

**Editorial Board**

**Editor in Chief:**

**Prof. Dr. Said I. Shalaby , MD, Ph.D.**  
 -Professor of Tropical Medicine & Infectious Hepato-Gastroenterology, Institute of Medical Research & Clinical Trials and Medical Center of Excellence ,National Research Centre(NRC), Egypt  
 -Former Deputy President of Academy of Scientific Research & Technology, Cairo, Egypt  
 -Former Chairman of Complementary Medicine Department, NRC, Egypt  
 -Former Editor -in -Chief of Bulletin NRC Springer Open  
 -WHO/FAO expert of Food born Parasites  
 -Convener of ARSO/ TC 03, Fish, fisheries and aquaculture WG 2: Sampling, laboratory analysis, testing, inspection and market compliance  
 -Overseas Advisor of the Academic Council of International Consortium of Contemporary Biologists, India  
 -Head of National Committee of Egyptian Standards for Medical Instruments as well as Head of National Committee for Fish & Fishery Products  
 -Head of Safety & Occupational Health Committee , NRC  
 e-mail : saidshalaby7@gmail.com  
 Cell Phone : +2 01223872074  
 YouTube Channel : <https://www.youtube.com/user/DrSaidShalaby/playlists>

**Editor:**

Dr. Mohammed Gulam Ahamad, PhD  
 Professor, Dept. of Computer Engineering,  
 College of Computer Engineering & Sciences  
 Prince Sattam Bin Abdulaziz University  
 Alkhafj, Saudi Arabia  
 ijترم@gmail.com , articles@ijترم.com

Mohammad Reza Kazemi, PhD  
 Professor, Dept. of Mathematical Statistics,  
 College of Mathematical Sciences  
 Fasa University, Fasa, Iran

Prof. A Dash, PhD  
 Professor, Department of Engineering & Management  
 Member of IEEE & British Science Association  
 Institute of Technology and Research, India  
 adash\_research@gmail.com  
 adash@ieee.org

Seetha P. B. Ranathunga, PhD  
 Department of Economics,  
 Professor of Social Sciences University of Kelaniya,  
 Kelaniya, 11600, Sri Lanka  
 seetha@kln.ac.lk

**Associate Editor:**

Milad Bahamirian, PhD  
 Professor, University of Applied Science and Technology  
 Iran  
 m.bahamirian@merc.ac.ir

Morteza Jafarpour, PhD  
 Professor, Vali-e-Asr University of Rafsanjan  
 Iran  
 m.j@vru.ac.ir

Arpit maru, PhD  
 Professor, SGSITS Indore  
 22arpitmaru@gmail.com

Dr. S Sureshkumar, Ph.D.  
 Professor, Dept. of Mathematics, Siddaganaga Institute of  
 Technology, Tumkur-572 103, Karanataka,  
 Country: India  
 ssk@sit.ac.in

**Member of Advisory Board:**

Duncan N. Irungu, PhD  
 Senior Professor, School of Business & Economics  
 Daystar University, Kenya  
 dirungu@daystar.ac.ke

Sallahuddin Hassan, PhD  
 Prof. School of Economics,  
 Finance and Banking UUM College of Business  
 Universiti Utara Malaysia  
 din636@uum.edu.my

Dr.G. Anandhi, PhD  
 Associate Professor  
 University of the People, California  
 Giri.Anandhi@uopeople.edu

Arash Khakzadshahandashi, PhD  
 Research and development expert at MAPNA group  
 Iran  
 khakzad\_a@mapnagroup.com

Dr. G.N.K. Suresh Babu, PhD  
 Professor, Computer Science  
 Acharya Institute of Technology, Bangalore – 560 090  
 sureshbabu@acharya.ac.in

Ebele Mary Onwuka PhD  
 Senior Lecturer, Department of Business Administration,  
 Nnamdi Azikiwe University Awka,  
 Anambra State, Nigeria  
 em.onwuka@unizik.edu.ng

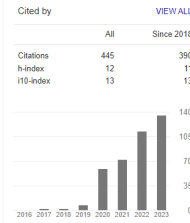
Dr. Ramachandran Guruprasad, PhD  
 Senior Technical Officer, CSIR-National Aerospace Laboratories,  
 Old Airport Road, Near Manipal Hospital,  
 Bangalore – 560 017  
 gprasada61@gmail.com

Siamak Hoseinzadeh, Ph.D  
 Professor, Mechanical Technical adviser for Incineration Plant,  
 Municipality of Sari, Mazandaran, Iran,  
 hoseinzadeh.siamak@gmail.com

Reza Fotuhi, PhD  
 Member of Electrical and Electronics Engineers (IEEE)  
 Iran  
 Fotuhi@ieee.org

Gectanjali Kale, PhD  
 Assistant Professor,

**GOOGLE SCHOLAR**



**IMPACT FACTOR**

Scientific journal impact factor : **6.736**  
 Cosmos Impact Factor:4.52  
 International Journal Impact Factor:2.1  
 Cite factor Impact Factor : 1.77  
 Infobaseindex: 1.5  
 I2OR Impact Factor:0.864

**PAPER SUBMISSION**

All Submissions through email only at  
[ijترم@gmail.com](mailto:ijترم@gmail.com)

**DOWNLOADS**

- > ijترم manuscript template
- > ijترم copyright form
- > ijترم\_Certificate
- > ijترم editorial board



**LICENSE**

*UJTRM is licensed under a Creative Commons Attribution 4.0 International License.*

Computer Engineering,  
Pune Institute of Computer Technology,  
gvkale@pict.edu

Abdullah Cakan,  
MS Mechanical Engineer  
Selcuk University  
Mechanical Engineering Department,  
Konya, Turkey  
acakan@selcuk.edu.tr

Giosuè Boscato, Ph.D.  
IUAV University of Venice  
Laboratory of Strength of Materials (LabSci)  
Via Torino 153/A – 30173 Mestre, Venice, Italy  
gboscato@iuav.it

Abhineet Anand, PhD  
Professor, School of Computer Science Engineering, Galgotias University,  
Greater Noida, G B Nagar, UP  
Abhineet.anand@galgotiasuniversity.edu.in;

Dr Gnaneswara Rao Nitta, PhD  
Professor of CSE Vignana's Foundations for Science, Technology  
and Research University, Vadlamudi, Guntur, AP  
nrg\_cse@vignamuniversity.org

Ranjit Shreshtha, PhD  
Professor, Division of Mechanical Engineering,  
Kongju National University,  
Cheonan 331-717, Korea  
sfranji@kongju.ac.kr

Dr Farhad Shafiepour Motlagh  
Professor, Mahallat Branch, Islamic Azad University,  
Mahallat, Iran  
farhad\_shafiepour@yahoo.com

Ezema Chukwuedorjie Nnaemeka, PhD  
Department of Electronic & Computer Engineering  
Nnamdi Azikiwe University Awka  
Anambra State, Nigeria  
ecnaxel@gmail.com

Prof (Dr) Prasanna B M R, Ph.D.  
Professor, Dept. of Mathematics, Siddaganga Institute of Technology,  
B. H. Road, Tumkur-572 103, Karnataka, India  
bmrp@sit.ac.in

Gad-Elkareem Abdrabou Mohamed, PhD  
Ass.Prof. at National Research Institute of Astronomy and Geophysics, Earthquakes department,  
Helwan- Cairo – Egypt  
gadElkareem@nriag.sci.eg

Prof. Vivek Mishra  
Dean, Sri Sharda Group of Institutions  
(Affiliated to Lucknow University, AKTU & BTE, U.P.)  
Lucknow, Uttar Pradesh, India  
visarit@gmail.com

**IMPACT OF THE INPUT SHAFT NUMBER OF REVOLUTIONS ON THE GEAR PAIR TEETH BENDING STRESS IN A PLANETARY REDUCER**

**Sasko Milev<sup>\*1</sup>**  
**Darko Tasevski<sup>2</sup>**  
**Mevludin Shabani<sup>3</sup>**  
**Lazar Jovanov<sup>4</sup>**  
**Zoran Dimitrovski<sup>5</sup>**

<sup>\*1,5</sup> Faculty of Mechanical Engineering, Goce Delcev University, Stip, North Macedonia  
Krste Misirkov, 10A, 2000, Stip; email: <sup>1</sup>sasko.milev@ugd.edu.mk ; <sup>5</sup>zoran.dimitrovski@ugd.edu.mk

<sup>2</sup> Faculty of Mechanical Engineering, Ss Cyril and Methodius University in Skopje, North Macedonia

<sup>3</sup> Faculty of Mechatronics, University of Business and Technology, Prishtina, Kosovo;  
email: mevludin.shabani@ubi-uni.net

<sup>4</sup> Ruen-Inox Automobile, Kocani, North Macedonia

---

**ABSTRACT**

This paper aimed to examine the possibilities to improve the characteristics of planetary reducers, considering their wide application in mechanical engineering. For this purpose, it was first investigated how the change in the speed of the input shaft of the planetary reducer affects the bending stress in the roots of the teeth of the gear pairs that are formed by the teeth of the driving shaft and the satellite gears, at a constant number of teeth and a constant transmission ratio of the planetary gear. Then it was investigated whether and how the change in the number of teeth on the driving shaft affects the bending stress of the teeth.

**Keywords:**

Planetary Reducer, Bending Stress, Gear Pair Teeth, Transmission Rate,

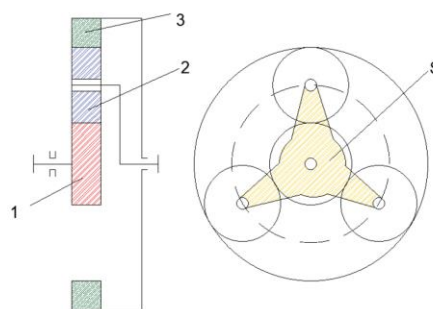
---

**INTRODUCTION**

Mechanical transmission is transfer of power or motion from the driving to the driven element through a mechanical assembly called a transmission. Transmissions consist of driving and driven rotate elements, transfer is achieved through their direct or indirect contact, for example, using a gear pair, belt, chain. Changing the number of rotations is accomplished by altering the diameter of the driving and the driven elements. Main characteristic of mechanical transmissions is their transmission ratio. When transmission ratio is greater than 1 ( $i > 1$ ), the number of rotations decreases, the torque increases, and the transmission is called a reducer. Conversely, when the transmission ratio is less than 1 ( $i < 1$ ), the number of rotations increases, the transmission is called a multiplier. Planetary transmissions belong to the group of mechanical transmissions, i.e., gear mechanical transmissions and the mechanical transfer is achieved by at least one element (the satellite gear), rotating not only around its own axis but also around another axis. These mechanical transmissions find application when is a need for transfer of large forces and a high number of rotations, with the imposed requirement for a small volume and weight of the transmission. In planetary transmissions, the force is distributed among multiple planetary gears [1] [2].

Advantages of planetary transmissions in relation to non-planetary ones are; high compactness expressed through small overall dimensions and low volume, low peripheral speeds, and thus a small level of noise, small mass, two or three times less than that of non-planetary transmissions, using the principle of separation of torque (between several planetary gears), very high efficiency–highest of all transmissions, relatively good cinematic abilities, relatively low noise level due to: lower

peripheral speeds, smaller dynamic forces due to smaller dimensions, high transmission ratio within one degree, etc. Although the large number of advantages, there are also disadvantages: geometric calculations are complex, the uneven distribution of load between planets is a complex and specific problem whose solution is not easy, the teeth of the planetary gears are rolled less favorably than in other gears, they have a complex construction, and a high risk of a defect of the entire transmission, high production technology is needed (due to the imposed quality requirements). Planetary transmissions have a number of capabilities in solving certain tasks in the field of power transmission: 1. Achieving a constant transmission ratio  $i = \text{const}$  can work as a reducer -  $|i| > 1$  (with one degree of freedom), it is most commonly used when the ring is fixed, i.e.  $\omega_3 = 0$ , less often when  $\omega_1 = 0$ , and rarely when  $\omega_s = 0$ , and as a multiplier -  $|i| < 1$  (with one degree of freedom) it is commonly used when  $\omega_3 = 0$  in hydro turbines, less often when  $\omega_1 = 0$ , and rarely when  $\omega_s = 0$ . 2. Implementation of stepwise change of transmission ratio in single and double drives. 3. Implementation of continuous change in transmission ratio, differential (with two degrees of freedom), achieving a high angular velocity and other application. The designation of planetary gear systems is done using the numbers 1 and 2, and the letters A and I. The numbers 1 and 2 indicate whether the gear system is single-stage or two-stage, while the letter A denotes central gears with external teeth (sun gear), and the letter I denotes central gears with internal teeth (ring gear). Simple planetary gear systems are of type 1AI, and they have a single-stage planetary set with one central gear with external teeth (sun gear) and one central gear with internal teeth. Planetary gear systems can be used as reducers, multipliers (with one degree of freedom), and differentials (with two degrees of freedom), depending on their construction [3].



**Fig. 1: Planetary gear train 1AI**

This planetary gear system consists of the following main components: central gear with external teeth 1 (sun gear), planetary gears 2, central gear with internal teeth 3 (ring gear), and carrier S. The central gear with external teeth is the driving element and is attached to the input shaft. It is coupled with the planetary gears, which perform two rotations, one around their axis and the other around the axis of the central gears. The central gear with internal teeth is in contact with the planetary gears but does not undergo rotational movement as it serves as a reaction member in this case. The carrier S is attached to the shafts of the planetary gears, and the output shaft is inserted into it, transmitting rotational movement to the output shaft as a result of the rotation of the planets around the axes of the central gears. Since this is a reducer, the number of rotations of the output shaft will be less than the number of rotations of the input shaft.

### OBJECTIVES

The main objective of the study is to determine the impact of the change of the number of revolutions of the planetary reducer input shaft on the gear pair ( $z_1-z_2$ ) teeth bending stress, when the number of driving shaft teeth is unchanged. Then the impact of the change of the number of driving gear teeth on the bending stress of the of the gear pair teeth is analyzed.

**METHODOLOGY****Kinematic analysis of 1AI planetary gear train**

The kinematic analysis is performed with the superposition method-Swamp rule. According to this method, individual elements of the planetary transmission are observed in different specified movements in few set movements. Number of rotations and the number of revolutions of the planetary reducer gears are given in Table 1 [4].

**Table 1: Kinematic Analysis of Planetary Gear Train**

Number of movements	Gear 1 ( $z_1$ )	Gear 2 ( $z_2$ )	Gear 3 ( $z_3$ )	Carrier S
1	+1	+1	+1	+1
2	-1	$\frac{z_1}{z_2}$	$\frac{z_1}{z_2} * \frac{z_2}{z_3} = \frac{z_1}{z_3}$	0
Sum of the first two motions	0	$1 + \frac{z_1}{z_2}$	$1 + \frac{z_1}{z_3}$	+1
3	$+n_1$	$-\frac{z_1}{z_2} * n_1$	$-\frac{z_1}{z_3} * n_1$	0
4	0	$\left(\frac{z_1}{z_2} + 1\right) * n_s$	$\left(\frac{z_1}{z_3} + 1\right) * n_s$	$+n_s$
Sum of the 3 <sup>rd</sup> and 4 <sup>th</sup> motions	$+n_1$	$-\frac{z_1}{z_2} * n_1 + \left(\frac{z_1}{z_2} + 1\right) * n_s$	$-\frac{z_1}{z_3} * n_1 + \left(\frac{z_1}{z_3} + 1\right) * n_s$	$+n_s$

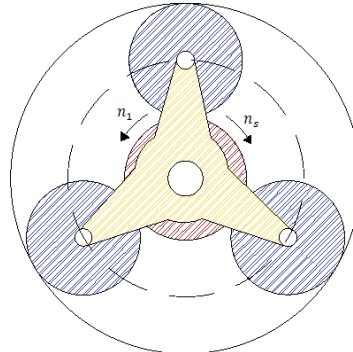
The number of rotations of gear 1:  $n_1 = n_{\text{input}}$

The number of rotations of carrier s:  $n_s = n_{\text{output}}$

From the kinematic analysis (Table 1), expressions are derived to determine the number of rotations of gear 2 (planetary gear) and the number of rotations of gear 3 (ring gear)

$$n_2 = -\frac{z_1}{z_2} * n_1 + \left(\frac{z_1}{z_2} + 1\right) * n_s; \quad n_3 = -\frac{z_1}{z_3} * n_1 + \left(\frac{z_1}{z_3} + 1\right) * n_s;$$

The overall transmission ratio can be obtained from the ratio of the number of rotations at the input and output of the planetary gear train. The direction of rotation of the output shaft, is opposite to the direction of rotation of the input shaft, where the central gear with external teeth (sun gear) is located.



**Fig. 2 : Direction of rotation of the carrier  $S$  and gear  $1$**

Since the central gear with internal teeth (ring gear) will be the reaction member, its number of rotations will be equal to zero. The overall transmission ratio will be:  $i = n_1 / n_2$

#### Coaxiality condition

Using coaxiality condition, which guarantees equal spacing between the first set (sun gear and planetary gears) and the second set (planetary gears with ring gear) the number of teeth on the central gear with internal teeth is:

$$z_3 = z_1 + 2z_2$$

In this expression, it is assumed that the gears will have straight teeth and no profile shifting ( $x=0$ ). The number of teeth on the central gear with external teeth (sun gear) and planetary gears are chosen arbitrarily. Due to the high number of rotations at the input, it is preferable to choose a smaller number of teeth of sun gear to reduce the size and, consequently, the mass of the gear. Once we have the numbers of all teeth, we can establish the transmission ratios between individual pairs of gears. In the case where the direction of rotation of the driven gear is opposite to the direction of rotation of the driving gear, a negative sign is assigned to the transmission ratio [5].

The transmission ratio between gear 1 (sun gear) and gear 2 (satellite gears) is:

$$i_{12} = -\left(\frac{z_2}{z_1}\right)$$

The transmission ratio between gear 2 (planetary gears) and gear 3 (ring gear) is:

$$i_{23} = \frac{z_3}{z_2}$$

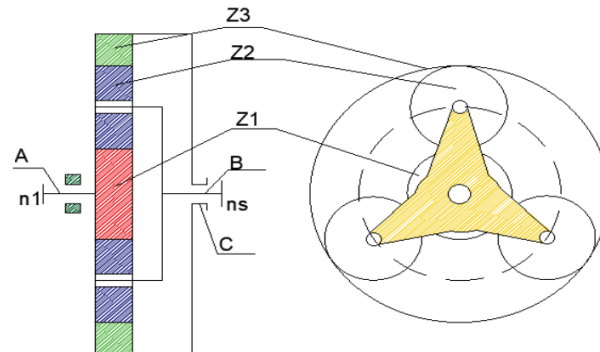
Hence, the actual standard transmission ratio is:

$$i'_0 = i_{12} * i_{23}$$

#### Sizing of the gears

The total power transmitted by the planetary transmission consists of two components: the power at the coupling  $P_k$  and the force due to the rolling of the teeth  $P_z$ , also known as kinematic power [4].





**Fig. 3: Shafts of the planetary transmission**

Input shaft is shaft A

$$P_A = \omega_A * T_A, [\text{W}]$$

Angular velocity of the shafts:

$$\omega_A = n_A * \frac{\pi}{30} [\text{s}^{-1}]$$

$$\omega_B = n_B * \frac{\pi}{30} [\text{s}^{-1}]$$

$$\omega_C = n_C * \frac{\pi}{30} [\text{s}^{-1}]$$

Torque of the input shaft A:

$$T_A = \frac{P_A}{\omega_A}$$

Torque of shaft B:

$$T_B = -(T_A + T_C) = -(1 - i_0 * \eta_0) * T_A$$

Torque of shaft C:

$$T_C = -i_0 * \eta_0 * T_A [\text{Nm}]$$

Due to the division of the power on  $P_k$  and  $P_z$ , the power on the shaft A is:

$$\begin{aligned} P_{ZA} &= \omega_{AS} * T_A \\ P_{ZA} &= (\omega_{A0} - \omega_{S0}) * T_A \\ P_{KA} &= \omega_{S0} * T_A [\text{W}] \end{aligned}$$

The powers on shaft B are:

$$\begin{aligned} P_{ZB} &= \omega_{BS} * T_B = (\omega_{B0} - \omega_{S0}) * T_B \\ P_{KB} &= \omega_{S0} * T_B \end{aligned}$$

The powers on shaft C are:

$$\begin{aligned} P_{ZC} &= \omega_{CS} * T_C = (\omega_{C0} - \omega_{S0}) * T_C \\ P_{KC} &= \omega_{S0} * T_C \end{aligned}$$

### Calculation of the orientation module of the gear pair $z_1 - z_2$

Module of the gear pairs  $z_1 - z_2$  is calculated using the formula [6]:

$$m \geq \sqrt[3]{\frac{2 * T_1}{\lambda * z_1 * \sigma_{FP}} * Y_{Fa} * Y_{Sa} * K_{F\alpha} * K_{F\beta} * K_v * K_a}$$

$Y_{Fa}$  - tooth shape factor,  $Y_{Sa}$  - stress correction factor,  $K_{F\alpha}$  - load distribution factor in the frontal cross-section for stress at the tooth root,  $K_{F\beta}$  - load distribution factor along the length of the tooth for stress at the tooth root,  $K_v$  - distribution factor of the peripheral force along the engagement of central gears with planetary gears,  $K_a$  - factor for uniform loading of the machine in continuous operation with the electric motor,  $\lambda$  - tooth width factor and  $\sigma_{FP}$  - root bending stress [6]. As a material for the manufacture of the central gear with external teeth, it is choiced induction-hardened steel.

Because there are three planetary gears in the planetary transmission, the total power is divided into 3 parts:

$$P_1 = \frac{P_{ZA}}{N}$$

The torque at the point of contact between gear 1 and one of the planetary gears 2 will be:

$$T_1 = \frac{P_1}{\omega_1}$$

### Calculation of the orientation module of the gear pair $z_2 - z_3$

Because this planetary transmission is of the 1AI type, all gears need to have the same module to achieve mutual engagement. It was necessary to calculate the orientation module of the gear pair, and then a module that meets the requirements for both gear pairs was selected.

We follow the same procedure as with the gear pair  $z_1 - z_2$ , so the next step is [6]:

$$m \geq \sqrt[3]{\frac{2 * T_2}{\lambda * z_2 * \sigma_{FP}} * Y_F * Y_\epsilon * K_{H\alpha} * K_{H\beta} * K_v * K_I}$$

$K_{H\alpha}$  - transverse load factor for contact stress,  $K_{H\beta}$  - face load factor contact stress

Because there are 3 planetary gears in the planetary transmission, the total power is divided into 3 parts:

$$P_2 = \frac{P_{ZC}}{N}$$

The torque at the point of contact between gear 3 and one of the planetary gears 2 will be:

$$T_2 = \frac{P_2}{\omega_2}$$

Where the angular velocity of the planetary gear is  $\omega_2$ , and it is:

$$\omega_2 = \frac{\pi * n_2}{30}$$

### Dimensions of the gears

The width of the teeth is determined from the expression for the tooth width factor.

The width of the teeth is  $\lambda = b/m$

The center distance between the two gear pairs is equal and is:

$$a_{12} = a_{23}, \quad a_{12} = \frac{(z_1 + z_2) * m}{2}$$



Diameters of gear 1 (central gear with external teeth)

Reference diameter:  $d_1 = m * z_1$ , Tip diameter:  $d_{a1} = d_1 + 2m$ , Root diameter:  $d_{f1} = d_1 - 2,5m$

Diameters of gear 2 (planetary gears)

Reference diameter:  $d_2 = m * z_2$ , Tip diameter:  $d_{a2} = d_2 + 2m$ , Root diameter:  $d_{f2} = d_2 - 2,5m$

Diameters of gear 3 (central gear with internal teeth)

Reference diameter:  $d_3 = m * z_3$ , Tip diameter:  $d_{a3} = d_3 + 2m$ , Root diameter:  $d_{f3} = d_3 - 2,5m$

### Inspection of teeth root bending stress

Gear pair  $z_1 - z_2$

The teeth root bending stress is calculated according to the equation [7]:

$$\sigma_{F1} = \frac{F_{tw1}}{b * m} * Y_{Fa} * Y_{Sa} * Y_B * K_{F\alpha} * K_{F\beta} * K_v * K_a$$

The force  $F_{tw1}$  is:

$$F_{tw1} = \frac{2 * T_1}{d_{w1}}$$

Gear pair  $z_2 - z_3$

$$\sigma_{F2} = \frac{F_{tw2}}{b * m} * Y_{Fa} * Y_{Sa} * Y_B * K_{F\alpha} * K_{F\beta} * K_v * K_a$$

The force  $F_{tw2}$  is:

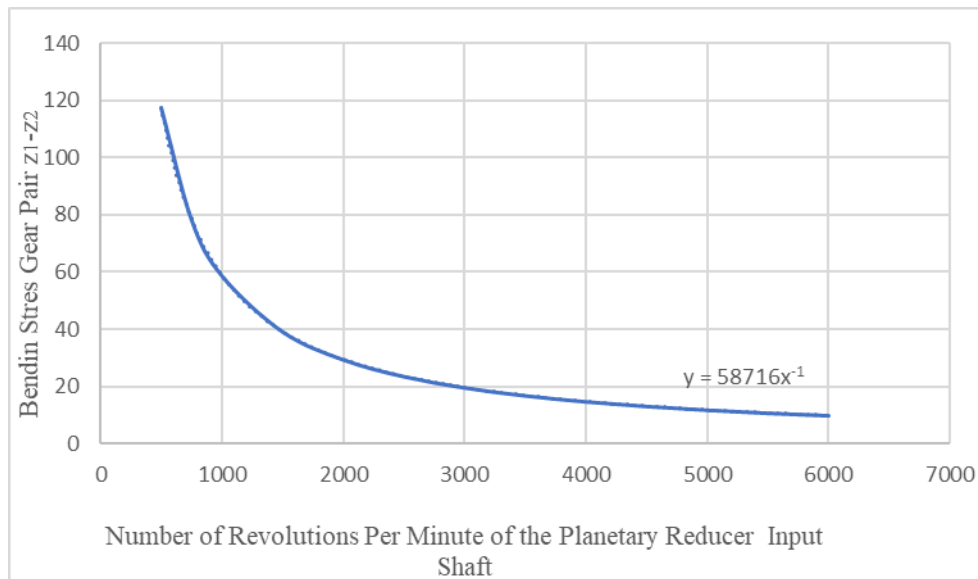
$$F_{tw2} = \frac{2 * T_2}{d_{w2}}$$

### RESULTS AND DISCUSSION

For the given values  $z_1=25$ ,  $z_2=85$  and  $z_3=195$ ,  $P=4.000[W]$  the planetary reducer transmission ratios were first calculated. Then, bending stresses in the roots of the teeth of the gear pair  $z_1-z_2$  were calculated for different numbers of revolutions of the input shaft, for a number of revolutions from 500 to 6,000 revolutions ( $z_2$  and  $z_3$  are not changed). Obtained values for  $z_1=25$  are shown in table 2. Based on the values from table 2 is drawn the diagram presented in the Figure 4. This curve actually determines the gear pair  $z_1-z_2$  bending stress change depending on the number of revolutions of the input shaft  $n_1$ . The equation of this curve is  $y=58716/x$  and it was calculated using approximation method.

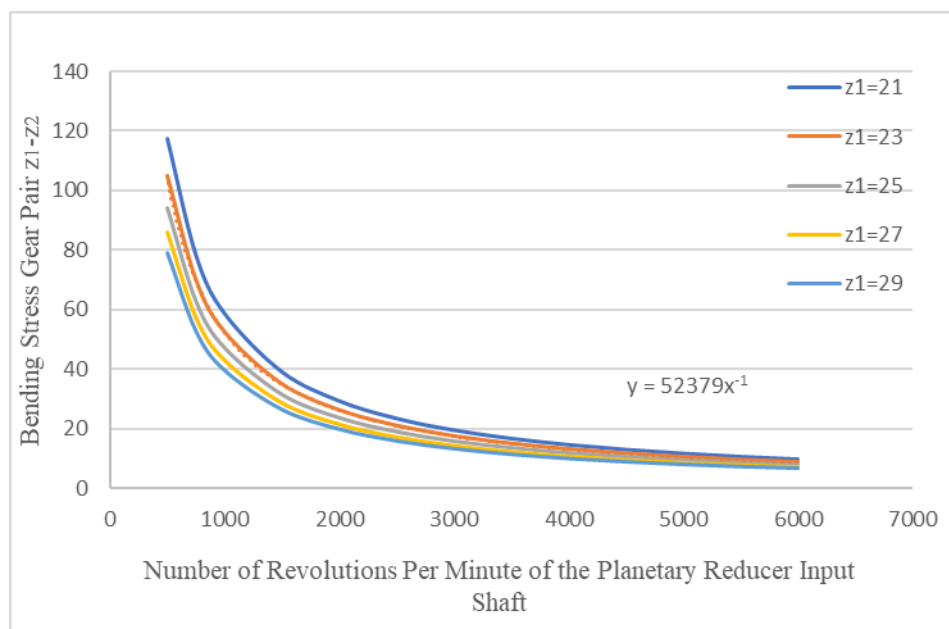
**Table 2: Bending stress of teeth of gear pair  $z_1-z_2$**

$Z_1$	Number of revolutions per minute of the Input Shaft of Planetary Reducer												
	500	750	1000	1500	2000	2500	3000	3500	4000	4500	5000	5500	6000
21	117.57	78.38	58.78	39.19	29.39	23.52	19.59	16.80	14.70	13.06	11.77	10.69	9.8
23	104.75	69.83	52.38	34.92	26.19	20.95	17.46	14.96	13.09	11.64	10.48	9.52	8.73
25	93.99	62.66	46.99	31.33	23.50	18.80	15.66	13.43	11.75	10.44	9.4	8.54	7.83
27	85.89	57.26	42.94	28.63	21.47	17.18	14.31	12.27	10.74	9.54	8.59	7.81	7.16
29	78.91	52.6	39.45	26.3	19.73	15.78	13.15	11.27	9.86	8.77	7.89	7.17	6.58



**Fig. 4: Bending Stress of the Gear Pair ( $z_1-z_2$ ) Teeth for Different Number of Revolutions of the Planetary Input Shaft**

Next calculations were made for the cases when the number of teeth on the driving shaft is changed from  $z_1=21$ ,  $z_1=23$ ,  $z_1=27$  to  $z_1=29$ . For each of these cases the bending stresses were calculated for different input shaft numbers of revolutions from 500 to 6000 revolutions per minute.



**Fig. 5: Bending Stress of the Gear Pair ( $z_1-z_2$ ) Teeth for Different Number of Revolutions of the Planetary Input Shaft for Different Values of  $z_1$**

### CONCLUSION

With a constant number of teeth in the planetary reducer (constant transmission ratio) and constant power of the input shaft, with an increase in the number of revolutions of the input shaft, the bending stress in the roots of the teeth of the gear pair  $z_1$ - $z_2$  decreases. The degree of bending stress reduction for a lower number of revolutions (up to 1000 revolutions per minute) is the greatest. With the further increase in the number of revolutions of the input shaft, the reduction of the bending stress on the teeth continues but with a reduced intensity. For the number of revolutions greater than 4000 revolutions per minute the reduction of the bending stress is insignificant. This fact can be used in cases where it is necessary for a certain planetary reducer to reduce the bending stress at the roots of the teeth.

From the Diagrams in Figure 5, it can be seen that the reducing the bending stress in the teeth ( $z_1=21,23,27,29$ ) also applies upon changing the number of teeth of the drive gear. By increasing the number of teeth of the drive gear, the bending stress in the teeth for the same number of revolutions of the input shaft decreases and vice versa.

### REFERENCES

- [1] Meriam J., Kraige L., Bolton J, Wiley (2018) *Engineering Mechanics*, Volume 2, Dynamics
- [2] Stamboliev D., Ss.Cyril and Methodius University in Skopje, Macedonia ((1976) *Prenosnici na vozilata*
- [3] Petrovic T, Ivanov T and Milosevic M, , *Forsch Ingenieures* (2009), *A new structure of combined gear trains with high transmission ratios*
- [4] Arnaudov K., Karaivanov D., CRC Press, Taylor & Francis Group (2018), *Planetary Gear Trains*
- [5] Simeonov S., Milev S., Goce Delcev University Stip (2019), *Praktikum po Masinski elementi*
- [6] Simeonov S., Goce Delcev University Stip (2017), *Masinski elementi*
- [7] Budinas R., Nissbet J., Mc Graw Hill Education (2020) *Shigley's Mechanical Engineering Design*
- [8] Jiang W., Wiley (2019). *Analyses and Design of Mechanical Elements*.