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INTELLIGENT USE OF ISO AND AGMA GEAR STANDARDS FOR COST EFFECTIVE SPUR GEAR DESIGN

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Keywords
Gear design,
Gear standards,
Gear rating,
Dimensionless number,
Cost effective design.

Abstract
ISO and AGMA standards provide the most accurate and commonly used spur gear design approaches but the design results obtained from both are differing from each other even under the same input parameters. The selected design approach has a significant influence on the results, therefore, if the design approaches are not rated for gear designs, the designers are not aware of the loss or gain on the cost and failure or success of the design. This paper uses ISO and AGMA gear standards to carry out spur gear designs considering the allowable range of gear speed ratios, transmitted power combinations and the failure conditions like bending and surface contact fatigue. These wide ranges of considerations which cover almost the most design applications in industrial practice allow to rate the design results obtained from both standards. The systematic method available in this study is generic and meets a need to select an appropriate gear standard by introducing dimensionless, "Geometric Rating Numbers, (GR_i)". The practical curves and charts help designers to select cost effective, interference-free spur gear design approach for a particular gear design.

MALIYET ETKIN DÜZ DIŞLI ÇARK TASARIMI GERÇEKLEŞTIRMEK IÇIN ISO VE AGMA STANDARTLARINDAN UYGUN OLANININ KULLANIMI

Anahtar Kelimeler
Dişli çark tasarımı,
Dişli çark standartları,
Dişli çarkların göreceli kıyaslanması,
Boyutsuz sayılar,
Maliyet etkin tasarım.

Öz
ISO ve AGMA standartları dişli çark tasarımında en yaygın olarak kullanılan ve doğru sonuçlar veren tasarım yaklaşımlarıdır. Ama aynı çalışma parametreleri altında dahi elde edilen tasarım sonuçları birbirlerinden farklılık göstermektedir. Tasarım için seçilen yaklaşım sonuçlar üzerinde önemli etkiye sahiptir. Bu sebeple, eğer tasarım yaklaşımları birbirlerine göre kıyaslanmaz ise tasarımcılar üzerinde maliyetten kazanç veya kayıp hakkında veya tasarımın başarılı veya başarısız olması ihtimali hakkında farkındalık oluşmayacaktır. Bu çalışma, ISO ve AGMA standartlarını kullanarak düz dişli çark tasarımları gerçekleştirir. Tasarımlarda dişli hız oranları, aktarılan güç değerleri, ve yüzey temas ve eğilme gerilmeleri gibi dişli çarklar yorulma gerilmeleri de dikkate alınmıştır. Bu geniş yelpazede değerlendirilen faktörler sayesinde elde edilen sonuçlar hemen hemen birçok endüstriyel alanda ihtiyaç duyulan uygulamaları kapsayacak niteliktedir. Çalışmada sunulan sistematik metot kendine özgü olup uygun tasarım yaklaşımı seçmeyi sağlar. Bunu "Geometric Rating Number (GRi)" adı verilen boyutsuz sayılar türeterek yapmaktadır. Elde edilen pratik eğriler ve grafikler tasarımcıya maliyet etkin, doğru tasarımı ortaya koyma noktasında yol gösterici niteliktedir.

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1. Introduction

Module (m) and face width (b) which determine the overall size of a gear are the most essential parameters for carrying out a gear design. But, finding these parameters requires iterative calculations that are considerably time consuming and dependent on expertise to find out the design outputs as selecting a proper module and determining the face width. For this reason, various design approaches including national, international standards and machine design textbooks provide a large number of formulae to perform a gear design with different level of difficulty and each one serves different results interestingly. As the results obtained from the approaches differing from each other, the selection of an appropriate design approach becomes more important for cost effective gear design. Because the cost is directly depended on overall size of a gear.

Most of the studies on gears have been concentrated on analysing the gear stresses such as decreasing bending and/or surface contact stresses. Finite Element Analysis (FEA) was used to analyse gear stresses to compare and verify the analytical results with numerical solutions (Gupta et al., 2012; Jebur et al., 2013; Tiwari et al., 2012; Karaveer et al., 2013; Shinde et al., 2009; Fetvacı et al., 2004). These studies showed that results of theoretical calculations have been in a good agreement with the results of FEA. Gear stresses have also been decreased by making profile modifications on gear tooth (Huang and Su, 2010; Li, 2007; Pedersen et al., 2010; Cavdar et al., 2005; Parthiban et al., 2013; Sankar and Natarai, 2011; Markovic and Franulovic, 2011).

Few studies were also made on the design of gears using expert systems which aim to find optimum design parameters (Geren and Baysal, 2000; Li et al., 2002, Gologlu and Zeyveli, 2009; Mendi et al., 2010).

Some efforts were spent on investigating the ratings of gear design standards. Cahala (1999) stated that American Gear Manufacturers Association (AGMA) 2001 (or the metric AGMA 2101) and International Organization for Standardization (ISO) 6336 produced significantly different gear ratings for both strength and pitting resistance. A translation technique between ISO and AGMA standards was tried by considering the principles of approaches including material effect, gear quality number and calculation methods. But the comparison was made on a narrow perspective investigating only with six test sample units consist of having different gear center distance, face width and two different gear reduction ratio. Cahala and Uherek (2007) performed ring gear set design by comparing AGMA 6014 and AGMA 321. However, the data requirements for gear design were mentioned with limited input parameters. Beckman and Patel (2000) compared ISO (1996), Deutsches Institut für Normung (DIN), American Petroleum Institute (API) and AGMA for high speed (1998) and low speed (1997) individual gear design results with each other. They highlighted the importance of understanding the different rating systems for the price and reliability of the gearbox. Li (2007) calculated gear stresses by using Japanese Gear Manufacturers Association (JGMA) and ISO standards, Kawalec et al. (2006), and Kawalec and Wictor (2008) made comparative analysis of tooth root strength by using ISO and AGMA standards then compared to verify the results with FEA.

The designs of gears have been performed widely in the literature. But the intelligent use of an appropriate gear design approach for casual or experienced designers has not been introduced yet. Because the most important design parameters, module (m) and face width (b), have not still been rated for the approaches. As the design approaches are not rated for gear tooth volume, the designers are not aware of the loss or gain on the cost and failure or success of the design. Therefore, this study aims to make a comprehensive comparison of the results obtained from ISO and AGMA. For this, (m) and (b) values are generated by the following considerations: (i) allowable gear speed ratios from 1:1 to 8:1, (ii) power transmitted values from 0,5 kW to 1000 kW, (iii) gear fatigue failure conditions that are bending and surface contact. These wider ranges for considerations cover almost the most design applications in industrial practices. Then the rating of the results obtained from standards was performed via a systematic methodology and achieved by introducing dimensionless, "Geometric Rating Numbers, (GR_i)". It shows that the approach used for the intelligent use of gear standards for cost effective spur gear design is satisfactory. The generic method now helps designers for selecting appropriate design approach while designing spur gears based on fatigue failure conditions such as either bending or surface contact.

ISO 6336 and 9085:2002 standards were used for bending and surface contact fatigue, respectively and AGMA 2101-D04 Standard used for both bending and surface contact fatigue failure criteria.

2. A General Systematic Methodology

This study introduces a general systematic method for carrying out the design of a spur gear and making a comparison of design results obtained from both AGMA and ISO standards. Figure 1 shows a flowchart including the comparison method and design steps for designing a spur gear. And Figure 2 shows combinations of gear speed ratio and power transmitted values for the two gear design principles.

As it is seen from Figure 1, design outputs, module selection and face width determination require iterative calculations which start with an estimation of initial module and it is iterated until face width is between 3p and 5p where p (π.m) is the circular pitch that is dependent on the selected module (Budynas and Nisbett, 2011). Based on the gear design standards, spur gears were designed considering both bending and surface contact fatigue failure criteria occured in tooth root as breakage or pitting on surface of the tooth, respectively.

Spur gears have an allowable range of speed ratio (m_G) from 1:1 to 8:1 and 21 power transmitted values were selected from 0,5 kW to 1000 kW considering the power of standard electric motors (Icarus Reference, 1998). The selected speed ratios and transmitted power values may cover the most of the design applications in practice, and allow to make reliable conclusions theoretically.

2.1. Material Selection for Gear Design

During the design of a gear box, the properties of pinion and gear materials must be in a good agreement for proper design because the mechanical properties of materials have to satisfy all service conditions.

The combination of a steel pinion and cast iron gear represent a well-balanced design for the comparison. Because cast iron has low cost, ease of casting, good machinability, high wear resistance, and good noise abatement. Cast iron gears typically have greater surface fatigue strength than bending fatigue strength (Ugural, 2003). Following table shows the material types used in this study.

Table 1. Mechanical Properties of Pinion and Gear Materials

Property	Pinion Materials		Gear Material
	Type I	Type II	
Density	7850 kg/m³	7850 kg/m³	7850 kg/m³
Yield Strength	441 MPa	1640 MPa	621 MPa
Ultimate Tensile Strength	586 MPa	1770 MPa	827 MPa
Modulus of Elasticity	200 GPa	200 GPa	170 GPa
Poissons's Ratio	0,3	0,3	0,3
Brinell Hardness	207 HB	510 HB	400 HB

In Table 1, it is seen that two different pinion materials (Type I and Type II) with highest and lowest strengths available for gears are taken into account in order to investigate the effect of material properties for the comparison of design results. (Type I: AISI 1030 Q&T @650 °C, Type II: AISI 4140 oil Q&T@205 °C; Gear Material: Duct. iron Q to bainite, GR.120-90-02).

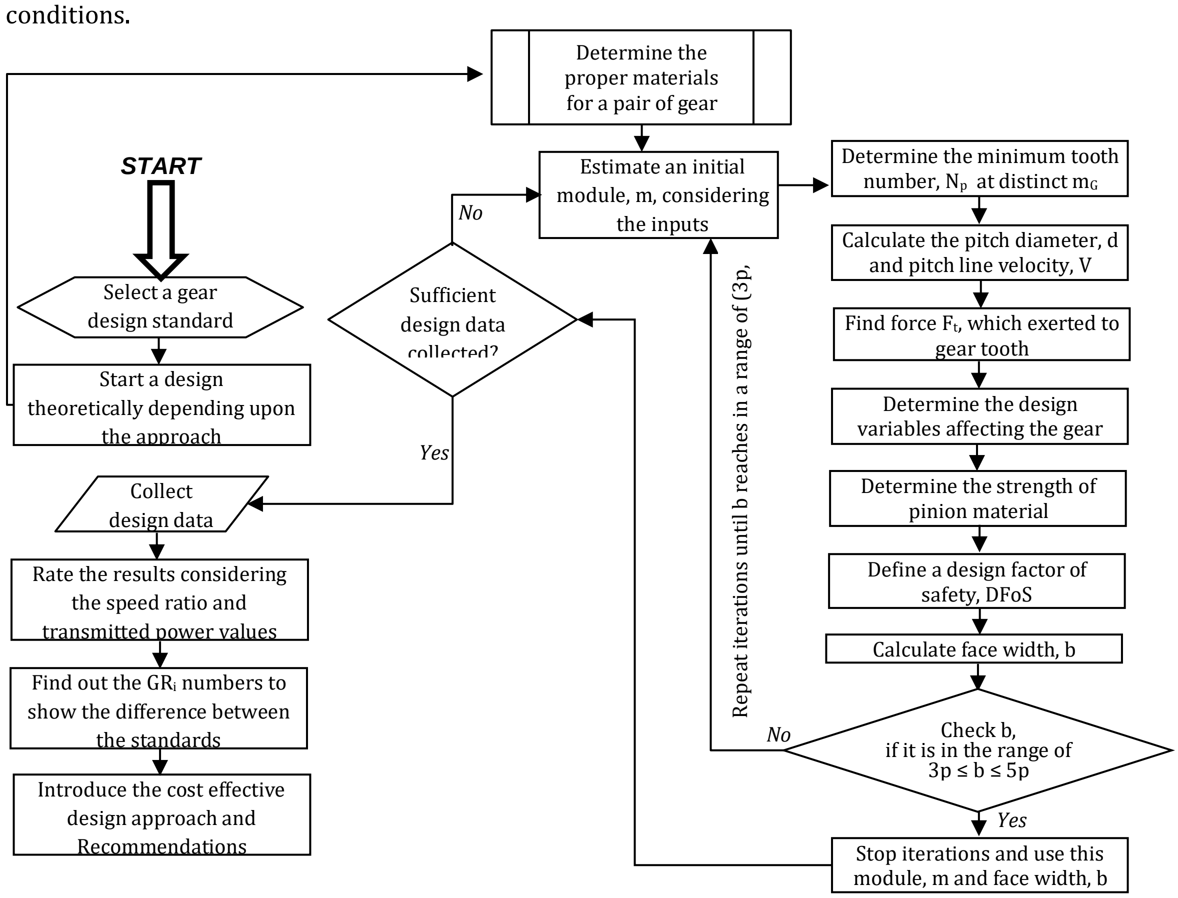


Figure 1. A General Systematic Method for Design and Comparison

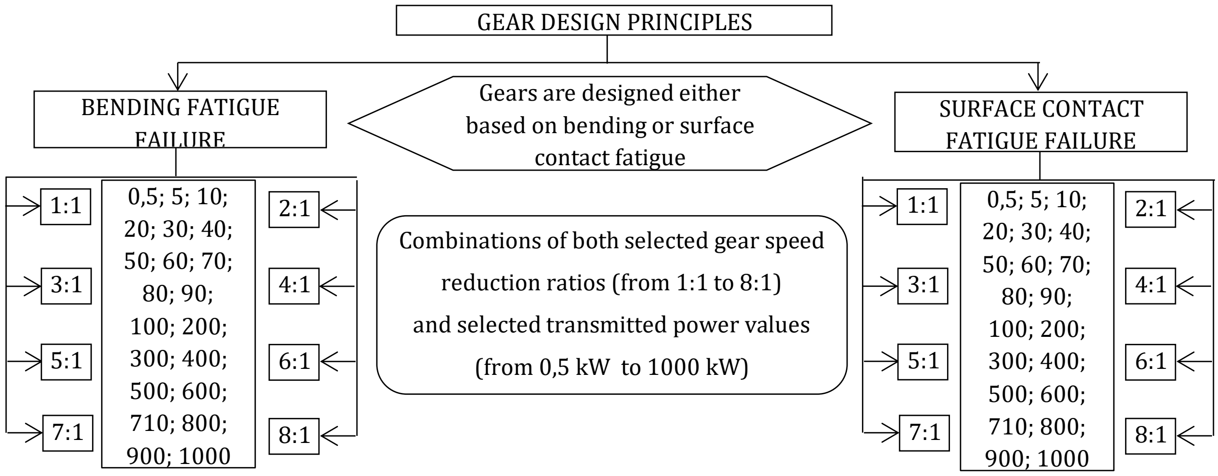


Figure 2. Combinations of Gear Speed Ratio and Power Transmitted Values

2.2. Input Parameters for Design

Table 2 introduces all input parameters comprehensively that should be determined during a design process. The study considers the precision gears and standard tooth geometry. The most common pressure angle (∅) of 20° and interference-free minimum pinion teeth numbers were selected for a compact gear design as commonly used in industry. The smallest numbers of teeth on the spur pinion without interference for one to one gear ratio (Np=13) and for higher speed ratios were calculated using the appropriate formula for the ∅ of 20° given by Budynas and Nisbett (2011). A range of 2:1 to 3:1; 4:1 to 6:1; and 7:1 to 8:1 share the same pinion tooth number as presented in Table 2. In order to achieve a fair comparison for two gear standards, all input parameters were kept identical.
However, the quality numbers for gears given in Table 2 are 8 and 9 for ISO and AGMA, respectively. This is due to the rule of 17 that was described by Cahala (1999) as the sum of the AGMA and ISO quality numbers describing the same gear is approximately 17.

Table 2. Input Parameters for Spur Gear Design

Input Parameters	Value
Pressure angle, ∅	20°
Gear tooth geometry (standard)	Interference free involute spur, full depth teeth
Input speed of a power source	1200 rpm
Number of life cycles, N	10^8
Design factor of safety, n_d	2,1
Reliability, %	99,9
Operating temperature, T	Moderate or low (~120°C)
Quality number for gear	ISO: 8 and AGMA: 9
Material properties of gear pair	see Table 1
Working characteristics of driving and driven machines	Uniform
Selected transmitted power range	0,5-1000 kW (@ 21 values)
Selected gear speed ratio range, m_G /Corresponding pinion teeth number (N_p)	1:1 /13, 2:1-3:1 /15, 4:1-6:1 /16, 7:1-8:1 /17
Design Criteria	Based on both bending fatigue and surface contact fatigue

2.3. Spur Gear Design

The design of spur gears is carried out based on selecting the module (m), and determining the face width (b). This is iterative process as there are two unknown design outputs that are (m) and (b) to obtain whereas there is one stress equation. Thus, module is estimated and calculations are iterated until the face width reaches in an accepted range as seen in Figure 1. This procedure were performed by using the approaches, ISO 6336 and 9085-2002 standards and ANSI/AGMA 2101-D04 Standard considering for both bending and surface contact fatigue failure criteria. Related stress and strength expressions are required to equate and rewrite for face width. The Followings provide the both. Please refer to corresponded literature for the symbolic notations. Table 3 provides the obtained face width (b) equations.

Based on Bending Fatigue Failure;

AGMA Fatigue Bending Stress,

$$\sigma_F = F_t.K_o.K_V.K_S.\frac{1}{b.m_t}.\frac{K_H.K_B}{Y_J} \quad (1)$$

AGMA Fatigue Strength for Bending,

$$\sigma_F \leq \frac{\sigma_{FP}.Y_N}{S_F.Y_\theta.Y_Z} \quad (2)$$

ISO Fatigue Bending Stress,

$$\sigma_F = \sigma_{F0}.K_A.K_V.K_{F\beta}.K_{F\alpha} \leq \sigma_{FP} \quad (3)$$

$$\sigma_{F0} = \frac{F_t}{b.m_n}.Y_F.Y_S.Y_\beta.Y_B.Y_{DT} \quad (4)$$

ISO Fatigue Strength for Bending,

$$\sigma_{FP} = \frac{\sigma_{F_{lim}}.Y_{ST}.Y_{NT}}{S_{F_{min}}}.Y_{\delta relT}.Y_{RrelT}.Y_X \quad (5)$$

Based on Surface Contact Fatigue Failure;

AGMA Fatigue Surface Contact Stress;

$$\sigma_H = Z_E.\sqrt{F_t.K_o.K_v.K_S.\frac{K_H}{d_{w1}.b}.\frac{Z_R}{Z_I}} \quad (6)$$

AGMA Fatigue Strength for Surface Contact;

$$\sigma_H \leq \frac{\sigma_{HP}}{S_H} \cdot \frac{Z_N}{Y_\theta} \cdot \frac{Z_W}{Y_Z} \quad (7)$$

ISO Fatigue Surface Contact Stress;

$$\sigma_H = Z_B . \sigma_{H0} \sqrt{K_A . K_V . K_{H\beta} . K_{H\alpha}} \leq \sigma_{HP} \quad (8)$$

$$\sigma_{H0} = Z_H . Z_E . Z_\varepsilon . Z_\beta . \sqrt{\frac{F_t}{d_1 . b_H} \cdot \frac{u+1}{u}} \quad (9)$$

ISO Fatigue Strength for Surface Contact;

$$\sigma_{HP} = \frac{\sigma_{Hlim} . Z_{NT}}{S_{Hmin}} . Z_L . Z_V . Z_R . Z_W . Z_X \quad (10)$$

Table 3. Face Width Equations for Design Approaches

Design Approach	Fatigue Failure	Face width, b	Eq.
ISO	Bending	$b = \frac{S_F . F_t}{\sigma_{F_{lim}} . Y_{ST} . Y_{NT} . Y_{\delta\ rel\ T} . Y_{R\ rel\ T} . Y_X . m_n} . Y_F . Y_S . Y_\beta . Y_B . Y_{DT} .\ K_A . K_V . K_{F\beta} . K_{F\alpha}$	(11)
	Surface Contact	$b = \left(\frac{Z_E}{\sigma_{H\ lim}}\right)^2 \cdot \left(\frac{Z_B . Z_H . Z_\varepsilon . Z_\beta .}{Z_{NT} . Z_L . Z_V . Z_R . Z_W . Z_X}\right)^2 \cdot \left(\frac{u+1}{u}\right) \cdot \frac{F_t}{d_1} . K_A . K_V . K_{H\beta} . K_{H\alpha} . S_H^{\ 2}$	(12)
AGMA	Bending	$b = \frac{S_F . F_t}{\sigma_{FP} . m_t . Y_J} \cdot \frac{Y_\theta . Y_Z}{Y_N} . K_O . K_V . K_S . K_H . K_B$	(13)
	Surface Contact	$b = \left(\frac{Z_E}{\sigma_H}\right)^2 \cdot \left(\frac{Y_\theta . Y_Z}{Z_N . Z_W}\right)^2 \cdot \left(\frac{Z_R}{Z_I}\right) \cdot \frac{F_t . K_o . K_v . K_S . K_H . S_H^{\ 2}}{d_{w1}}$	(14)

3. Geometric Rating Numbers, GR_i

After finding the design outputs (m and b), m times b (m.b) results are obtained and combined to form a more like a geometrical value which may be used as a representative for the cross-sectional area at the pitch diameter. This is because half of the circular pitch (p/2=π.m/2) approximately equals to tooth thickness in SI units. The geometrical value is going to be used to compare the results of each approach of gear design. Hence, a new dimensionless parameter which may be called as "Geometric Rating Number", GR_i, may be defined specifically for relative comparison of the each as;

$$GR_i = \frac{\frac{\pi . m_{ISO} . b_{ISO}}{2}}{\frac{\pi . m_{AGMA} . b_{AGMA}}{2}} = \frac{m_{ISO} . b_{ISO}}{m_{AGMA} . b_{AGMA}}$$

(15)

Where m_{ISO} and b_{ISO} are the module and face width obtained from ISO Standard, and module, m_{AGMA} and face width, b_{AGMA} are obtained from AGMA Standard. Eq. 15 bases the relative comparison according to the AGMA Standard.

4. Results and Discussion

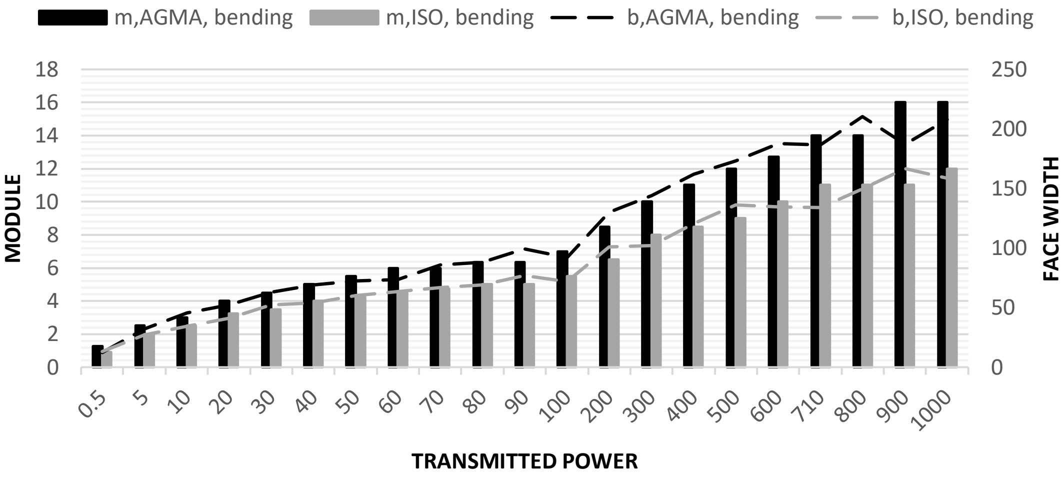
4.1. Design Outputs, m and b

Only the design of pinion is usually carried out and the gear is sized based on the design of the pinion. This is because pinion is the smallest and weakest member in meshing couple and rotates more than the gear itself for the speed ratios greater than 1:1. In this study, the same approach was used to obtain the design outputs and to make comparison of the results obtained from the approaches. As Figure 2 shows, the spur gear designs are carried out for eight gear speed ratios at 21 power transmission values. This gives 168 design results for just one and 336 design results were collected when both bending and surface contact failure criteria are considered.

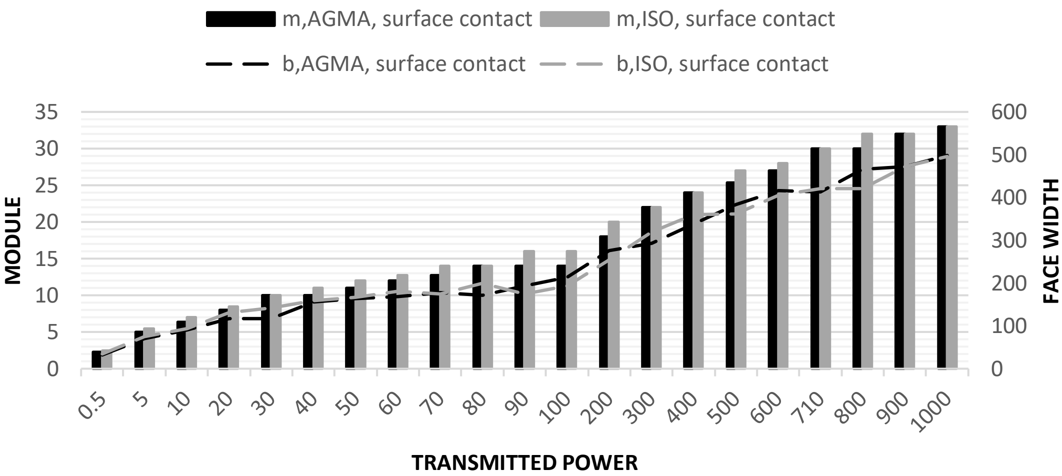
Table 5. Results of Design Outputs for Module (m) and Face Width (b) at 3:1 Speed Ratio

Material type	Transmitted Value, kW	Fatigue Failure	AGMA Standards		ISO Standards	
			Module, m_{AGMA}, mm	Face width, b_{AGMA}, mm	Module, m_{ISO}, mm	Face width, b_{ISO}, mm
Type I	10	Bending	3,5	51,20	3	45,85
Type I	200	Bending	10	145,98	8	125,59
Type I	50	Surface Contact	12	175,56	16	206,16
Type I	400	Surface Contact	25,4	384,66	32	447,93
Type II	5	Bending	2,5	32,17	2	26,90
Type II	300	Bending	10	144,05	8	102,13
Type II	40	Surface Contact	10	156,27	11	158,28
Type II	800	Surface Contact	30	465,76	32	420,23

Beside this, two different types of pinion materials were considered which yield 672 design results. These results only belong to one obtained from a gear standard. When both ISO and AGMA gear standards are considered, a total of 1344 design results were obtained. Due to the limited space here, only some of the results are tabulated randomly at a speed ratio of 3:1 as an example in Table 5. Figure 3 to 6 represent the results at different speed ratios for only material Type II. These show the change of module and face width against transmitted powers for both of the design approaches based on bending and surface contact fatigue failure, respectively. The close scrutinize on the figures show the effect of speed reduction ratio on the results. Although both ISO and AGMA standards show a very similar trend as seen in figures, the results varies due to the inherited features of the approaches.

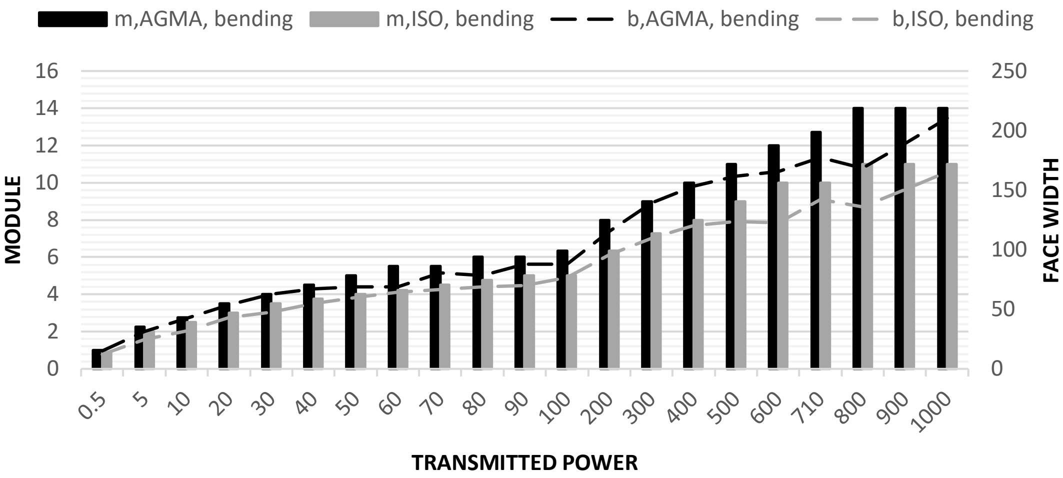


(a)

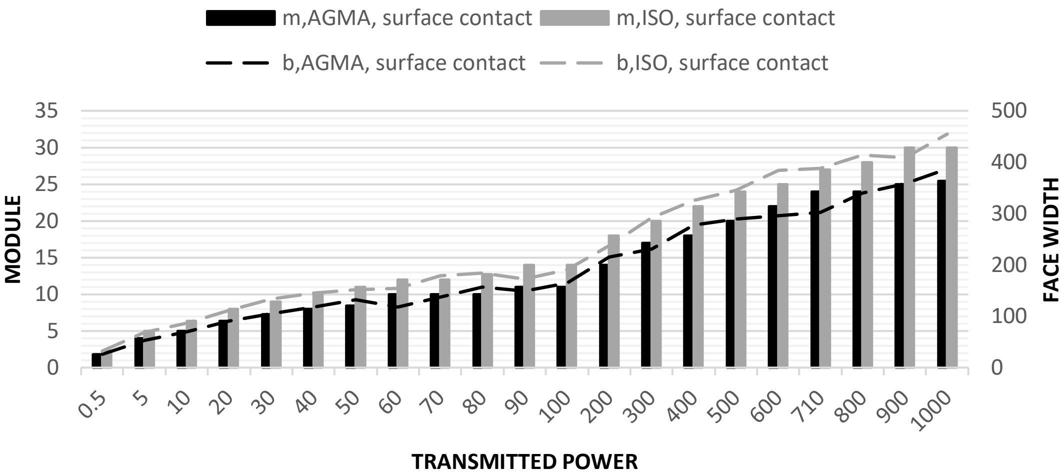


(b)

Figure 3. The Change of m and b Depending on Transmission at 1:1 Speed Ratio Based on; (a) Bending Fatigue Failure, (b) Surface Contact Fatigue Failure

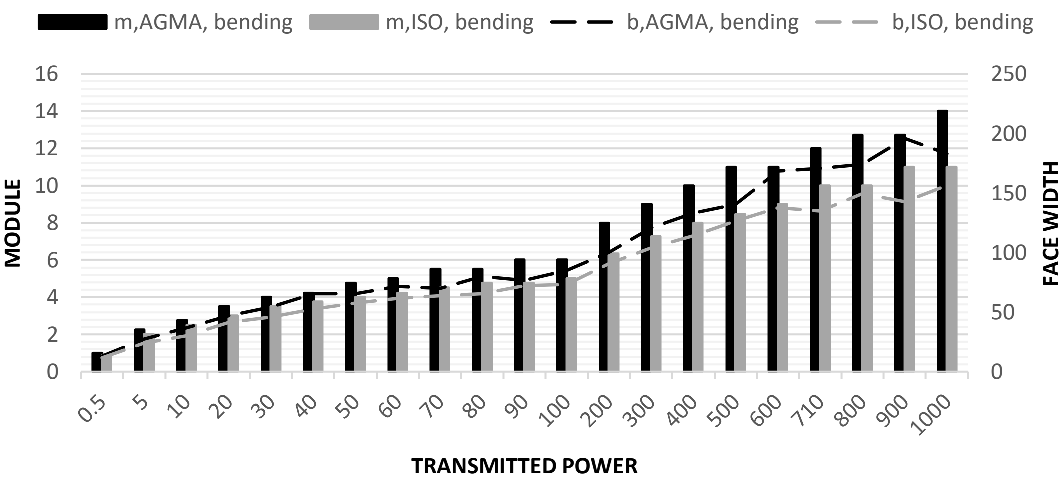


(a)

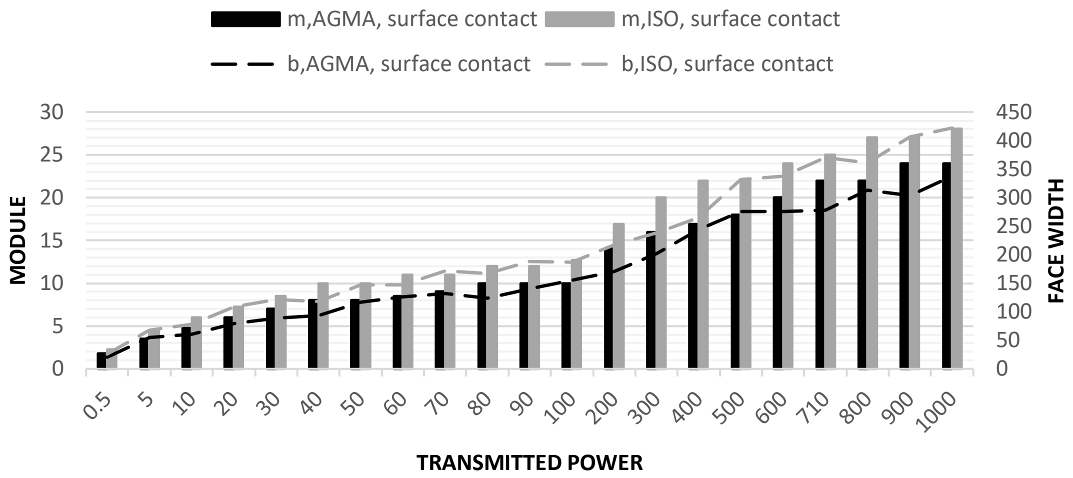


(b)

Figure 4. The Change of m and b Depending on Transmission at 3:1 Speed Ratio Based on; (a) Bending Fatigue Failure, (b) Surface Contact Fatigue Failure

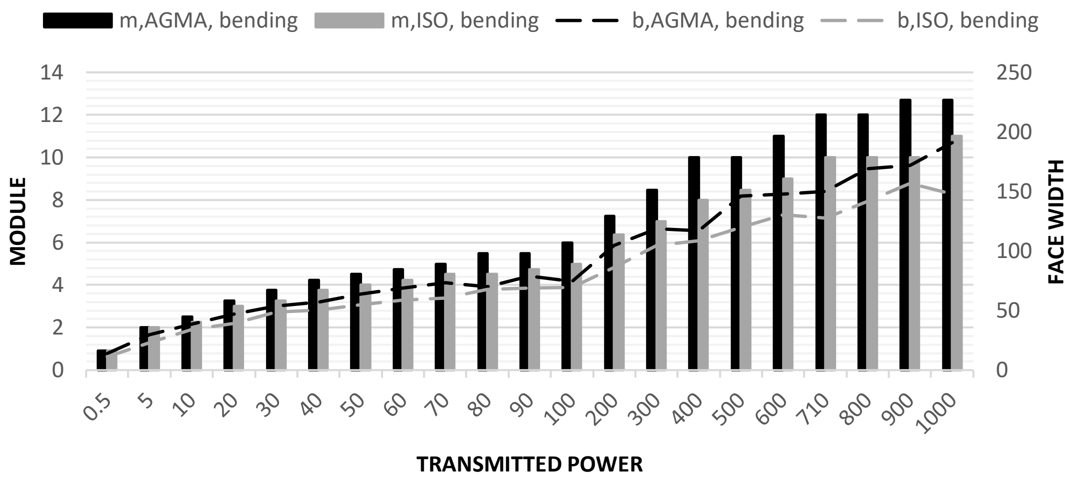


(a)

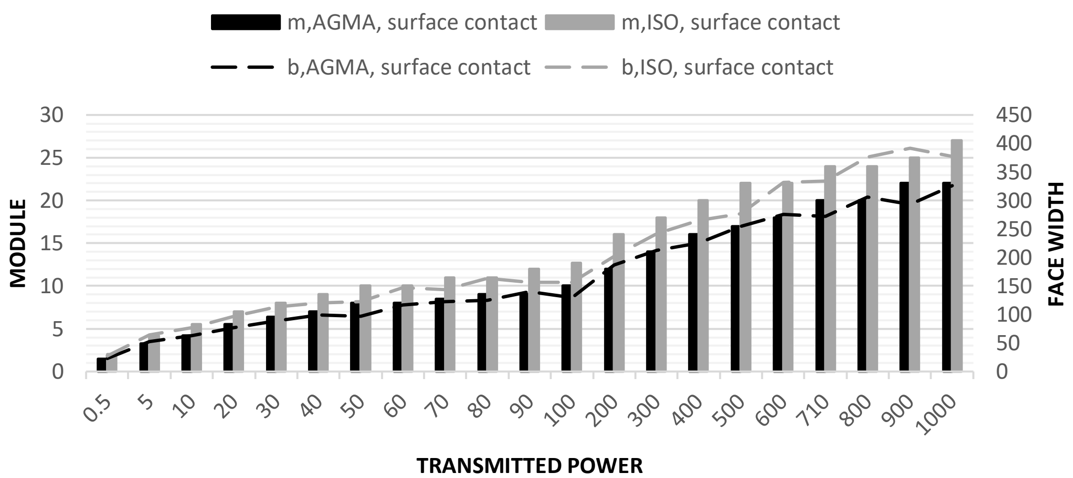


(b)

Figure 5. The Change of m and b Depending on Transmission at 5:1 Speed Ratio Based on; (a) Bending Fatigue Failure, (b) Surface Contact Fatigue Failure



(a)



(b)

Figure 6. The Change of m and b Depending on Transmission at 8:1 Speed Ratio Based on; (a) Bending Fatigue Failure, (b) Surface Contact Fatigue Failure

4.2. Rating of the GR_i Results

The similar trends obtained from Figure 3 to 6 allowed to make a relative comparison of the approaches. For this reason, GR_i numbers have been developed in order to see the effect of both module and face width together. GR_i numbers were obtained at all combinations of gear speed ratio and transmitted power values for both ISO and AGMA standards. The results are introduced by radar charts and given in Figure 7 and 8 for the materials, Type I and Type II considering the fatigue failure criteria. In these figures, the numbers around the radars (from 0,5 to 1000) refer to transmitted power values, and the inner circles with corresponding values (from 0 to 1,2) in radar charts represents the GR_i scale.

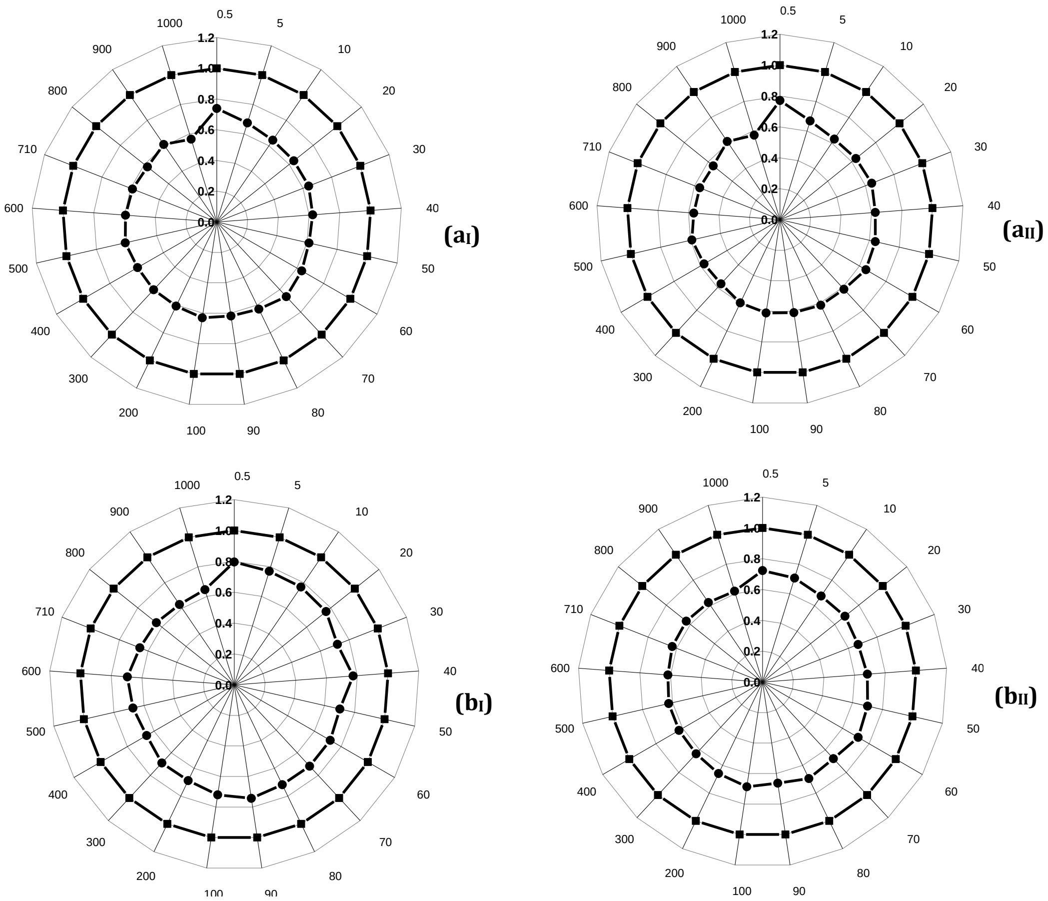
Table 6 shows the mean GR_i numbers obtained from the results using statistical analysis. Rating of the results obtained from gear design appraches were made by taking AGMA Standard as reference, and so its value is always one in each radar chart. Therefore, it is seen that ISO Standard gives smaller results than AGMA Standard for the bending fatigue criteria. On the contrary to this, ISO Standard give greater results than AGMA Standard when the fatigue failure criteria is the surface contact stress.

Table 6 also verifies that the results are free from the material properties since the obtained results from two different material types are very close to each other.

As it is seen in Table 6, when the design of spur gear is made based on bending fatigue, the difference of GR_{AGMA} and GR_{ISO} is (1,00-0,72) 0,28 and (1,00-0,68) 0,32 for the materials, Type I and Type II, respectively. On the other hand, if the design of spur gear is made based on surface contact fatigue, the difference in between GR_{ISO} and GR_{AGMA} (1,48-1,00) is 0,48 and (1,41-1,00) 0,41 for the materials, Type I and Type II, respectively.

Table 6. Mean GR_i Numbers for the Standards

Design Approach	Fatigue Failure	GR_i	
		Type I	Type II
ISO 6336 Standard	Bending	0,72	0,68
ISO 9085-2002 Standard	Surface Contact	1,48	1,41
ANSI/AGMA 2101-D04 Standard	Bending	1,00	1,00
	Surface Contact	1,00	1,00



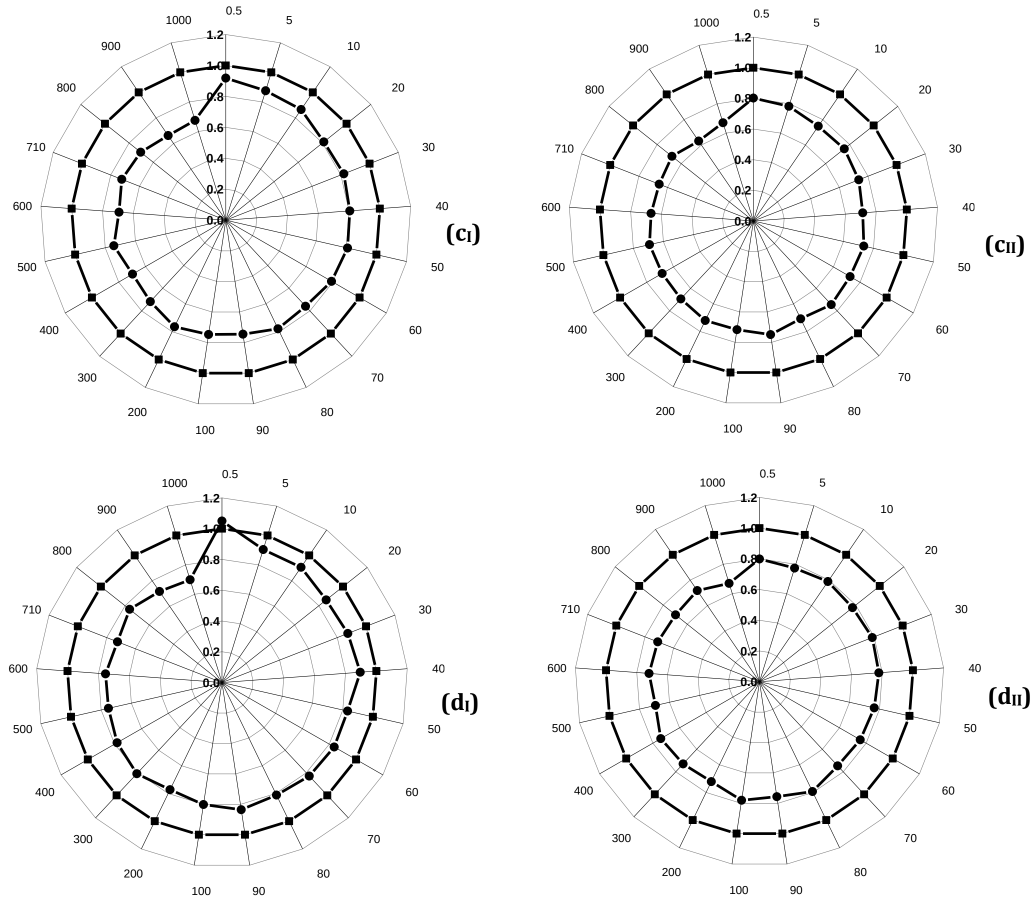
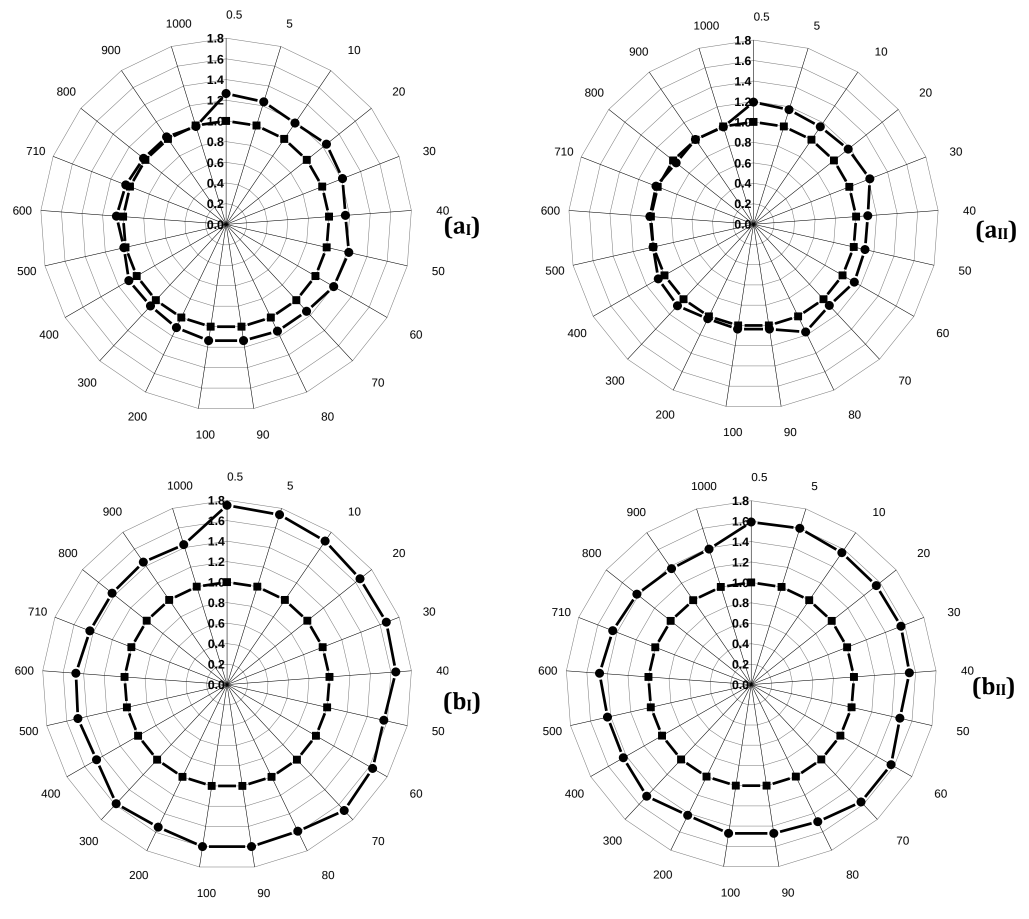


Figure 7. Comparison of GR_i Results for the Standards Based on Bending Fatigue Failure Criteria at a Speed Ratio of (a) 1:1, (b) 3:1, (c) 5:1, (d) 8:1 *(subscript I indicates material Type I and II indicates material Type II)*



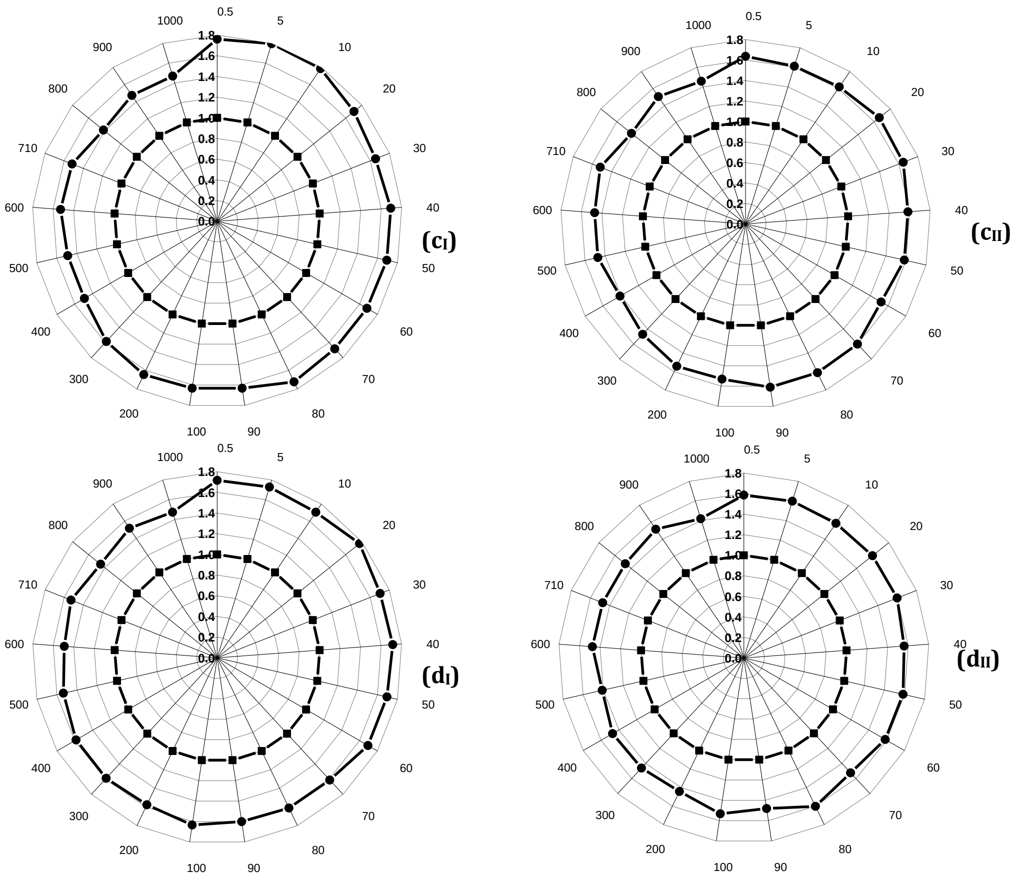


Figure 7. Comparison of GRi Results for the Standards Based on Surface Contact Fatigue Failure Criteria at a Speed Ratio of (a) 1:1, (b) 3:1, (c) 5:1, (d) 8:1 *(subscript I indicates material Type I and II indicates material Type II)*

5. Conclusion

The most commonly used gear design standards, ISO and AGMA, were considered to carry out a spur gear design, and the design results were compared under the same design conditions. The study obtained conditions to recommend the suitable gear design approach intelligently by introducing useful outputs, practical curves and charts. Hence, the designers can achieve geometrical optimization for volume/weight critical designs which directly affects the cost. And the following conclusions were derived from the results;

- Under the same conditions, increasing the transmitted power will cause to select bigger module as expected whereas the increasing gear speed ratio provides to select a little smaller module. This is because, the number of teeth on a bigger gear member, which increases contact ratio, increases the number of gears in mesh. This allows more tooth to share the load. As a result of this, the force exerted on each tooth on a pinion decreases. Thus, gear stresses decrease and the expressions give smaller module for a better design.
- The relative comparison obtained by GR_i numbers exactly showed that ISO Standard gives smaller design outputs while considering the bending fatigue. On the other hand, if the design is carried out based on surface contact fatigue, the smaller design outputs are obtained when AGMA Standard is used. Therefore, for industrial applications, a designer should use ISO Standard if the tooth root fatigue failure is the primary concern and AGMA Standard is recommended if the tooth surface fatigue failure is the primary concern, respectively.
- As the GR_i number considers a comparative gear size and mean values of it are obtained irrespective of power and speed ratios, ISO Standard provides 28% (for the lowest strength) to 32% (for the highest strength) smaller gear tooth volume considering the bending fatigue failure while AGMA provides 41 to 48% smaller gear tooth volume considering the surface contact fatigue failure for the lowest and highest material strength range. Actually using materials having different properties have little or no effects since a very similar trends were obtained between Type I and Type II. This means that the findings cover all ranges of material properties.

Briefly, cost effective design can now be achieved and a designer can be aware of lost or gain on the cost and failure or success of the design with the aid of useful outputs provided by this study.

Nomenclature

b	: Face width
b_{AGMA}	: Face width obtained from AGMA
b_{ISO}	: Face width obtained from ISO
GR_i	: Geometric rating number
GR_{AGMA}	: Mean GR_i for AGMA approach
GR_{ISO}	: Mean GR_i for ISO approach
m	: Module
m_{AGMA}	: Module obtained from AGMA
m_{ISO}	: Module obtained from ISO
m_G	: Gear speed ratio
Np	: Interference-free pinion teeth number
p	: Circular pitch
$\varnothing$	: Pressure angle

Conflict of Interest

No conflict of interest was declared by the authors.

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