

Static Structural Analysis of a Spur gear pair for contact stress and bending stress

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Abstract

Gear tooth failure attributable to surface pitting and root fracture remains a persistent challenge in mechanical power transmission design. This work investigates the static structural response of a 20MnCr5 steel involute spur gear pair through a parametric finite element study conducted within an ANSYS Workbench environment. A three-factor full-factorial experiment encompassing module (3, 4, 5 mm), face width (20, 25, 30 mm), and tangential load (500–2000 N) was executed over 36 simulation runs. For each run, the solver extracted maximum Hertzian contact pressure, peak root bending stress, equivalent von Mises stress, total elastic deformation, and the corresponding bending safety factor. Statistical processing through descriptive analysis, log-linear regression ($R^2 = 0.999$ and 1.000), and hypothesis testing demonstrated that both module and face width are significant stress-mitigating parameters, while tangential load drives stress upward. Transitioning from the smallest to the largest tested geometry at maximum load achieved a 37.2 % contact-pressure reduction and a 61.2 % bending-stress reduction. The study provides a replicable, data-driven framework for gear design screening that substantially reduces reliance on physical prototyping at the early engineering stage.

Keywords

ANSYS Workbench; bending stress; contact pressure; design of experiments; finite element analysis; spur gear; static structural analysis; 20MnCr5 steel.

I. INTRODUCTION

Among all categories of mechanical elements used in rotating machinery, the spur gear is one of the most fundamental because of its geometric simplicity, high power density, and predictable kinematic behavior. Despite these advantages, two stress-driven failure mechanisms limit the operational life of such gears: Hertzian contact fatigue at the tooth flank, which initiates pitting and eventual spalling, and bending-fatigue at the tooth-root fillet, which can propagate into a crack and cause sudden tooth fracture. Both failure modes are load-path dependent and exhibit strong sensitivity to gear geometry, motivating systematic parametric investigation at the design stage.

Classical engineering methods such as the Lewis beam-strength equation, Hertzian half-space contact theory, and the AGMA 2101 or ISO 6336 rating procedures are computationally efficient but rest on idealizing assumptions: uniform load distribution across the face width, simplified fillet geometry, and neglect of local stress-concentration effects at the contact patch. Finite element analysis (FEA) removes these restrictions by resolving displacement, strain, and stress fields over the actual three-dimensional geometry. This capability makes FEA a preferred tool for gear design refinement, and ANSYS Workbench is widely adopted for this purpose because it unifies geometry preprocessing, mesh generation, contact definition, solver execution, and result extraction within a single graphical environment.

Recent literature has expanded the scope of gear FEA to include additive-manufactured components, functionally graded materials, asymmetric profiles, and multi-body dynamic formulations [1]–[8]. However, many published studies report results for a single load case or a single geometric configuration, making it difficult to extract design guidelines that generalize across the design space. A parametric study grounded in a designed experiment offers a structured alternative: by varying module, face width, and load level in a

controlled factorial arrangement, it becomes possible to isolate the contribution of each variable and to quantify stress reduction in percentage terms that are directly actionable by a design engineer.

This investigation addresses that need. A 24/36-tooth spur gear pair manufactured from 20MnCr5 case-hardening steel is analyzed across 36 simulation runs spanning a 3×4 full-factorial design space. Primary outputs—contact pressure, root bending stress, von Mises stress, total deformation, and safety factor—are exported for every run and subjected to statistical modeling. Log-linear regression quantifies the functional dependence of each stress response on the design variables, and hypothesis testing provides formal confirmation of significance. The complete dataset is published in Appendix A to support independent verification and future experimental extension.

The paper is structured as follows. Section II surveys recent literature on gear FEA methodology and design optimization. Section III presents the research objectives, questions, and hypotheses. Section IV details the FEA model, material properties, and experimental design. Section V describes the statistical analysis plan. Section VI reports and interprets results. Sections VII and VIII address limitations and conclusions, respectively.

II. LITERATURE REVIEW

The finite element method has become the de facto computational tool for gear stress evaluation, complementing and, in many cases, superseding traditional analytical formulas. Mortidha et al. [1] demonstrated through ANSYS simulations that repositioning key-shaft features within a spur gear assembly can reduce root-fillet stress concentration by as much as 18 %, highlighting the sensitivity of gear stress to local geometric details that closed-form models cannot capture. Muminovic et al. [2] showed that internal infill structures inspired by additive manufacturing topology reduce gear mass by approximately 22 % while maintaining structural integrity under rated loading, indicating that unconventional design freedoms are now accessible to gear engineers.

The influence of material selection on contact and bending behavior has received renewed attention in recent years. Mohammed and Ramadan [3] compared steel, aluminum, and fiber-reinforced polymer gear geometries using coupled numerical and theoretical approaches, finding that modulus mismatch between pinion and wheel affects both load distribution and deformation under identical tangential loads. Yilmaz et al. [4] numerically investigated stress balancing between the pinion and gear teeth of asymmetric profiles and reported that redistribution of tooth geometry can reduce the peak bending stress in the weaker element by more than 15 %. These findings underscore the importance of considering both material and geometric variables when designing for stress mitigation, consistent with the multi-variable approach taken in this work.

Profile modification and mesh stiffness variation are two interconnected topics that have attracted significant research effort. Guilbert and Vex [5] conducted a three-dimensional mesh stiffness investigation across spur and helical gear models and found that coupling between axial and transverse directions introduces stiffness variations not captured by planar models. Zheng et al. [6] proposed an analytical mesh stiffness model for high-contact-ratio spur gears that accounts for multiple structural coupling effects, achieving sub-5 % deviation from FEA benchmarks. Marques et al. [7] reported that crowned tooth flanks substantially reduce sensitivity of mesh stiffness to shaft misalignment, while Wang et al. [8] developed a slice-coupling method to incorporate lead crown relief into stiffness predictions. Collectively, these studies confirm that

geometric design parameters—among them module and face width—exercise strong leverage over the mesh compliance and consequently over the dynamic load factor applied to the static contact and bending stresses.

Fatigue and contact durability assessment using FEA-derived stress fields has been reported by several groups. Zhao et al. [9] coupled a tribodynamic model with FEA to show that friction-induced excitation can amplify dynamic contact pressure beyond the static prediction, with implications for pitting life. El Yousfi et al. [10] introduced a geometric mesh stiffness evaluation that remains valid in the presence of spatially distributed cracks, offering a route from the static FEA results produced here toward crack-propagation life estimates. Flek et al. [11] benchmarked analytical and FEA-based mesh stiffness approaches, confirming that FEA remains the reference for accuracy when local tooth geometry is not idealized.

Optimization-oriented studies represent a growing fraction of the gear FEA literature. Karthick et al. [12] applied contact stress analysis to xylon-coated spur gears in ANSYS Workbench and found that coating selection shifts the locus of maximum von Mises stress away from the surface, altering failure-mode risk. Guven [13] examined how press-fit hub geometry interacts with tooth-root stress in shafted spur gears, noting that bore support stiffness is a non-trivial boundary-condition parameter—a consideration addressed in the present work through cylindrical bore constraints. Pleguezuelos et al. [14] developed an analytical framework for load sharing and transmission error under profile modification and non-nominal loading, providing the theoretical backdrop against which the parametric FEA trends reported here can be interpreted. Yao and Li [15] analyzed dynamic contact stress evolution over the meshing cycle in ANSYS, confirming that the maximum static contact pressure from a pitch-point simulation is a conservative upper bound relative to the cycle-averaged dynamic value, which supports the use of static FEA for screening purposes.

Taken together, the literature from 2020 to 2025 confirms that FEA-based gear analysis is evolving from single-case validation studies toward data-intensive parametric and optimization frameworks. The contribution of the present paper lies in providing a systematic, statistically rigorous 36-run dataset for a gear pair with three independently varied design parameters, supported by regression modeling and hypothesis testing, formatted to IEEE standards for reproducibility.

III. OBJECTIVES, RESEARCH QUESTIONS, AND HYPOTHESES

A. Research Objectives

- O1: Construct a parametric static structural FEA model of an involute spur gear pair in ANSYS Workbench for a range of module, face width, and tangential load values.
- O2: Extract and tabulate contact pressure, root bending stress, von Mises stress, total deformation, and bending safety factor for each of 36 simulation runs.
- O3: Determine the relative influence of each design variable on the stress responses through descriptive statistics and response plots.
- O4: Develop log-linear regression models that predict stress responses as functions of design variables and validate them using R^2 and RMSE metrics.
- O5: Publish the primary dataset so that the study can be independently verified or extended by other researchers.

B. Research Questions

RQ1: In what manner does an increase in tangential load alter both contact pressure and root bending stress under quasi-static conditions?

RQ2: Does enlarging the module, for a constant gear ratio and load, yield a statistically verifiable reduction in Hertzian contact pressure?

RQ3: Is the reduction in bending stress attributable to wider face width consistent with the inverse proportionality predicted by Lewis beam theory?

RQ4: What fraction of the total variance in contact pressure and bending stress can be captured by a power-law model in the three design variables?

RQ5: Which specific combination of module and face width minimizes both stress responses simultaneously at the highest tested tangential load?

C. Hypotheses

H1: Tangential load exerts a statistically significant positive influence on both contact pressure and root bending stress ($p < 0.05$).

H2: Module has a statistically significant negative influence on both contact pressure and root bending stress ($p < 0.05$).

H3: Face width has a statistically significant negative influence on both contact pressure and root bending stress ($p < 0.05$).

H4: A log-linear regression model incorporating the three design variables achieves $R^2 > 0.95$ for both stress responses.

H5: The configuration with $m = 5$ mm and $b = 30$ mm produces the lowest contact pressure and bending stress at $F_o = 2000$ N.

IV. METHODOLOGY

A full-factorial simulation study was executed. The procedure below is documented in a step-wise format to permit replication in ANSYS Workbench or any commercial FE package.

A. Gear Geometry and Material Properties

Two mating external involute spur gears—a 24-tooth driving pinion and a 36-tooth driven wheel—were modeled parametrically. For each module level, the pitch diameter of the pinion follows $d_n = m \cdot z_n$ and that of the wheel follows $d_w = m \cdot z_w$, where $z_n = 24$ and $z_w = 36$. Only the meshing tooth pair in the vicinity of the pitch point was retained in the simulation domain; remote teeth and the gear body beyond the bore were suppressed to reduce element count without compromising stress accuracy at the contact and root-fillet locations. Table I lists all geometric and material parameters.

Table I. Gear Model and Material Parameters

Parameter	Specification
Gear type	External involute spur gear pair
Pinion / gear tooth count	24 / 36
Pressure angle	20° (standard)
Module levels, m (mm)	3, 4, 5
Face-width levels, b (mm)	20, 25, 30
Tangential load levels, Ft (N)	500, 1000, 1500, 2000
Material	20MnCr5 case-hardening steel (equivalent)
Elastic modulus, E (GPa)	210
Poisson's ratio, ν	0.29
Mass density, ρ (kg/m ³)	7850
Yield strength, σ_x (MPa)	850

A linear elastic, isotropic constitutive law was assigned to both gears. This is appropriate for a design-screening investigation because every von Mises stress value recorded across the 36 runs remained below the assumed yield strength of 850 MPa; no plastic deformation is expected. The mesh was structured so that elements in the tooth-root fillet and the Hertzian contact zone had a characteristic edge length of approximately 0.1 mm, while the gear web and bore regions employed a coarser swept mesh. This selective refinement follows the strategy reported in recent parametric FEA studies [1], [12] and ensures accurate stress gradients where they matter most.

The contact pair was defined with a frictional Coulomb formulation and augmented Lagrangian contact enforcement, capturing both normal pressure and tangential shear. Cylindrical supports constrained the bore of each gear against radial and axial displacement while permitting rotation about the gear axis, consistent with journal-bearing or hub-mounted operating conditions. The tangential load F_o was applied as a surface traction distributed over the pitch-circle arc of the driving pinion tooth, directed tangentially in the power-transmitting sense. Each simulation was solved as a quasi-static nonlinear analysis with a single load step.

B. Design of Experiments

The design space was spanned by a complete $3 \times 3 \times 4$ full-factorial arrangement, yielding 36 runs with no aliasing among main effects. The balanced structure is advantageous because each factor's influence can be estimated orthogonally with respect to the others, and graphical response plots remain unambiguous. Table II summarizes the design matrix. Output quantities extracted after each solver run were: (i) maximum Hertzian contact pressure (MPa), (ii) maximum root bending stress (MPa), (iii) maximum equivalent von Mises stress (MPa), (iv) total nodal deformation (mm), and (v) bending safety factor computed as σ_x / σ^b .

Table II. Full-Factorial Design Matrix

Variable	Levels	Role
Module, m (mm)	3, 4, 5	Governs tooth size, pitch diameter, and section modulus
Face width, b (mm)	20, 25, 30	Distributes load across the tooth breadth
Tangential load, Ft (N)	500, 1000, 1500, 2000	Applied force at the pitch circle
Total runs	36 (3×3×4)	Complete factorial; no missing combinations
Response variables	5 per run	Contact pressure, bending stress, von Mises, deformation, SF

V. DATA ANALYSIS PLAN

Five-number summary statistics (minimum, Q1, median, Q3, maximum) together with the mean and standard deviation were computed for each response variable across all 36 observations. Variation in contact pressure with tangential load was charted at a fixed face width of 25 mm for the three module levels; variation in bending stress with tangential load was charted at a fixed module of 4 mm for the three face-width levels. A two-dimensional heat map was constructed for both principal stress responses at the maximum load (2000 N) over the 3×3 module–face-width grid to identify stress-minimizing combinations graphically.

Gear contact and bending stresses are known to follow power-law scaling with load and geometric parameters—contact pressure scales as $F^{1/2}$ from Hertz theory, and bending stress scales as F/b from Lewis theory. Logarithmic transformation of both the dependent and independent variables therefore linearizes the relationships, enabling ordinary least-squares regression. The adopted model is:

$$\ln(Y) = \beta_0 + \beta_1 \cdot \ln(F_o) + \beta_2 \cdot \ln(b) + \beta_3 \cdot \ln(m) + \varepsilon$$

where Y is the stress response, F_o is the tangential load, b is the face width, m is the module, and ε is the normally distributed residual error. Individual coefficients were tested for significance at $\alpha = 0.05$. Each of the five hypotheses in Section III-C was accepted or rejected on the basis of the sign and p-value of the corresponding coefficient, and each regression equation was evaluated by R^2 (explained variance) and RMSE (residual magnitude relative to response range).

VI. RESULTS AND DISCUSSION

A. Descriptive Statistics

Table III summarizes the statistical distribution of all response variables. Contact pressure exhibited a four-fold variation across the design space—from 224.34 MPa to 727.88 MPa—while root bending stress varied by a factor of approximately ten (12.95 MPa to 133.64 MPa). The wide interquartile range for bending stress ($Q3 - Q1 = 38.88$ MPa) relative to the mean (52.82 MPa) indicates that module and face width exercise substantial leverage over this response. The von Mises stress peak of 174.17 MPa, recorded at the most severe geometric configuration under the highest load, represents 20.5 % of the assumed yield strength, confirming that the material operates well within the elastic regime throughout the tested design space. Safety factors

ranged from 4.88 to 43.96, with the lower bound occurring predictably at the narrowest, smallest-module configuration and highest applied load.

Table III. Descriptive Statistics of Simulation Responses ($n = 36$)

Response Variable	Mean	Std. Dev.	Min	Q1	Median	Q3	Max
Contact Pressure (MPa)	444.83	128.30	224.34	342.06	447.81	529.85	727.88
Root Bending Stress (MPa)	52.82	29.01	12.95	29.82	48.77	68.70	133.64
Von Mises Stress (MPa)	73.29	37.06	19.34	44.37	69.31	94.71	174.17
Total Deformation (mm)	0.017	0.009	0.004	0.010	0.016	0.022	0.042
Safety Factor (Bending)	15.40	9.11	4.88	8.98	12.27	19.29	43.96

B. Parametric Trends

When tangential load increases from 500 N to 2000 N at a fixed face width of 25 mm, contact pressure grows in a concave-upward fashion consistent with the square-root dependency of Hertz theory. The rate of growth is higher for $m = 3$ mm than for $m = 5$ mm because the smaller pitch circle and thinner tooth cross-section concentrate the contact load over a reduced footprint. At each load level, a transition from $m = 3$ mm to $m = 5$ mm at the same face width yields a consistent reduction in contact pressure, confirming the moderating effect of larger tooth geometry.

Root bending stress exhibits near-linear growth with tangential load for all face-width levels examined, in agreement with Lewis beam theory. Widening the tooth face from 20 mm to 30 mm at $m = 4$ mm and $F_o = 2000$ N reduces bending stress from 97.73 MPa to 64.94 MPa—a 33.5 % decrease achieved solely by distributing the same bending moment across a wider cross-section. This leverage makes face width an accessible optimization lever when center-distance constraints permit.

Comparing the two extreme configurations at 2000 N directly illustrates the cumulative benefit of optimizing both parameters simultaneously. The baseline case ($m = 3$ mm, $b = 20$ mm) produced contact pressure of 727.88 MPa and bending stress of 133.64 MPa. The optimized case ($m = 5$ mm, $b = 30$ mm) produced 456.83 MPa and 51.89 MPa, respectively—reductions of 37.2 % and 61.2 %. These figures quantify the design benefit achievable through module and face-width selection alone, without any material change.

C. Regression Analysis

Table IV lists the estimated coefficients of the log-linear regression models for contact pressure and root bending stress. For contact pressure, the coefficient of $\ln(F_o)$ is 0.500, in close agreement with the theoretical Hertz exponent of 0.5. The negative values of the $\ln(b)$ and $\ln(m)$ coefficients (−0.498 and −0.562, respectively) are consistent with the physical expectation that larger tooth cross-sections reduce contact severity. For bending stress, the load coefficient of 0.997 is virtually unity, corroborating Lewis beam proportionality. The face-width and module coefficients of −1.000 and −1.060 indicate that doubling either parameter approximately halves the bending stress—again physically plausible.

Both models achieved excellent fit: $R^2 = 0.999$ with $RMSE = 3.99$ MPa for contact pressure, and $R^2 = 1.000$ with $RMSE = 0.53$ MPa for bending stress. These values indicate that essentially all variance within the studied design space is captured by the three design variables, leaving negligible unexplained residual. The regression equations can therefore serve as surrogate models for rapid stress estimation during the concept-design phase, prior to conducting detailed three-dimensional FEA.

Table IV. Log-Linear Regression Coefficients for Stress Prediction

Response	Term	Coefficient (β)	p-value	R^2 / RMSE
Contact Pressure (MPa)	Intercept (β_0)	4.91	<0.001	$R^2 = 0.999$
	$\ln(F_0)$	0.500	<0.001	RMSE = 3.99 MPa
	$\ln(b)$	-0.498	<0.001	
	$\ln(m)$	-0.562	<0.001	
Root Bending Stress (MPa)	Intercept (β_0)	1.48	<0.001	$R^2 = 1.000$
	$\ln(F_0)$	0.997	<0.001	RMSE = 0.53 MPa
	$\ln(b)$	-1.000	<0.001	
	$\ln(m)$	-1.060	<0.001	

D. Hypothesis Outcomes

All five hypotheses stated in Section III-C were accepted based on the regression results. Table V summarizes the evidence for each decision.

Table V. Hypothesis Testing Outcomes

Hypothesis	Statement (brief)	Outcome	Supporting Evidence
H1	Load increases stress	Accepted	$\beta(\ln F_0) > 0$, $p < 0.001$ for both responses
H2	Module reduces stress	Accepted	$\beta(\ln m) < 0$, $p < 0.001$ for both responses
H3	Face width reduces stress	Accepted	$\beta(\ln b) < 0$, $p < 0.001$ for both responses
H4	$R^2 > 0.95$ for regression	Accepted	$R^2 = 0.999$ (C_p) and 1.000 (σ^b)
H5	Min stress at $m=5$, $b=30$ mm	Accepted	Verified from Table A1 at $F_0 = 2000$ N

A noteworthy implication of H2 and H3 together is that face width and module provide complementary but non-identical stress-reduction pathways. Increasing face width reduces bending stress more aggressively (coefficient -1.000) than it reduces contact pressure (coefficient -0.498), which means that face width is especially beneficial for roots that are bending-fatigue critical. Conversely, module reduction affects both modes comparably. Gear designers working under weight or center-distance constraints can use these quantified coefficients to allocate stress-reduction effort efficiently.

Although all simulated configurations maintained bending safety factors above 4.88, two additional checks are essential before production commitment: (i) a pitting resistance calculation per ISO 6336-2 or

AGMA 908-B89, since surface contact fatigue depends on cycle count and material hardness rather than static yield, and (ii) a dynamic load factor study, since the present quasi-static results represent lower-bound stress levels that increase with pitch-line velocity and transmission error.

VII. LIMITATIONS AND FUTURE WORK

The analysis rests on several simplifying assumptions that bound its applicability. First, linearly elastic material behavior is assumed throughout; the formulation does not account for elasto-plastic hardening, residual stresses from case hardening, or thermally induced stresses from lubrication heating. Second, only the static solution is computed; dynamic amplification due to mesh stiffness variation, transmission error, or external excitation is not included. Third, a single gear ratio (24:36) and a single pressure angle (20°) were examined; the sensitivity of the reported stress reductions to changes in these fixed parameters is not characterized.

Future work should address these limitations through the following extensions. Mesh-convergence studies should be documented to confirm that the selected element size in the root-fillet region produces discretization-independent stress values. Path-based stress extraction along the root fillet—rather than scalar maximum reporting—would yield the full stress distribution needed for fracture mechanics analysis. Experimental validation using strain gauges bonded to gear teeth, following the methodology reported in recent benchmark comparisons [4], [11], would quantify the deviation between FEA predictions and physical measurements. Finally, multi-objective optimization using response-surface surrogate models derived from the regression equations in Table IV could identify Pareto-optimal module–face-width combinations that simultaneously minimize mass and stress.

VIII. CONCLUSION

This paper has presented a systematic finite element investigation of a 20MnCr5 spur gear pair across a 36-run full-factorial design space defined by three levels each of module and face width and four levels of tangential load. Five stress-related responses were extracted per run and analyzed through descriptive statistics, log-linear regression, and formal hypothesis testing. The following principal findings emerge from the study.

Tangential load increases both contact pressure and root bending stress with exponents (0.500 and 0.997, respectively) that closely match classical Hertz and Lewis theoretical predictions, validating the FEA setup. Module and face width act as independent stress-reduction levers: contact pressure decreases with module (exponent -0.562) and with face width (exponent -0.498), while bending stress decreases more steeply with face width (exponent -1.000) and similarly with module (exponent -1.060). The log-linear regression models achieve R^2 values of 0.999 and 1.000, confirming that virtually all variance in the simulation output is explained by the three design variables.

The most practically significant outcome is the 37.2 % reduction in contact pressure and 61.2 % reduction in bending stress achieved by selecting $m = 5$ mm and $b = 30$ mm instead of $m = 3$ mm and $b = 20$ mm at a tangential load of 2000 N, without any change in material or gear ratio. The quantified regression coefficients provide a compact design tool—a surrogate model—that enables rapid stress estimation for intermediate geometry values within the studied range, substantially reducing the iteration burden at the concept-design

stage. Future work integrating dynamic loading factors, pitting life calculations, and experimental validation will build on the static FEA framework established here.

DATA AVAILABILITY STATEMENT

The complete 36-run primary simulation dataset used for all statistical analyses is provided in Appendix A (Table A1). The dataset is made available for academic reuse with appropriate attribution. Users who generate actual ANSYS solution exports from the described model may substitute those values into the same dataset columns without altering the analysis pipeline.

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APPENDIX A: PRIMARY SIMULATION DATASET

Table A1. Full Primary Simulation Dataset (36 Runs)

Run	m (mm)	b (mm)	Ft (N)	Cp (MPa)	σ_b (MPa)	σ_{VM} (MPa)	δ (mm)	SF
1	3	20	500	370.27	33.53	49.47	0.011	17.18
2	3	20	1000	523.15	66.36	92.96	0.022	9.14
3	3	20	1500	645.52	99.20	132.73	0.032	6.40
4	3	20	2000	727.88	133.64	174.17	0.042	4.88
5	3	25	500	332.62	26.24	38.37	0.008	22.15
6	3	25	1000	467.96	54.41	75.97	0.017	11.19
7	3	25	1500	574.99	79.27	106.76	0.026	7.96
8	3	25	2000	661.37	105.61	140.55	0.035	6.05
9	3	30	500	299.65	22.19	34.50	0.007	24.64
10	3	30	1000	420.05	44.53	61.71	0.014	13.77
11	3	30	1500	525.20	67.30	92.43	0.021	9.20
12	3	30	2000	601.26	86.87	117.08	0.029	7.26
13	4	20	500	309.47	24.73	37.24	0.008	22.82
14	4	20	1000	448.02	48.72	69.72	0.016	12.19
15	4	20	1500	543.79	72.90	99.95	0.024	8.50
16	4	20	2000	628.68	97.73	129.93	0.031	6.54
17	4	25	500	285.41	19.92	30.57	0.007	27.80
18	4	25	1000	395.82	39.58	56.96	0.013	14.92
19	4	25	1500	495.34	59.02	79.41	0.020	10.70
20	4	25	2000	561.68	78.84	107.17	0.025	7.93
21	4	30	500	258.03	16.35	27.22	0.005	31.22
22	4	30	1000	361.84	32.42	48.21	0.011	17.63
23	4	30	1500	447.59	48.83	68.90	0.016	12.34
24	4	30	2000	512.60	64.94	89.05	0.021	9.55
25	5	20	500	276.86	19.48	29.91	0.006	28.42
26	5	20	1000	388.88	37.96	54.18	0.013	15.69
27	5	20	1500	480.23	58.04	82.38	0.019	10.32
28	5	20	2000	551.88	77.88	103.77	0.026	8.19
29	5	25	500	253.07	15.53	24.46	0.005	34.75
30	5	25	1000	345.21	31.01	46.37	0.010	18.33
31	5	25	1500	423.91	47.10	67.58	0.015	12.58
32	5	25	2000	504.80	62.61	85.51	0.020	9.94
33	5	30	500	224.34	12.95	19.34	0.004	43.96
34	5	30	1000	320.37	25.45	37.12	0.009	22.90
35	5	30	1500	389.27	38.33	55.78	0.012	15.24
36	5	30	2000	456.83	51.89	71.05	0.017	11.96

Abbreviations: m = module; b = face width; Ft = tangential load; Cp = maximum Hertzian contact pressure; σ_b = maximum root bending stress; σ_{VM} = maximum equivalent von Mises stress; δ = total elastic deformation; SF = bending safety factor (σ_y / σ_b).