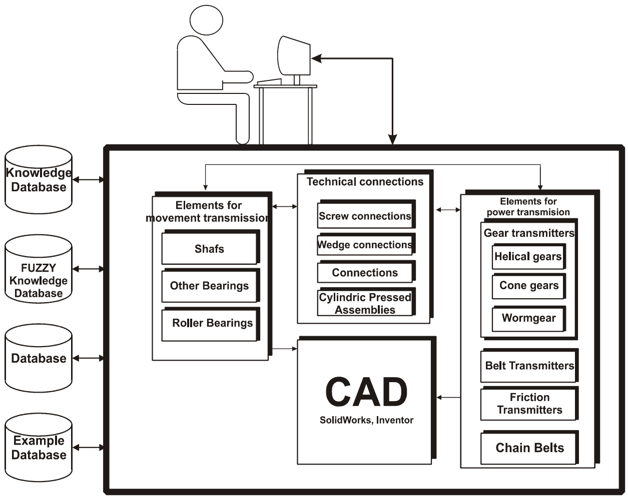

Fig. 1. Integrated system for power transmitters design PTD

In the module of PTD, used for rotating parts calculations, there are two program modules for shafts and axles calculations – program submodule for shaft dimensioning and program module for strength check of shafts and axles according to the DIN 743 standard.

2. STRENGTH CHECK OF THE SHAFTS AND AXLES ACCORDING TO THE DIN 743 STANDARD

Stress and strain analysis is some of the main tasks expected from design engineers to fulfill. Design is based on the theory of material strength and possibilities of materials properties usage. From the early beginnings of engineering era, engineers have been trying to get functional dependency between loads and dimensions of the elements. In most of the cases, there was a big uncertainty and complex mathematical dependency of several factors and people started to use already gathered knowledge and similarities. Engineers started to make probe tools that had familiar properties and behavior and later compared other manufactured parts with probe tool. That was the early beginning of the standards creation and progress in design theory.

The German standard DIN 743 is still a novel standard, prepared by the German institute for standardization and the Institut für Maschinenelemente und Maschinenkonstruktion of the TU of Dresden, Germany with the main objective was to make available for the engineering community a standard focusing on strength analysis of shafts and axles. The standard is based on the standard TGL 19340 of the former German Democratic Republic, the VDI 2226 of the Federal Republic of Germany and the FKM guideline compiled by the IMA Dresden, Germany. The proof of strength is based on the calculation of a safety factor against fatigue and against static failure. The safety factors have to be higher than a required minimal safety factor. If this condition is fulfilled, proof is delivered.

The standard consists of four parts:

- Introduction, analysis method
- Stress concentration factors and fatigue notch factors
- Materials data
- Examples

The analytical proof considers bending, tensile/compressive and shear stresses due to torsion. However, shear stresses due to shear forces are not considered, hence use of this standard for short shafts requires caution.

Only the fatigue limit is used in the proof, no proof for finite life strength is delivered. Materials data are based on 10^7 stress cycles with a probability of survival of 97,5%. The safety factor required in the standard covers only the uncertainty in the analysis procedure. Additional safety factors or an increased safety factor due to uncertainties in the load assumptions and due to the effects of a failure are not defined. They have to be defined by the engineer. The notch factors for feather keys are questionable since no difference is made for the different key forms. All loads (bending, tensile/compression, shear) are in phase.

The standard is limited to non-welded steels in the range of –40C° to 150C°. The environment has to be non-corrosive for application of this standard.

One of the biggest problems of engineers and technicians during work is safety factor determination for some specified element or system. Safety factor is measure of designer's uncertainty about some parameters of the system and safety for unwanted deformation avoiding.

Development of the knowledge and faster information exchange deliver more and more improved methods for safety factors calculations. There are no universal methods for every structure. For every design for any specific structure specific method for safety factor's design has been designed, but similarities for every safety factor determination exist. Shaft's and axle's design has to shape and dimension shafts and axles, find critical sections in order to determine geometrical discontinues and stress increases.

Standard DIN 743 involves equations for safety factor determination in critical sections of shafts and axles according to the two basic criteria:

1. safety in relation to the plastic deformation of the part (static safety factor),
2. safety in relation to the dynamical strength of the material (dynamic safety factor).

Calculations include torque, pressure/tension and deflection of the shafts and axles. Shearing is not concerned into the calculations.

Results determined according to the DIN 743 remove any doubts about safety in critical sections of shafts and axles and fulfil demands of the engineers for successful, safer and complete dimensioning.

2.1. Static safety factor

Static safety factor has to be calculated during maximal loads given to the shaft. These loads are given during start up of the shaft and these high loads deliver maximal stresses in critical sections of the shaft.

Calculated value of the safety factor should be greater or equal to the minimal value of the safety S_{min}.

$$S \geq S_{min} \tag{1}$$

where: S_{min}=1.2 according to the DIN 743.
If simultaneously pressure/tension, deflection and torque load the shaft, safety factor is determined as (2):

$$S = \frac{1}{\sqrt{\left(\frac{\sigma_{zd\max}}{\sigma_{zdFK}} + \frac{\sigma_{b\max}}{\sigma_{bFK}}\right)^2 + \left(\frac{\tau_{t\max}}{\tau_{tFK}}\right)^2}} \tag{2}$$

where:
σ_{zdmax} - maximal normal stress delivered by the pressure/ tension;
σ_{bmax} - maximal normal stress delivered by the deflection;
τ_{tmax} - maximal tangential stress delivered by the torque;
σ_{zdFK} - yield stress of material for pressure/tension;
σ_{bFK} - yield stress of material for deflection;
τ_{tFK} - yield stress of material for torque.
Yield stress values are determined according to (3) ion and according to (4) for torque.

$$\sigma_{zd,bFK} = K_1\left(d_{eff}\right) \cdot K_{2F} \cdot \gamma_F \cdot \sigma_S\left(d_B\right) \tag{3}$$

$$\tau_{tFK} = \frac{K_1\left(d_{eff}\right) \cdot K_{2F} \cdot \gamma_F \cdot \sigma_S\left(d_B\right)}{\sqrt{3}} \tag{4}$$

where:
$K_1(d_{eff})$ - technological factor of the influence delivered by the size;
K_{2F} - factor of static strength;
γ_F - factor of yield stress increase;
$\sigma_S(d_B)$ – yield stress of the probe (probe shaft).
According to the standard DIN 743 static strength factorK_{2F} can have values from1 to 1.2, while factor of yield stress increase γ_F can have values from 1 to 1.15. These values can increase static safety factor for 10 to 20%. It is important to point out that static safety factor is dramatically influenced by technological factor of the influence delivered by the size $K_1(d_{eff})$.

2.2. Dynamic safety factor

Dynamic safety factor is determined as ratio of amplitude dynamical strength and amplitude stress in critical sections of the shaft. Just like with static safety factor, it is necessary to determine which loads attack shaft in critical sections and which stresses they deliver. It is important to find median and amplitude stresses, as well.
Calculated value of dynamical safety factor has to be greater or equal to the minimal value of the safety factor $S_{min.}$

$$S \geq S_{min} \tag{5}$$

where: S_{min}=1.2

In the case of the simultaneous loads on the shaft od axle, where pressure/tension, deflection and torque are combined, dynamical safety factor is determined according to the (6).

$$S = \frac{1}{\sqrt{\left(\frac{\sigma_{zda}}{\sigma_{zdADK}} + \frac{\sigma_{ba}}{\sigma_{bADK}}\right)^2 + \left(\frac{\tau_{ta}}{\tau_{tADK}}\right)^2}} \tag{6}$$

where:

σ_{zda} - amplitude stress delivered by the pressure/tension of the shaft;

σ_{ba} - amplitude stress delivered by the deflection of the shaft;

τ_{ta} - amplitude stress delivered by the torque of the shaft;

σ_{zdADK} - amplitude dynamical strength for the pressure/ tension;

σ_{bADK} - amplitude dynamical strength for the deflection;

τ_{tADK} - amplitude dynamical strength for the torque.

Impact factors K_σ, K_τ

Impact factors K_σ i K_τ collect all influences to shafts strength in critical sections of the shaft. They help determination of the border values of the dynamical strength of the shaft and they are determined by the equation (7) for the case of load when pressure/tension and deflection simultaneously load shaft, if load is torque, standard recommends equation (8).

$$K_\sigma = \left(\frac{\beta_\sigma}{K_2\left(d\right)} + \frac{1}{K_{F\sigma}} - 1\right) \cdot \frac{1}{K_V} \tag{7}$$

$$K_\tau = \left(\frac{\beta_\tau}{K_2\left(d\right)} + \frac{1}{K_{F\tau}} - 1\right) \cdot \frac{1}{K_V} \tag{8}$$

where:

β_σ - factor of the stress concentration in the case of the loads: pressure/tension and deflection (united and separately);

β_τ - factor of the stress concentration in the case of the torque;

$K_2(d)$ - geometrical impact factor function of the size;

$K_{F\sigma}$ - factor of the surface roughness for normal stresses;

$K_{F\tau}$ - factor of the surface roughness for tangential stresses;

K_V - factor of the surface hardening.

Ratio of the amplitude and median value of the stresses during increase of the loads, standard DIN 743 gives two different cases of the shape strength calculations (Fig. 2):
Case 1 (σ_{mv} = const., τ_{mv} = const.)
Basis of the safety factor lies in the change of the amplitude of the loads for input drive. Median equivalent loads are constant and calculated as:

$$\sigma_{mv} = \sqrt{\left(\sigma_{zdm} + \sigma_{bm}\right)^2 + 3\tau_{tm}^2} \tag{9}$$

$$\tau_{mv} = \frac{\sigma_{mv}}{\sqrt{3}} \tag{10}$$

Case 2 (σ_{mv} / $\sigma_{zd,ba}$ = const., τ_{mv} / τ_{ta} = const.)

Calculation is based on the preposition that change of the drive loads, ratio of the amplitude and median stresses remains constant.

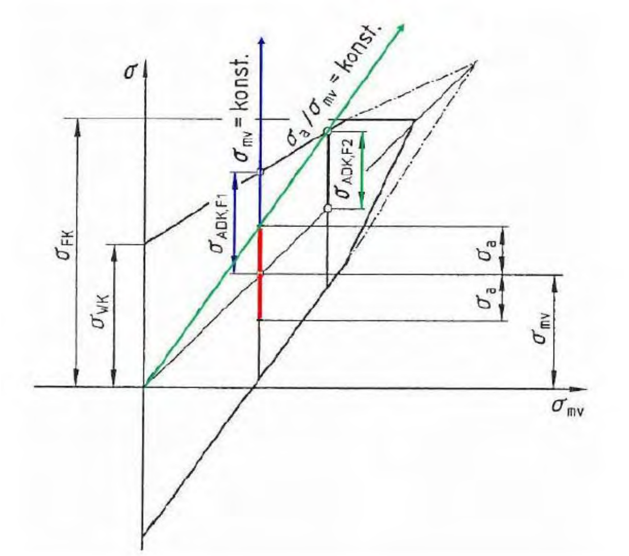

Fig. 2. Smiths diagram

Dynamical strength for probe for pure two-sided loads is calculated in (11),(12) and (13).

$$\sigma_{zdW} = 0.4 \cdot \sigma_B \qquad (11)$$

$$\sigma_{bW} = 0.5 \cdot \sigma_B \qquad (12)$$

$$\tau_{tW} = 0.3 \cdot \sigma_B \qquad (13)$$

where: σ_B – ultimate stress of the probe.
Dynamical strength for pure dynamical loads of the shafts and axles is determined with equations (14),(15) and (16).

$$\sigma_{zdWK} = \frac{\sigma_{zdW}\left(d_B\right) \cdot K_1\left(d_{eff}\right)}{K_\sigma} \qquad (14)$$

$$\sigma_{bWK} = \frac{\sigma_{bW}\left(d_B\right) \cdot K_1\left(d_{eff}\right)}{K_\sigma} \qquad (15)$$

$$\tau_{tWK} = \frac{\tau_{tW}\left(d_B\right) \cdot K_1\left(d_{eff}\right)}{K_\tau} \qquad (16)$$

where:
$\sigma_{zdW}(d_B)$ - dynamical strength for load pressure/tensile for probe with diameter d_B
$\sigma_{bW}(d_B)$ - dynamical strength for deflection of the probe with diameter d_B
$\tau_{tW}(d_B)$ - dynamical strength for torque of the probe with diameter d_B
$K_1(d_{eff})$ - technological impact factor of the size.
Cumulative impact factors K_σ i K_τ influence to the boundaries of the dynamical strength as it is shown in the figure 3. Increase of the impact factors K_σ i K_τ values of the dynamical strength decrease for the pure dynamical load of the shafts and axles.

Factor of the stress concentration β_σ i β_τ

Increase of the stress is influenced with the dis-continuum of the two surfaces on the shaft or axle. Stress increase for the static loads is determined with shape factors α_σ and α_τ, while this increase for dynamical loads is determined with factors of the stress concentration β_σ and β_τ.
Shape factors α_σ and α_τ do not depend of material and they are calculated according to the (17) and (18).

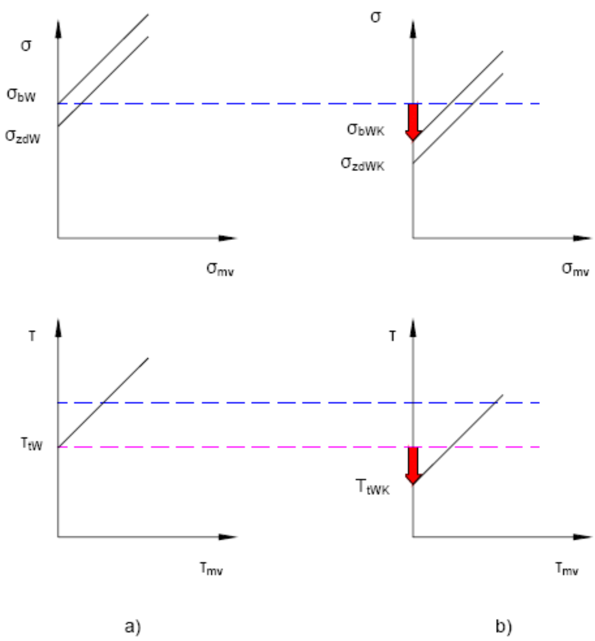

Fig. 3. Influence of the factors K_σ i K_τ to the dynamical stength
a) Dynamic strength of the probe
b) Dynamic strength of the shaft

$$\alpha_\sigma = \frac{\sigma_{\max K}}{\sigma_n} \qquad (17)$$

$$\alpha_\tau = \frac{\tau_{t\max K}}{\tau_n} \qquad (18)$$

where:
σ_{maxK} - maximal normal stress in the critical section of the shaft;
σ_n - nominal normal stress;
τ_{maxK} - maximal tangential stress in the critical section of the shaft;
τ_n - nominal tangential stress;
Factors of the stress concentration β_σ and β_τ are determined over the ration of dynamic stress of the probe and dynamic strength of the part – shaft/axle. Factors of the stress concentration are determined as (19) for pressure/tensile and deflection, and as (20) for the torque loads.

$$\beta_\sigma = \frac{\sigma_{zd,bW}\left(d\right)}{\sigma_{zd,bWK}} \qquad (19)$$

$$\beta_\tau = \frac{\tau_{tW}\left(d\right)}{\tau_{tWK}} \qquad (20)$$

where:
$\sigma_{zd,bW}(d)$ - dynamical strength for pure dynamical load pressure/tensile and deflection for probe with diameter d.
$\tau_{tW}(d)$ - dynamical strength for pure dynamical load torque for probe with diameter d.
$\sigma_{zd,bWK}$ - dynamical strength for pure dynamical load pressure/tensile and deflection for shafts and axles.
In some cases, factors of stress concentration β_σ and β_τ are experimental values. This is the case for pressed

connections and shafts with pins. In this situation huge impact to the factors of the stress concentration have corrective factors of the size $K_3(d)$ and $K_3(d_{BK})$.

3. PROGRAM MODULE FOR STRENGTH CALCULATIONS OF THE SHAFTS AND AXLES ACCORDING TO THE DIN 743

Calculation of the axles and shafts according to the standard DIN 743 is complex and iterative process requiring great number of iterative steps for solution searching.
Diagram of the calculations process is given in Figure 4.

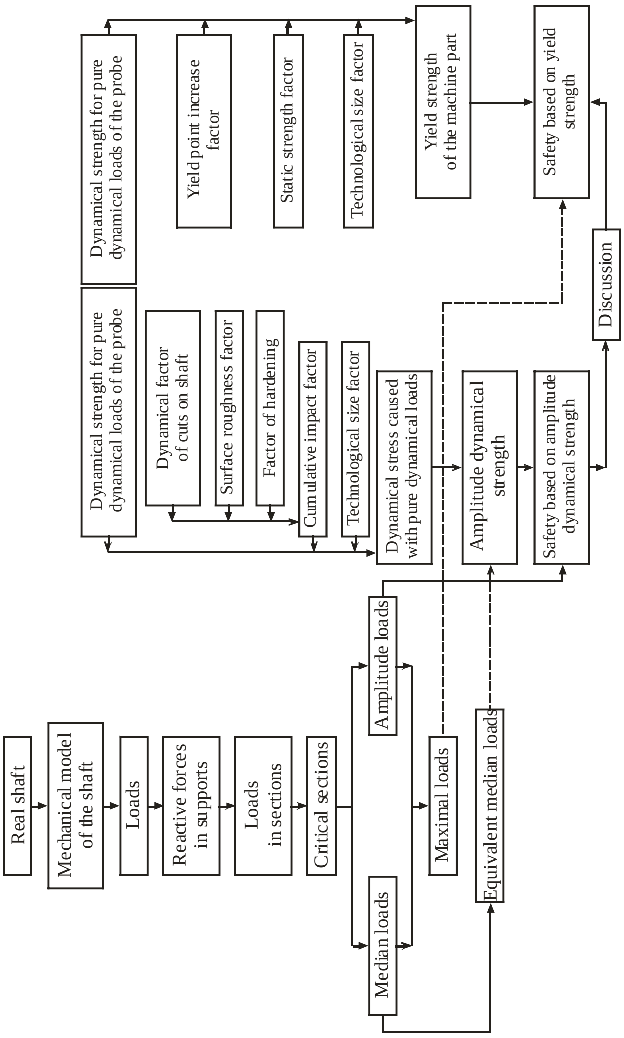

Fig. 4. Diagram of the calculations process

Figure 5 shows user interface and main menu of the program. User of the program can select type of the shaft and geometrical shape of the shaft. This includes variations and all engineering shapes.
After selection of the geometry for safety factor determination, dialog menu is opened and further parameters of the shaft can be determined. (Fig. 6).
After geometry definition, software calculates safety factor in critical area. New window and dialog box enable definition of parameters about type of loads and shaft material (Fig. 5).
User interface, shown in Figure 7, has three parts:

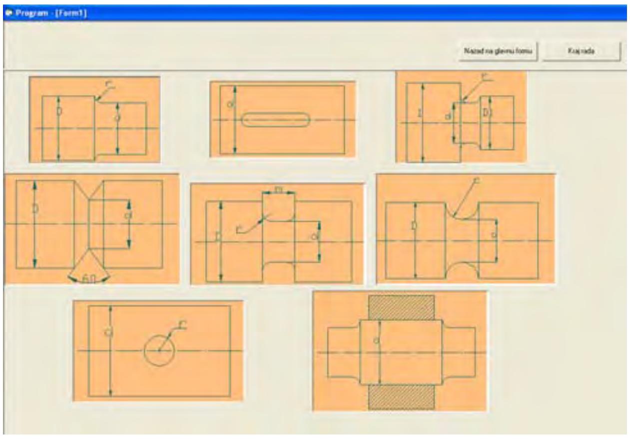

Fig. 5. Main menu

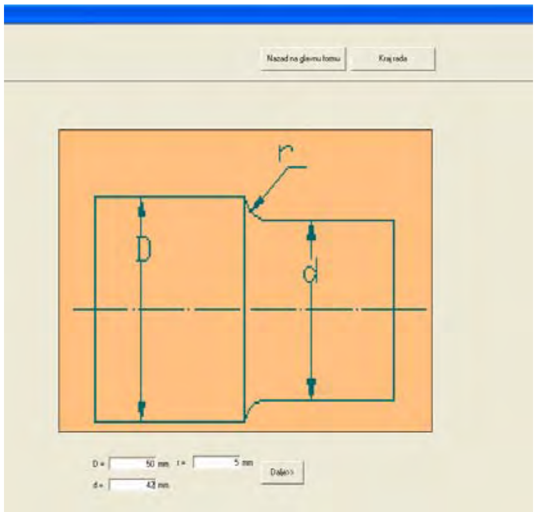

Fig. 6. Geometry definition - shaft

1. definition of the type of loads,
2. definition of the surface condition,
3. definition of the shaft material.

Selection of the load type gives possibility for user to choose pressing/tensile, deflection and torque. Values tht user can input are amplitude force (F_{zda}), median force (F_{zdm}), amplitude deflecting moment (M_{ba}), median deflecting moment (M_{bm}), amplitude torque moment (T_a) i median torque moment (T_m).

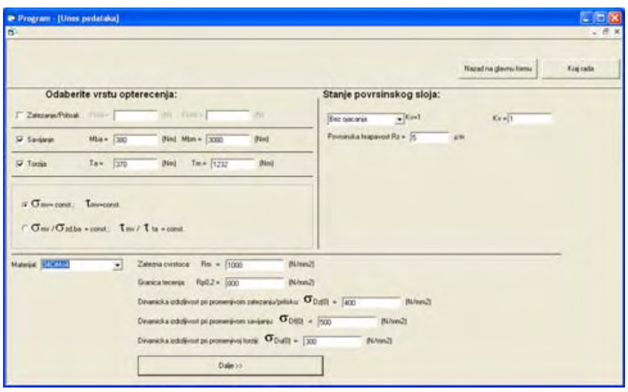

Fig. 7. Data Input

User can select case 1 (σ_{mv} = const., τ_{mv} = const.), or case 2 ($\sigma_{mv} / \sigma_{zd,ba}$ = const., τ_{mv} / τ_{ta} = const.)

Next step is definition of the surface condition. Options are: chemo – thermal processes, hardening, nitrating, carbo-nitrating, sand hardening etc.

It is necessary to input surface roughness R_z.

Definition of the materials comes from the data base, formed from DIN EN 10025, DIN EN 10113, DIN EN 10084, DIN EN 10083 and DIN 17211. For every material, program finds adequate ultimate stress, yield stress, dynamical strength for every type of recommended load.

Calculations give static and dynamic safety factors in selected critical section. Figure 8 gives part of the listing of output data from the calculation.

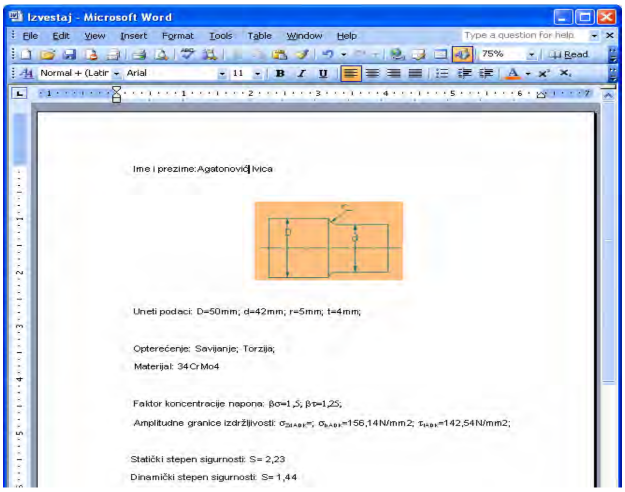

Figure 8. *Output Results*

CONCLUSION

Following observations can be carried out on the DIN method:

1) DIN 743 has proved as simply usable method for strength calculations of shafts and axles. Calculations about time-depended strength and specters of loads, standard does descript in tales 4 and 2, but this is not involved with this program module does not include it.
2) The DIN 743 method permits the accurate calculation of the safety factor in the case of superposed rigging, bending and torsion stresses.
3) This method is applicable for usage with the values of fatigue limit $\sigma_{zdW}(d_B)$, $\sigma_{bW}(d_B)$ and $\tau_{tW}(d_B)$ determined on the unnotched specimens or these ones given in the annexes of the DIN standard.
4) The method considers all the known influences on the fatigue limit of the shaft, using empirical calculation factors determined on the base of a great volume of experimental results.
5) The nominal stress values (i.e. σ_{ba}, σ_{zda} and τ_{ta}) that intervene in the expression (6) are uniform. These are calculated approximately without considering of real loading modelled with the load spectrum or the load sequence.
6) Program module for shaft and axle carrying strength, developed according the DIN 743 module is art of the program system PTD, developed at Faculty of Mechanical Engineering, Nis.

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FACULTY OF TECHNICAL SCIENCES WAS FOUNDED ON MAY 18TH 1960, AS FACULTY OF MECHANICAL ENGINEERING OF NOVI SAD AND WAS ORIGINALLY A PART OF THE UNIVERSITY OF BELGRADE . AFTER THE UNIVERSITY OF NOVI SAD HAD BEEN FOUNDED ON JUNE 28, 1960, THE FACULTY BECAME AN INTEGRAL PART OF THE UNIVERSITY OF NOVI SAD TOGETHER WITH SIX OTHERS FROM VOJVODINA. WITH ESTABLISHMENT OF THE DEPARTMENT OF ELECTRICAL ENGINEERING AND THE DEPARTMENT OF CIVIL ENGINEERING, THE FACULTY CHANGED ITS NAME INTO THE FACULTY OF TECHNICAL SCIENCES ON APRIL 22ND 1974. TODAY, THE FACULTY OF TECHNICAL SCIENCES IS THE BIGGEST FACULTY OF THE UNIVERSITY OF NOVI SAD AND A LEADER IN EDUCATION AND RESEARCH AS WELL AS IN THE IMPLEMENTATION OF THE BOLOGNA DECLARATION REFORMS. IT COVERS AN AREA OF 30,000 M2 OCCUPYING THE CENTRAL POSITION AT THE UNIVERSITY CAMPUS ON THE RIVER DANUBE.

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