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New Design Formulae for Safety and Precision in the Fatigue Engineering of Mechanical Components and Structures

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Abstract

This paper introduces two new formulae, termed the Nori Fatigue Formulae, for determining the maximum allowable fatigue stress in mechanical components and structures with significant stress concentrations. These formulae will eliminate the usage of code-sensitive safety factors along with other factored values from the entire mechanical design engineering work, ranging from a safety pin to a spacecraft. The first formula gives the maximum allowable fatigue stress in tension, and the second formula gives the maximum allowable fatigue stress in shear. These formulae are established by methodically using all the vital mechanical properties of steel, ultimate tensile strength (T), yield strength (Y), reduction of area (R), and elongation (E), as obtained from the engineering stress–strain curve generated by the globally standardized tensile testing. This framework enables effective utilization of the vast amount of tensile test databases already available worldwide, particularly reduction of area and elongation, for computing the maximum permissible fatigue stress of a given material in the given condition and application. Consequently, the proposed approach eliminates the need for generating additional experimental data, which often entails substantial investments of time and financial resources. Such a unified approach, previously unavailable, is essential in view of the divergent design philosophies and design factors adopted across international design codes, standards, and handbooks over time. To make this paper more comprehensive, several new formulae are provided for use in various design situations routinely encountered. Also provided additional applications for the formulae offered in the first two papers. Thus, this paper fully encompasses the previous two papers of this new-design-formulae series.

Keywords: Allowable stress, design factor, design stress, failure stress, fail-safe precise design, factor of safety, maximum allowable fatigue stress, maximum permissible stress, maximum safe stress, maximum working stress, safety margin

INTRODUCTION

Like humans and other forms of life, all mechanical components, machines, and structures must die one day. However, this stance does not exempt mechanical designers from their prime responsibility of ensuring the required level of safety, performance, durability, and cost-effectiveness, even though design engineering activities suffer from several deficiencies and uncertainties, as well as a great variety of component geometries, metals, manufacturing processes, heat treatments, load cycles, loading types, and service conditions [1–5]. Despite the best efforts of the engineering and research communities through new theories, approaches, and design philosophies, fatigue-caused failures and failure-caused tears have not disappeared. In addition to this despair, damage to society and investments has been increasing rapidly owing to the ever-increasing number of humans,

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automobiles, aircraft, machinery, structures, boilers, piping, and other equipment. In view of this agonizing situation, a new fatigue design approach is formulated to provide a scientifically convincing solution to this devastating problem that has been erasing the smiles of people innocently depending on the fatigue-failure-prone aircraft, vehicles, machines, structures, and other tools that are surprisingly failing even within the stipulated guarantee period. To support society and industry, in this study, new fatigue formulae called Nori Fatigue Formulae (NFF) for safe and precise design engineering work are presented, which also carry several other direct and indirect benefits [6–10]. The NFF, created by the Nori Design Philosophy and Nori Material Philosophy, carries all the required potential to put an enduring end to the fatigue failures identified by the world as early as 1829. Research in this direction has become necessary because human life losses and economic losses are escalating day by day and severely affecting the technological world since the birth of mechanical elements and machines. NFF can also be called the Nori tire formula because T, Y, R, and E are applied to derive these formulae [11–17].

These universally applicable formulae Nori fatigue formulae (NFF) ensure the most sought-after relationship between the fatigue properties and the fundamental mechanical properties of steel, thus inspiring their ready adoption by worldwide engineering communities, standards organizations, and international code authorities for mechanical design, materials, testing, inspection, validation, certification, reliability assurance, and public safety [18–23]. Primarily, NFF are developed (1) to eliminate the factor of safety (safety margins or design margins) from the design process, (2) to eliminate design problems and risks associated with the present practice of determining the maximum allowable fatigue stress through the application of a factor on a single property, such as ultimate tensile strength or yield strength, and (3) to eliminate the costs, time, complexities, problems related to the simulation of service loads and conditions, and other ambiguities of creating S–N curves, S–N–P curves, P–P plots, or Q–Q plots [24–31].

CONTRIBUTIONS TO STRESS, STRAIN, AND FATIGUE ENGINEERING

Fracture phenomena were studied, and it was stated that the strength of columns varies inversely as their lengths, but directly as some ratio of their cross sections. It was later observed that rods under tension had strength proportional to their cross-sectional area. The laws of elasticity were established, followed by the publication of the laws of motion. The theory of beams was developed through contributions that advanced structural understanding. The modulus of elasticity was introduced as a constant defining the stiffness of a material within the elastic limit. Equations for normal stresses in beams under bending were formulated. The concept of the “zero-line” of mechanical stress was determined, correcting earlier findings. A general theory of elasticity was then expressed in mathematical form and applied to construction. The terms “stress” and “strain” were introduced, with stress defined as pressure per unit area and strain as the ratio of change in length to original length. It was recognized that elastic modulus is a material property independent of the second moment of area, leading to the foundation of modern structural analysis. Poisson’s ratio was published. Fatigue failures due to repeated loading were identified, and early failure reports were documented. To prevent such failures and extend service life, twisted steel wire rope was invented and widely used. The first article on fatigue failures was published, along with the development of testing machines for wire ropes. The phenomenon of metal fatigue was further explained in lectures. The effect of stress concentrations was recognized through investigations of structural failures. Funding was later provided for studying the effects of load changes on iron structures to improve safety. The term “fatigue” was formally introduced in reports on mechanical failures. Finally, systematic fatigue testing methods were established.

The concept of endurance limit was introduced through reports on fatigue failure of railway axles. A mean stress equation using tensile strength was presented, followed by another similar approach also involving tensile strength. Microscopic cracks were identified as the origin of fatigue failures. The first version of a well-known book on strength of materials was published, and a definition for true strain (natural strain) was proposed. A log-log relationship for S-N curves was developed using earlier test data, leading to the introduction of a law describing fatigue behavior. The terminology of intensive and extensive properties for materials was introduced, along with another mean stress equation. Important work on fracture mechanics was published, marking the birth of this field. Improvements in beam theory were presented, and earlier beam equations were corrected. An additional constraint, known as the safe design region for fatigue limit and yield strength, was introduced for ductile materials. A rule for cumulative damage was proposed and later promoted as a linear damage hypothesis for design purposes. Experiments on fretting fatigue were conducted for the first time. A mean stress approach based on yield strength was proposed. An English version of a major strength of materials text was published. A work-hardening theory was introduced. Finite fatigue life was discussed as a design goal. An end-quench test for evaluating hardenability was introduced. Fretting was identified as a mechanical process involving micro-slip. It was established that fretting reduces the fatigue strength of metals.

A method for hardenability calculation was proposed. An organization dedicated to the advancement of stress and strain analysis and related technologies was founded. Recommendations were made regarding the number of test specimens required for constructing S-N curves. The occurrence of adhesion in fretting contact was discovered. An equation was developed to explain fatigue crack growth in terms of plastic strain at the crack tip. A publication on stiffness and deflection of complex structures introduced the term “finite-element method” and provided its first comprehensive treatment. The concept of the stress-intensity factor was established, contributing significantly to fracture mechanics. Methods were proposed for predicting the growth rate of fatigue cracks. Experimental analysis introduced the crack opening displacement (COD) theory, followed by the development of the crack tip opening displacement (CTOD) concept. A formula for mean stress calculation was presented. Scientific investigation into fretting fatigue began. The J-integral concept was introduced to study elastic-plastic fracture mechanics. The rain-flow counting algorithm was developed for use with linear damage hypotheses. A general formulation for nonlinear elastic frames applicable to inelastic cases was proposed. Fretting was identified as a three-stage process, and explanations were provided for how plastic deformation at the crack tip can reduce fatigue crack growth rate due to a wedging effect. Structural specifications based on damage tolerance, assuming the existence of crack-like defects in critical areas, were introduced. A uniform material law was proposed based on extensive fatigue data. It was identified that many failures in gas turbine engines of military jet aircraft were due to high-cycle fatigue. A standard on Failure Mode and Effects Analysis was published. A new hardenability test and related methodologies, specifications, and terminology were introduced. Design formulae were developed for calculating maximum allowable design stress without using a factor of safety for non-fatigue conditions. Additional formulae were introduced for calculating maximum allowable fatigue stress without using a factor of safety for components and structures with insignificant stress raisers.

Thus, the much-needed war waged by research and engineering communities on fatigue-caused catastrophic failures is fully energetic even today, as the cost of fatigue failures reached 4% of the GDP of a nation, in addition to the countless loss of precious lives. Therefore, as a consequence of every fatigue-caused accident, it became a routine practice, across the globe, to set up experimental and research programs related to fatigue. Since World War II, researchers and other investigating teams have generated vast amounts of data, and by the turn of the 20th century, the list of research publications and reports had crossed 100,000 on fatigue failures that killed thousands of people and damaged investments running into billions of dollars. Unfortunately, this painful story is uninterruptedly running to date, even though commendable efforts and considerable money are constantly going into the process of engineering, design verification, refinement of design philosophies, failure theories, and fracture mechanics, manufacturing processes, inspection techniques, testing procedures, and validation methods to derive safe and reliable design practices. As rightly established by numerous research studies, fatigue strength, by all means, must be kept below the yield strength of the material. However, how much below became the first tormenting question, how accurately we arrived there became the second question, and how quickly we arrived there became the third question. The fourth and the biggest question that immediately arises is how to convince the global authorities of the methodology chosen to arrive at the maximum allowable fatigue stress (strength) of a given material in the given condition for a given application. These billion-dollar questions have been standing upright in front of us, almost ever since

1829, the year in which metal fatigue was identified and subsequently regarded as a major threat to human life and investment.

THE COMPELLING NEED FOR FAILURE PREVENTION

Ever since the identification of fatigue failure and prevention of metal-fatigue, it has become the first and foremost demand of our rapidly growing technological world, as metal-fatigue possesses enormous potential to cause catastrophic failures that, in turn, cause agonizing human loss, investment loss, and other dire consequences in society. Further, these difficult-to-predict failures, in which the crack propagation rate can be as high as 2,500 m/s (i.e., 7.3 times the velocity of sound in air at room temperature), are demoralizing the scientific and engineering communities and decelerating technological and economic progress by severely reducing the morale of engineers and scientists. In view of these social and economic problems, the demand for a simple, effective, and efficient solution has become the most compelling need of the hour. *All said and done, prevention of fatigue failures is the only way, the easiest and surest, to arrest catastrophic failures along with their unbearable consequences, both on society and technological advancement. Since prevention is better than cure, and nothing in life is more valuable than life itself, assured reliability and surety for safety must alone form the two wings of the mechanical design engineering process.* Unfortunately, despite substantial efforts and huge investments of time and money, our technological world is still at a crossroads and eagerly waiting for a simple, logical, and scientifically convincing solution to this distressing problem, even though there is enough to be done in the areas of understanding fatigue mechanisms and material behavior under fluctuating stresses and changing environments that cannot be simulated in a laboratory to develop the data that is crucial for producing fatigue-failure-free designs.

GENERAL PROBLEMS OF THE DESIGNER.

Although humans inevitably make mistakes, they strive for perfection. The mechanical design process invariably demands zero errors to eliminate fatigue-induced accidents. This, in turn, demands complete and error-free inputs, defect-free materials, zero stress-raisers, zero residual stresses, defect-free welding, defect-free heat treatment, flawless manufacturing (machining, forging, casting, rolling, etc.), zero overload, zero abuse, zero misuse, skillful operation, timely maintenance, complete and accurate data on service conditions, and so on. This list of designer needs and genuine aspirations virtually goes on and on. Unfortunately, none of these crucial factors are in the hands of the designer, yet the designer is under pressure to ensure all the required levels of safety, economy, performance, durability, reliability, timely and positive damage detectability, compactness, convenience, aesthetics, and profitability of the structure. In addition, failure mechanisms, which are sensitive to numerous parameters, have yet to be understood to the required extent by the scientific and research communities. In addition to the aforementioned responsibilities, designers must also consider other important factors, such as recyclability, environmental laws, compatibility, financial viability, and market demands. With these and several other design problems in mind, we are certainly compelled to look at other important aspects, such as the material and manufacturing sides, to meet the safety and reliability requirements that the public has been demanding.

PROBLEMS FROM MATERIALS

Initially, it was assumed that the material was homogeneous and isotropic. Second, the materials are seldom clean and defect-free. Third, the working stresses are rarely predicted by the designer. Fourth, the strength (tensile strength, compressive strength, yield strength, shear strength, torsional strength, bending strength, etc.) measured in the laboratory is not a single property because the specimen is not subjected only to the visible forces, although we customarily call them tensile, compressive, shear, or bending forces. From a fatigue perspective, the elastic region of the stress–strain curve governs the performance, whereas the plastic region of the curve contributes to damage tolerance and protection against failure. Surprisingly, most high-strength steels, regarded as ductile in the uniaxial tensile test, fail in other tests without appreciable macroplastic deformation; this is often attributed to the difference in the micromechanics of the separation of brittle and ductile fractures. Therefore, it is safe to design mechanical elements, machines, and structures with materials that offer adequate elastic deformation (maximum Y/T ratio of 0.8165, rounded off to 0.8, as observed by the author) because ductility, a valuable measure of plastic deformation (yielding), must be maintained at a level that ensures safety against fatigue and other difficult-to-detect fracturing modes that result in catastrophic failures.

HURDLES IN FATIGUE DESIGN

Even though all possible design care, technologically advancing materials (with better control on crack propagation), modern testing methods, and best overall efforts are continuously reaching new heights, failures do occur due to:

1. Design deficiencies arise from unpredictable loads and loading conditions that cause undue stress, strain, deformation, local tensile stress, and local plastic strain.
2. Design inputs are erroneously assessed and extracted from various design codes, standards, handbooks, textbooks, and research publications.
3. Test data and failure theories, such as the maximum normal stress theory (Rankine theory), maximum shearing stress theory (Tresca–Guest theory), maximum normal strain theory, total strain theory (Beltrami theory), distortion energy theory (Huber–Von-Mises Hencky–s theory), Mohr's failure theory, Griffith theory, Marco-Starkey cumulative damage theory, Henry cumulative damage theory, Gatts cumulative damage theory, Marin cumulative damage theory, Larson–Miller theory, Manson–Hafered theory, Corten–Dolan cumulative damage theory, Goodman's linear relationship, Gerber's parabolic relationship, Soderberg's linear relationship, the elliptical relationship, and Manson's double linear damage rule, are sensitive to numerous assumptions and are often subjected to limitations.
4. Strain amplitude versus fatigue life relations,
5. geometric intricacies; unavoidable stress raisers; and varying material properties and defects, including defect type, size, location, orientation, and material cleanliness.
6. Non-uniformity in grain size, chemical composition, microstructure, macrostructure, welding heat input, and heat-affected zone (HAZ) properties
7. Effect of residual and other unspecified elements and compounds in steel,
8. Effect of furnace charge elements,
9. Effect of steel-slag reactions,
10. Effect of billet size and reduction ratios,
11. Effects of nonzero mean stress and nonzero mean strain,
12. Effect of loading rates on material properties,
13. Relationship between directional fatigue properties and loading direction, as well as the influence of multiaxial stress states
14. Influence of residual-stress-effect and temperature-effect on design stress,
15. Extreme levels of strength, hardness, and hardness-differentials,
16. Manufacturing imperfections, such as those resulting from heat treatment and finishing operations, arise from uncontrollable and inconsistent activities.
17. Work-hardening and its varying behaviors,
18. Unpredictable effects of combined stress,
19. Varying residual stresses (mechanical, metallurgical, and thermal) resulting from numerous manufacturing processes.
20. Unstable ratio of residual stresses to applied stresses,
21. Unavoidable variations, i.e., inconsistency in dimensions and fits due to permissible tolerances,
22. Undetected, unrecorded, and ignored deviations from specified design/drawing limits,
23. Unnoticed assembly defects,
24. Testing deficiencies (problems in simulating real service-load/cycles and environment)
25. Inspection errors and instrument errors,
26. Operation and maintenance defects,
27. fluctuating time-temperature-stress controlled material conditions,
28. Records of quality rating/defect-level-rating that are not person-insensitive.

STRESS-RAISERS AND THEIR INEVITABILITY

In effect, every defect in a material is a stress-raiser, and defect-less material, in the real world, is a myth. Similarly, a design without a stress raiser is a myth, since section changes, bolt holes, rivet holes, threaded holes, slots, grooves, keyways, splines, welding, machining, heat treatment, etc., are unavoidable. In addition, manufacturing without causing stress is a myth. Thus, stress-raisers can be regarded as inevitable. Regrettably, no single person or cause can be pinpointed for lapses and failures that surface during investigations; hence, corrective actions are required. In a nutshell, design, manufacturing, testing, inspection, validation, and reliability assurance are processes filled with endless uncertainties and limitations of widely varying degrees. Thus, the question of safety and reliability becomes as complex as possible, as the design engineering process is directly linked with human life loss and investment loss. Therefore, the design process demands a new, simple, and effective approach for the efficient and early elimination of fatigue failures that account for nearly 95% of all mechanical service failures. Here, it is heartbreaking to note that the USA, in 1982, placed the cost of fatigue failures at $119 billion per year, i.e., 4% of its then national income, and today it stands at $907 billion per year. Owing to these and several other difficulties with safety margins that are also frequently subjected to changes, the author has developed the Nori Fatigue Formulae to ensure the required level of safety against fatigue failures and eliminate the factor of safety (FOS), which also results in disagreements, under-designs, and over-designs that attract a host of other serious problems from multiple angles. Since the introduction of structural, boiler, pressure vessel, nuclear, piping, automobile, and other mechanical design codes and standards by reputed international standards organizations (ASME, SAE, ASCE, API, BSI, ASTM, DIN, GOST, CEN, ISO, EN, JIS, BIS, GB, etc.) for systematic engineering and safety, the methodology chosen by the design codes (application of a factor of safety or safety margin either on UTS or on Yield or on both) for arriving at the maximum allowable stress has become a topic of disagreement and debate, and therefore, has become a concern for the worldwide design engineering community. This situation has finally led to repeated revisions of codes and standards, as the factor of safety applied in the design process has a direct impact on safety, weight, size, cost, service life, and economy. Considering the difficulties associated with the factored values of the maximum allowable stress, the safety margins are different at different times for a given design code and different for different design codes at any given time. The latter problem caused more concern than the former, which is fairly accepted since it is viewed as necessary by expert committees and justified by the improvements taking place from time to time in design analysis, steel manufacturing technology, and inspection methodology. However, the second problem, which deserves to be viewed as serious, is the inability to be absorbed by the design community in general and by the sensible group in particular because of the lack of uniformity among the international design codes and standards. Thus, the revision of the factor of safety, change of design philosophy, and lack of uniformity among international design codes and standards have become causes of concern, as these problems have direct and indirect effects on international trade, manufacturing practices, and economic progress, even though safety for human life and investment is of prime importance.

THE KEY PROBLEM IN THE FIELDS OF MECHANICAL, STRUCTURAL, AND FATIGUE ENGINEERING

The question of where to operate a given material under given conditions for a given application has become the most bothersome question since the birth of systematic mechanical design engineering. Against the backdrop of this mammoth problem, researchers and failure investigators have observed that nearly 55%–60% of aircraft components and structures fail under fatigue, in addition to 14% under overload. Considering boilers for a similar scenario, 80% of fire-tube boilers and 28% of water-tube boilers were reported to fail because of fatigue. Hence, the Nori Fatigue Formulae will serve as an effective and efficient solution for the global design engineering community. In addition, NFF ensures peace of mind to all those traveling for long hours on ground and off ground equipment.

A REVIEW REQUEST FOR ADDRESSING A LONG-STANDING SAFETY PROBLEM

In view of the proposed safety cap on the Y/T ratio, the author requests international safety and design code authorities to review high-strength ISO bolt classes, such as 10.9 and 12.9, and non-ISO grades, such as 14.9 and 16.9, as well as high-strength and ultra-high-strength steels, such as BS: EN: 10025-6 grades S620, S690, S890, and S960, and non-ISO grades, such as S1100 and S1300, because the Y/T ratio of these HSLA steels, ultra-high-strength steels, and several other routinely used steels is rapidly approaching unity and posing considerable problems to fabrication activity, in addition to presenting a significant threat to public safety. In addition, the sensitivity to stress concentrations varies directly with the material strength. Furthermore, the fracture toughness, which decreases with increasing tensile strength, justifies the proposed safety cap (both Y/T and E/R at a maximum of 0.8) along with several other reasons. In recent years, the use of high-strength materials has increased steeply; therefore, a separate committee is warranted to review high-strength materials.

NORI FATIGUE FORMULAE

The proposed Nori fatigue formulae (NFF)-1 and 2 can successfully eliminate terms such as the factor of safety or safety margin from the entire design engineering activity because the worldwide design community is not fully convinced of the factored values of the maximum allowable stress or maximum permissible stress. For this, the NFF is interlinked and applied to all four basic and derived material properties (tensile strength, yield strength, elongation, reduction of area, yield-to-tensile ratio, elastic constants, etc.) to calculate the maximum allowable fatigue stress of any given steel under any given condition, because all four fundamental properties have unique significance and reveal the overall history of the material. Most importantly, the primary properties (T-Y-R-E) are obtained from the standardized stress–strain curve and are also used to calculate the elastic constants of the material, which are used in a host of mechanical design engineering works. To distinguish the maximum allowable fatigue stress, calculated using the proposed Nori fatigue formula, it is termed NMAFS (Nori Maximum Allowable Fatigue Stress). NMAFS can be calculated effortlessly and accurately using the following formulae: The units for NMAFS are the same as those for the tensile and yield strengths. Additional uses of these formulae are provided in Appendix A.

$$NMAFS_{in\ tension} = \frac{Y^4.E}{T^3.R} \qquad (1)$$

$$NMAFS_{in\ shear} = \frac{Y^5.E}{T^4.R} \qquad (2)$$

T is the ultimate tensile strength in MPa, PSI, KSI, and other units. Y is the 0.2% offset yield strength in MPa, PSI, KSI, etc., R is the reduction in area, in percentage.

E is the elongation in 50.8 mm (2-inch), percentage.

Y/T is the yield-to-tensile ratio, which is dimensionless.

E/R is known as the nori ductility ratio (ratio of the percentage of elongation to the percentage of reduction in area) and is a dimensionless quantity.

For example, the maximum allowable fatigue stress in tension (for the design of components and structures with significant stress raisers) for steel A (tensile strength =876 MPa (127 KSI), yield strength =731 MPa (106 KSI), elongation: 20%, and reduction of area: 60%) is 142 MPa (20.54 KSI). Similarly, for the same material, the maximum allowable fatigue stress in shear (for the design of components and structures with significant stress raisers) is 118 MPa (17.14 KSI). Notably, this value of 118 MPa (17.14 KSI) is equal to the value obtained using the ASME methodology. Thus, the ASME and Nori methodologies are found to be complementary to each other. However, the NMAFS calculated in this study provides stress values that are unequivocally designer- and code-insensitive. Further details of the comparison and differences between the two methodologies are provided in Appendices B, C, and D.

ELEMENTS OF NORI FATIGUE FORMULAE

Both directly and indirectly, for the calculation of the maximum allowable fatigue stress of a given material under the condition, the NFF interlinks and involves all the parts of the uniaxial engineering stress–strain curve, all the regions under the curve, structure-sensitive and structure-insensitive mechanical properties, and elastic and plastic deformations, as all these factors are essential for safety, economy, under-design control, and over-design control, because fatigue strength is not a single property like toughness.

Tensile Strength (T)

The ultimate tensile strength (UTS) or tensile strength (T) is synonymous with the globally established uniaxial tensile test. It is an intensive and structure-sensitive property. It is the highest point (the maximum load a metal can withstand before fracture) of stress on the stress–strain diagram, that is, the point up to which the deformation is uniform. It is the only fundamental mechanical property that the uniform material law (UML) uses for the development of the strain–life curve. It is an appreciated and well-established indicator of hardness (the resistance offered by the molecules of the material against separation by the penetrating action of the external object). It is effortlessly computable, economically measurable, and easily reproducible. It is the most prominent part of the material specification readily obtainable for all materials under all conditions, unlike the yield strength. It is a quick check for material conformance and a valid design parameter for brittle materials, as many materials do not exhibit a clear yield strength. Moreover, Goodman and several other eminent researchers have methodically related the endurance limit (fatigue limit) to the tensile strength. Gerber and Goodman used the tensile strength in their mean stress equations. It is a critical parameter in metal-forming processes because brittleness causes rupture. Globally, many steel and heat treatment conditions are designated by their tensile strength. Importantly, research has correlated tensile strength with the speed of crack propagation. Therefore, this unique and widely used basic mechanical property must be allowed to rule the global material specifications, even though the yield strength has recently gained importance based on logical reasoning.

Yield Strength (Y)

For ductile materials, the yield point is the endpoint for elastic behavior and the beginning point for plastic behavior on the stress–strain curve; that is, below this point, the material returns to its original shape, and beyond this point, the material does not return to its original shape because it undergoes permanent or plastic deformation. Therefore, mechanical, structural, and bolted-joint engineering use this point as the upper limit to avoid catastrophic failures and associated accidents, loss of life, and loss of investment. Thus, the yield point is a well-defined and widely used strength parameter, a structure-sensitive property, the upper practical limit of the load that can be applied without significant deformation (i.e., Hooke's law ending point), a useful measure of resistance to plastic deformation, the most critical property for metalworking, and the limiting elastic strength in all practical engineering applications. Unfortunately, the tensile and yield strengths cannot be related; however, the ratio of yield to tensile (Y/T) or tensile to yield (T/Y) is important for bending, rolling, and forging operations. Steel researchers, heat treaters, mechanical designers, fabricators, metallurgists, materials engineers, and processing communities use this ratio. In the stress–strain diagram, the stress corresponding to a strain of 0.002 is termed the yield stress, even though other strain values are used. The Hall–Petch equation relates the yield strength to grain diameter.

Reduction of Area (R)

A structure-sensitive property, a better-rated ductility parameter resulting from the most complicated triaxial stress state of the necking process, the most important measure of deformation required to produce fracture, the first valuable measure of ductility that carries unusual significance, a sensible parameter for the evaluation of quality and structural variations in the material, a valued parameter (reduction of area in the through-thickness direction, that is, Z-direction) used to correlate resistance to lamellar tearing, a parameter that is rapidly gaining special attention owing to its relatively lower value in the through-thickness (short-transverse) direction compared to the longitudinal or transverse directions, and finally, a sure and simple candidate for effective feedback for process improvements. Although the values of elongation and reduction of area serve as valuable measures of ductility and toughness, they represent entirely different types of material behavior, which are important for mechanical and civil engineering designers, as well as material property designers. Owing to the complexity of the necking process, the reduction in area and elongation cannot be related; similarly, tensile strength and yield strength cannot be related. Thus, material research communities are forced to deal separately with these four fundamental mechanical properties (tensile strength, yield strength, elongation, and reduction of area).

Elongation (E)

It is a globally standardized and easily measurable measure of ductility, a significant parameter offering two different modes (uniform elongation before necking and localized elongation during the process of necking), a structure-sensitive property, the percentage of plastic strain before fracture, a noble indicator of strain-hardening capacity, a second valuable measure of ductility, and a dependable indicator of toughness. The higher the elongation, the higher the toughness and ductility, which are important parameters for bending and shaping operations. Keeping all other failure-causing parameters constant, ductility can be applied readily and safely as an effective tool for the prevention of catastrophic failures because ductility is more of a quality than a property. Supporting this view, research on the crashworthiness of vehicles and aircraft has steadily narrowed down to elongation as a promising risk-reduction parameter. Therefore, the ductility should be as high as possible. Conversely, it must be recognized that the higher the ductility, the higher the component size and weight, and the lower the overall economy, because the maximum allowable stress will be proportionately small. Therefore, the inclusion of both ductility parameters (elongation and reduction in area) is essential in the material design (mechanical property design) process, as ductile failure is the most sought-after failure mode for mechanical and structural design. Interestingly, these ductility parameters also serve as effective quality control measures and steel processing conditions. Because ductility is grain size- and structure-sensitive, it can also serve as a scientific, logical, and profitable tool for fine-tuning the mechanical properties during steel manufacturing, finishing, and heat treatment.

Yield-to-Tensile Ratio (Y/T, the Ratio of Strength Parameters)

It is a recognized indicator of the safety margin and a well-recognized measure of ductility and toughness. In addition, it is a simple method for determining the ability of plastic deformation. An investigation by the author revealed that a material meets all the designer requirements, such as safety, reliability, weight effectiveness, size effectiveness, and cost effectiveness, if the yield-to-tensile ratio (YTR) is maintained at or below 0.80 (0.8165). Incidentally, SAE/ISO bolt property classes 4.8, 5.8, 6.8, 8.8, and 9.8 support this observation. To ensure safety, some national construction codes specify 1.25 as the minimum requirement for the T/Y ratio (i.e., 0.8 Y/T). This code requirement also supports the proposal by the author, because the larger the ratio (T/Y), the greater the energy absorption capacity before failure, that is, an adequate warning before failure. Most importantly, the YTR indicates the distribution pattern of the given material; that is, the ratio in which the material is used. For example, a YTR of 0.8 indicates that 80% of the material's potential is used for the development of strength (yield strength), and the remaining 20% is used for the development of ductility. Recently, the YTR has steadily become a specification requirement in many global standards and assessment documents. If the thrust is higher for the economy, the YTR is higher, and the economy is higher. For this purpose, SAE/ISO bolt property classes 10.9, 12.9, and HSLA structural steels, such as S890 and S960, serve as examples.

Ratio of Elongation-to-Reduction of Area (E/R, the Ratio of Ductility Parameters)

To ensure safety, through ductility, the latest development is to use the ratio of strength parameters, that is, Y/T, as a specification requirement. However, this vital measure will bear fruit only when corroborated with the other ductility parameter, that is, E/R. Hence, to further assure safety, E/R must also be included in steel specifications because it is equally important for safety and will serve as a guaranteed reinforcement for ductility, as ductility is the most sought-after ability to deform plastically without fracturing. Therefore, NFF installed adequate ductility through this double check to safeguard against sudden failures because ductility also supports the redistribution of stress. Additionally, adequate ductility is indispensable for the prevention of lamellar tearing in relatively thicker members. For understandable reasons, mechanical designers continue to suffer from inadequate and inaccurate design inputs, along with materials and manufacturing processes that cannot be defect-free to the extent required in the foreseeable future. To safely and prudently accommodate these genuine problems, we need materials with sufficient ability of forgiveness (ductility), that is, materials with adequate capability to flow plastically before fracturing. Thus, ductility must be given the responsibility of playing the crucial role of protecting from catastrophic failures, accidents, loss of life, and loss of investment arising from inadequate and erroneous design inputs, notches and other stress raisers, material defects, manufacturing defects, testing deficiencies, and inspection errors. Therefore, the inclusion of all ductility parameters (reduction of area, elongation, E/R, and Y/T) is essential for the design of materials, machines, and structures because ductile failure is the most preferred failure mode by designers. Interestingly, these ductility parameters, hitherto unapplied in mechanical and material design processes, also serve as effective quality control measures. The adoption of this concept readily grants everlasting relief to mechanical designers and provides permanent shielding to society and investment against brittle failures. By all means, coming to us in four different forms (E, R, Y/T, and E/R), ductility at an adequate level can alone eliminate these no-warning failures. Since ductility is also grain-size sensitive, this structure-sensitive property will serve as an effective tool for property modifications during material balancing activity. For this research, the ratio of elongation to reduction of area (E/R) is termed as the Nori ductility ratio (NDR) and applied in NMAFS-1 and 2. Beneficially and eventually, the material in the given condition offers the best combination of safety, reliability, performance, durability, weight effectiveness, size effectiveness, and cost effectiveness if the NDR, that is, E/R, is maintained at 0.8 (0.8165 is the maximum limit proposed by the author) with a minimum elongation of 27% in 2-in. Thus, for safely housing the aforementioned problems, the designer certainly needs an all-time helping hand, that is, a material with sufficient capacity of forgiveness (ductility, that is, Y/T and E/R at 0.8 maximum, and elongation at 27% minimum), since the material with adequate ability to flow plastically before fracturing plays a key role in protecting from catastrophic failures and thus accidents, loss of life, and loss of investment.

ADVANTAGES OF NORI FATIGUE FORMULAE

Most importantly, NFF can drastically reduce fatigue testing time and costs and provide a common platform for the worldwide design engineering and materials engineering communities to achieve absolute uniformity in design, which is essential for reducing new product development, facilitating rapid industrial growth, strengthening the global economy, and promoting international trade. Interestingly, the proposed NFF remains the same across all units of measure, product forms, heat treatment, finishing, and surface roughness conditions. Henceforth, the maximum allowable fatigue stress in tension and in shear will be decided by the steel itself and not by the designers, design codes, or standards. This desirable feature of NFF assures the worldwide design engineering communities and places them on a common, logical, convenient, and scientifically convincing platform–the dream desire of the author since 1973. Consequently, the unpleasant debates and disagreements arising from factored values of maximum allowable stress are eliminated because NFF will allow the material to indicate its maximum allowable stress, which will be safe and economical. Thus, for the first time, Nori Fatigue Formulae removed human involvement and successfully enabled the material to communicate its maximum allowable fatigue strength clearly in terms of its safe application in design calculations. Therefore, steel and alloy designers can confidently apply Nori Fatigue Formulae to maximize the strength, ductility, safety, and economy of present and future alloys. As a result, the world also enjoys efficient conservation of natural resources. NFF evaluates and balances even the weld metal and heat-affected zone (HAZ) properties. It is hoped that the Nori Fatigue Formulae, together with the rational and logical philosophy on which it is built, will be readily adopted by national and international standards organizations and authorities in the fields of safety, materials, and design codes and standards, as well as by research centers, educators, and authors of machine design textbooks and handbooks. Such adoption would promote the common cause of uniformity, a rational and unified approach, simplicity, rapid progress, and hassle-free international trade.

Since mechanical design process, to a large extent, is the application of mechanical properties of materials (steels), for the first time, two fundamental strength properties (tensile and yield), two ductility parameters (elongation and reduction of area), yield-to-tensile ratio (an extremely valuable material characteristic steadily entering steel specifications) and another important material characteristic (Nori Ductility Ratio i.e., E/R) are included in the design process to eliminate the present and future problems of mechanical design process, and for installing safe and steady technological progress against catastrophic failures of equipment like vehicles, railroad equipment, boilers, pressure vessels, process piping, ropeways, aircrafts, ships, etc., including movement-supporting bridges and other structures. As we know, to avoid accidents, a good driver must necessarily allow the mistakes of others; similarly, the materials we use must be kind-hearted so that the material will forgive the inevitable design problems, material defects, manufacturing errors, and inspection mistakes for safeguarding us against catastrophic failures and accidents. For this, the ductility of the material must be adequate to safeguard against sudden failures. Thus, the forgiveness (ductility) of the material is used as the bedrock of the new design philosophy called the Nori design philosophy. This viewpoint is strongly supported by the above-cited design, material, and manufacturing problems that cannot be eliminated in the foreseeable future. Moreover, similar to toughness, fatigue strength is not a single property. It has been repeatedly found that the fatigue strength of a given steel under given conditions is essentially the combined strength of all the fundamental properties (T, Y, R, and E) and the derived ones, such as Y/T and E/R. Thus, NFF purposefully involved all six elements to utilize their prominence and uniqueness.

CONCLUSION

NFF has been thoroughly studied for its built-in features and has been extensively verified. It has been found that NFF can be easily and readily adopted in all design areas (highly critical, critical, subcritical, and noncritical/general engineering applications), hitherto adopting various approaches, methodologies, safety factors, design tools, and design philosophies. Hence, the author requests the international design code authorities, national and international standards organizations, authors of handbooks and textbooks, researchers of mechanical and structural engineering, safety authorities, teaching communities, and international trade authorities for early review and adoption of NFF so that fatigue-caused failures, failure-caused accidents, investment losses, and loss of human life can quickly reach the most sought-after infinitesimal levels, as well as for the rapid globalization of markets for manufacturers and nations working with a particular design code or methodology. To sum up, appearing as a simple parameter, the maximum allowable stress of the design engineering activity is silently controlling the global economy (by affecting product economics), public safety (by affecting product reliability), and consumption of natural resources. Therefore, the importance and role of this design parameter need no elaboration. Finally, NFF will effectively reduce the consumption of natural resources, eliminate the crisis, and search for new resources by optimizing product size, weight, and durability. If future technology measures the elastic or proportionality limit precisely, yield (Y) can be replaced by the elastic limit or proportionality limit in the NFF. For this research, the author defined mechanical design engineering as "Understanding the consequences of loads and loading patterns, whether mechanical or thermal, for the given material in the given condition, and for the given application." Although this paper ends here, Appendix A is provided to make this paper comprehensive and to provide additional applications of the formulae provided in the first two papers (Calculate Allowable Stress, Quit Factor of Safety - Nori VSN Murthy, 2013, see reference 61, and Formulae for Safe Design Against Fatigue Failures, (Part-I) - Nori VSN Murthy, 2016, see reference 65).

Competing Interests

The author declares no conflict of interest.

Data Availability Statement

The authors confirm that the data supporting the findings of this study are available in this article.

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APPENDICES

Appendix A

Summary of formulae for various mechanical and structural design applications with varying conditions (Formulae for safe design of components and structures subjected to various kinds of loads, load cycles, and design considerations. These formulae are provided here not only to make this paper comprehensive but also to provide convenience to the mechanical designer.)

1	Nori Formula for calculation of maximum allowable stress in tension, for general design of components and structures under (1) dead/static/steady loads, (2) fewer duty cycles, (3) low pressure and non-coded equipment, and (4) where fatigue and stress-raisers are not design considerations.	$Y^2 . E \div T . R$
2	Nori Formula for calculation of maximum allowable stress, in shear, for general design of components and structures under (1) dead/static/steady loads, (2) fewer duty cycles, (3) low pressure and non-coded equipment, and (4) where fatigue and stress-raisers are not design considerations.	$Y^3 . E \div T^2 . R$
3	Nori Formula for calculation of maximum allowable fatigue stress in tension, for components and structures, where fatigue is a design consideration, but not the stress-raisers, i.e., stress-raisers are considered insignificant.	$Y^3 . E \div T^2 . R$
4	Nori Formula for calculation of maximum allowable fatigue stress, in shear, for components and structures, where fatigue is a design consideration, but not the stress-raisers, i.e., stress-raisers are considered insignificant.	$Y^4 . E \div T^3 . R$
5	Nori Formula for calculation of maximum allowable fatigue stress in tension, for components and structures, where both fatigue and stress-raisers are design considerations.	$Y^4 . E \div T^3 . R$
6	Nori Formula for calculation of maximum allowable fatigue stress, in shear, for components and structures, where both fatigue and stress-raisers are design considerations.	$Y^5 . E \div T^4 . R$
Based on the criticality rating (non-critical, subcritical, critical, and extremely critical) of components and structures, the formula for maximum allowable stress will be as follows.		
7	Nori Formula for calculation of maximum allowable stress in tension, for design of components and structures that are considered non-critical.	$Y^2 . E \div T . R$
8	Nori Formula for calculation of maximum allowable stress, in shear, for design of components and structures that are considered non-critical.	$Y^3 . E \div T^2 . R$
9	Nori Formula for calculation of maximum allowable stress in tension, for design of components and structures that are considered subcritical, i.e., the criticality level is medium.	$Y^3 . E \div T^2 . R$
10	Nori Formula for calculation of maximum allowable stress, in shear, for design of components and structures that are considered subcritical, i.e., the criticality level is medium.	$Y^4 . E \div T^3 . R$
11	Nori Formula for calculation of maximum allowable stress in tension, for the design of components and structures that are considered critical, i.e., the degree of criticality is high.	$Y^4 . E \div T^3 . R$
12	Nori Formula for calculation of maximum allowable stress, in shear, for design of components and structures that are considered critical, i.e., the degree of criticality is high.	$Y^5 . E \div T^4 . R$
13	Nori Formula for calculation of maximum allowable stress in tension, for design of components and structures that are extremely critical, i.e., the degree of criticality is highest.	$Y^5 . E \div T^4 . R$
14	Nori Formula for calculation of maximum allowable stress, in shear, for design of components and structures that are extremely critical, i.e., the degree of criticality is highest.	$Y^6 . E \div T^5 . R$

Notes for Appendix A

The maximum allowable stress obtained from the above universally applicable formulae (1 to 14) can be used directly in the design process. These formulae apply to all steels, in all conditions and product forms, and are valid for any system of units. These formulae can be selected based on the application/criticality level for safeguarding the people and the investment against catastrophic failures caused by under-design, metal-fatigue, and mechanical-property imbalances. The above formulae will also eliminate over-design, which obviously causes overweight, oversize, over-cost, and over-consumption of natural resources. The above formulae apply to all the T-Y-R-E values obtained for all test directions (longitudinal, transverse, and short-transverse), offset yield values, test temperatures, gauge lengths, stress-raiser severity factors, notch sensitivity factors, and surface roughness conditions arising from different manufacturing processes and component geometries. The maximum allowable stress, as given by the above formulae, can be increased—thereby reducing component size, weight, and cost—by applying the principles of the Nori Steel Triangle, the Nori Material Laws, and the Nori Unified Material Ratio, derived from the Nori Material Philosophy, since these ensure that, for a given steel and a given ductility level, the allowable stress is maximized.

Appendix B

Comparison of maximum allowable stress values given by ASME methodology and NORI formula (Maximum allowable stress for design of components and structures where both fatigue and stress-raisers are design considerations)

Steel	Tensile strength (UTS), MPa (KSI)	0.2% offset yield strength, MPa (KSI)	Reduction in area (%)	Elongation in 2-inch, i.e., 50.80 mm (%)	Maximum allowable fatigue stress (shear) as per Nori formula, MPa (KSI)	Maximum allowable fatigue stress (shear) as per ASME methodology, MPa (KSI)	Difference between the ASME stress value and the value given by the Nori Formula
	T	Y	R	E			
Steel-A	876 (127)	731 (106)	60	20	118 (17.15)	118 (17.15)	The difference is theoretically insignificant and practically zero.
Steel-B	2206 (320)	1731 (251)	11	5	298 (43.19)	298 (43.20)	
Steel-C	1600 (232)	1420 (206)	49	12	216 (31.36)	216 (31.32)	
Steel-D	634 (92)	517 (75)	69	26	86 (12.48)	86 (12.42)	
Steel-E	965 (140)	827 (120)	58	17	131 (18.99)	130 (18.90)	
Steel-F	669 (97)	559 (81)	30	10	91 (13.13)	90 (13.10)	
Steel-G	731 (106)	565 (82)	31	15	98 (14.21)	99 (14.31)	
Steel-H	2013 (292)	1682 (244)	36	12	273 (39.65)	272 (39.42)	

Notes for Appendix B

Design Methodology

According to the ASME design code, the permissible fatigue stress (shear stress) for components with stress raisers (e.g., shafts with keyways, holes, threaded holes, and splines) is 30% of the yield strength in tension or 18% of the ultimate tensile strength, whichever is minimum × 0.75. This proven computation methodology was applied to steels A–L to calculate the maximum allowable stress. The Nori Formula used for calculating the comparable stress is Y^5. E ÷T^4.R, please see serial number 6 in Appendix A.

Discussion

The ASME methodology considers two strength parameters (ultimate tensile strength and yield strength) and applies one parameter (offering the minimum value) with multiplying factors (safety factors/safety margins) to compute the maximum permissible stress. In contrast, the Nori formula applies to all four basic parameters (T, Y, R, and E, i.e., both the strength and ductility parameters). In addition to these four parameters, the Nori formula applies to both ductility ratios (Y/T and E/R ratios). Importantly, the Nori formula does not involve numbers and digits that are prone to reviews, changes, debates, and disagreements. Thus, the Nori formula delivers higher accuracy for stress values by incorporating at least six parameters, unlike the ASME approach, which relies on only one. This significant improvement in design efficiency benefits global engineering by providing safer and more efficient designs.

Observation

The above comparison (Appendix B) indicates that the difference between the two methodologies (ASME and NORI) is minimal, thus prompting the ready adoption by all national and international design code authorities.

Conversion of Units

1 KSI = 6.89476 MPa, 1 PSI = 0.00689476 MPa, 1 KSI = 1000 PSI, 1 MPa = 145.038 PSI, 1 MPa = 0.145038 KSI

Appendix C

Comparison of maximum allowable stress values given by ASME methodology and NORI formula (Maximum allowable stress for design of components and structures where fatigue is a design consideration, but not stress-raisers, i.e., the effect of stress-raisers is considered insignificant)

Steel	**Tensile strength (UTS), MPa (KSI)**	**0.2% offset yield strength, MPa (KSI)**	**Reduction in area (%)**	**Elongation in 2-inch, i.e., 50.80 mm (%)**	**Maximum allowable fatigue stress (shear) as per Nori Formula, MPa (KSI)**	**Maximum allowable fatigue stress (shear) as per ASME Methodology, MPa (KSI)**	**Difference between the ASME stress value and the value given by the Nori Formula**
	T	Y	R	E			
Steel-a	1351 (196)	1207 (175)	60	17	243 (35.29)	243 (35.28)	The difference is theoretically insignificant and practically zero.
Steel-b	1393 (202)	1089 (158)	27	13	251 (36.40)	251 (36.36)	
Steel-c	565 (82)	448 (65)	35	16	102 (14.80)	102 (14.76)	
Steel-d	786 (114)	655 (95)	64	24	142 (20.62)	142 (20.52)	
Steel-e	448 (65)	331 (48)	59	36	81 (11.79)	81 (11.70)	
Steel-f	827 (120)	724 (105)	69	21	148 (21.41)	149 (21.60)	
Steel-g	972 (141)	827 (120)	58	20	176 (25.51)	175 (25.38)	
Steel-h	552 (80)	414 (60)	35	20	100 (14.46)	99 (14.40)	

Notes for Appendix C

Design Methodology

According to the ASME design code, the permissible fatigue stress (shear stress) for components with stress raisers (e.g., shafts with keyways, holes, threaded holes, and splines) is 30% of the yield strength in tension or 18% of the ultimate tensile strength, whichever is lower. This computational methodology was applied to steels A–L to calculate the maximum allowable stress. The Nori Formula used for calculating the comparable stress is Y^4. E T^3.R; please refer to serial No. 4 in Appendix A.

Discussion

The ASME methodology considers two strength parameters (ultimate tensile strength and yield strength) and applies one parameter (offering the minimum value) with multiplying factors (safety factors/safety margins) to compute the maximum permissible stress. In contrast, the Nori formula applies to all four basic parameters (T, Y, R, and E, i.e., both the strength and ductility parameters). In addition to these four parameters, the Nori formula applies to both ductility ratios (Y/T and E/R ratios).

Importantly, the Nori formula does not involve numbers and digits that are prone to reviews, changes, debates, and disagreements.

Thus, the Nori formula delivers higher accuracy for stress values by incorporating at least six parameters, unlike the ASME approach, which relies on only one. This significant improvement in design efficiency benefits global engineering by providing safer and more efficient designs for various structures.

Observation

The above comparison (Appendix C) indicates that the difference between the two methodologies (ASME and NORI) is minimal, thus prompting the ready adoption by all national and international design code authorities.

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Conversion of Units

1 KSI = 6.89476 MPa, 1 PSI = 0.00689476 MPa, 1 KSI = 1000 PSI, 1 MPa = 145.038 PSI, 1 MPa = 0.145038 KSI

Appendix D

Comparison between the ASME methodology and the NORI methodology

No.	ASME methodology for arriving at the maximum permissible stress value.	NORI methodology for arriving at the maximum permissible stress value.
1	Consider only two parameters (ultimate tensile strength and yield strength) and apply just one (the minimum value). This yields a design efficiency of 1/6, or 16.67%.	To enhance safety, performance, and precision in engineering, six parameters were applied: two strength parameters (T and Y), two ductility-related parameters (R and E), and two vital ratios (T/Y and E/R). This approach achieves full design efficiency (6/6, or 100%).
2	Rely on numerical values prone to review, revision, debate, and disagreement.	Avoids numerical values entirely, eliminating such issues.
3	The accuracy of the maximum allowable stress value depends on human judgment, designer experience, source reliability, and the factor of safety (FOS).	There is no human involvement. As such, there is no scope for this kind of issue.
4	Final stress values remain sensitive to sources and designer choices due to varying FOS.	No scope for this kind of problem.
5	The FOS varies due to code revisions, differing over time within codes and across codes at any given time. This variability frustrates global design communities and equipment users.	Formulas given in Appendix A will eliminate all these problems since FOS is eliminated. Desirably and indisputably, the material is made to speak in explicitly clear terms about its capacity, i.e., maximum permissible stress.
6	Applying factors of safety typically results in underdesign or overdesign, rarely achieving precise design. Under-design exposes society to risks such as loss of life and investment from premature or catastrophic failures. Conversely, overdesign increases weight, cost, size, and natural resource consumption unnecessarily.	Since the material is speaking in clear terms and offering maximum allowable stress values with no scope for debates and disagreements, precise designs are by default, and there is no room for overdesign, underdesign, premature failures, catastrophic failures, and their ever-associated consequences on society, product economics, and natural resources.
7	Non-uniform design stress values, due to varying FOS, result in inconsistent product sizes for the same parameters.	Due to the absolute uniformity in the design-stress values, sizes will remain the same for components and products designed across the globe.
8	Due to varying allowable stress values, the globalization of markets became sluggish and problematic.	Absolute uniformity in design stress values could accelerate market globalization. This, in turn, fosters rapid growth in the global economy.
9	Periodic revision of design codes and standards for fine-tuning factors of safety is an ongoing process that consumes considerable time and money. Such revisions also cause concern within the industry, as they are affected whenever design codes are updated.	This problem is eliminated as the factor of safety is eliminated from the design process. Upon adoption, UMR serves as an efficient and reliable factor of safety.
10	Revalidation of existing equipment due to revisions in the FOS is a tedious process that consumes significant time and money. In many cases, it may impose decommissioning of the equipment, thereby resulting in financial loss to the user.	These and several other problems and financial losses are eliminated since FOS is eliminated.
11	No mention about values (T-Y-R-E) obtained in other test directions like transverse and short transverse, i.e., through thickness.	The formulae given in Appendix A are common for test values taken in all the test directions, like longitudinal, transverse, and short transverse.
12	Changes to material parameters such as R, E, Y/T, and E/R do not affect the maximum allowable stress value. This misleads mechanical and materials designers.	Any change, however small it may be, made in any of the six parameters will alter the maximum allowable stress value. This feature is a must for absolute precision in design engineering.

