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ABSTRACT

FORCE AND EFFORT ANALYSIS OF UNFASTENING ACTIONS IN DISASSEMBLY PROCESSES

by
Manuela Sonnenberg

Fastening is the process of connecting one or more parts together with the aid of fastening elements. Unfastening, the reverse of fastening, is the process of separating components from each other by removing or detaching fastening elements. So far, the unfastening process is not well understood, and the analysis about it is not very extensive.

However, the need for disassembly is currently increasing. First, parts have to be taken apart for service and repair, and secondly, for the recycling process. Therefore, there is a need to consider unfastening during the design process in order to enable efficient disassemblies.

The purpose of this dissertation is to develop an analytical model, which enables unfastening analysis during the design of new products. Specifically, (i) a standard nomenclature for defining unfastening related parameters and variables is introduced, (ii) the U-Effort model for deriving the unfastening effort for a variety of commonly used fasteners is developed, (iii) the U-Effort model to model unfastening motion and hence estimate disassembly complexity is extended, and (iv) the U-Force model for estimating the required unfastening force in the case of cantilever and cylindrical snap fits is developed.

The U-Effort model is a detailed study about the unfastening effort and the design attributes of commonly used fasteners. There is a difference between unfastening effort

and unfastening force. Unfastening effort depends on several influencing factors, whereas the unfastening force is a more direct calculated value. The influencing attributes for the unfastening effort include the geometry and shape of the fastener and the condition at the end-of-life of the product.

In the U-Force model, unfastening considerations are included in the design phase, mainly through the calculation of unfastening forces. The U-Force model is applied to the cantilever and cylindrical snap fit integral attachments.

The U-Effort and the U-Force models can be used by designers to evaluate the unfastening suitability of new and existing product designs. Fastening elements can be selected based on functionality and the least unfastening effort. The developed models can assist industrial companies engaged in demanufacturing plan their recycling and reuse activities.

FORCE AND EFFORT ANALYSIS OF UNFASTENING ACTIONS IN DISASSEMBLY PROCESSES

by
Manuela Sonnenberg

A Dissertation
Submitted to the Faculty of
New Jersey Institute of Technology
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Doctor of Philosophy in Mechanical Engineering

Department of Mechanical Engineering

May 2001

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APPROVAL PAGE

FORCE AND EFFORT ANALYSIS OF UNFASTENING ACTIONS IN DISASSEMBLY PROCESSES

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Sodhi, Raj S., Sonnenberg, Manuela, and Das, Sanchoy,
"Use of Snap-Fit Fasteners in the Multi-Life-Cycle Design of Products,"
Proceedings of IEEE International Symposium on Electronics & the Environment, Danvers, MA, May 1999

Juette, Friedrich and Sonnenberg, Manuela,
"Fliesspressen von Mehrfachverzahnungen,"
Umformtechnik, 1/97, S. 26-29, Meisenbach Verlag Bamberg, Germany

This thesis is dedicated to
my family
and
my friends

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LIST OF SYMBOLS

A	beam length
ABS	acrylonitrile-butadiene-styrene
B	beam width
B_{max}	upper margin value of the basic unfastening effort for specific fastener
B_{min}	lower margin value of the basic unfastening effort for specific fastener
C	offset
C_a	accessibility influence factor
C_e	environmental exposure influence factor
C_i (C_1, C_2, ...)	influence factors for different fasteners
C_m	material influence factor
C_t	tool influence factor
d	diameter
d	diameter at the joint
d	penny size (for nails)
d	beam depth
d_i	internal diameter of the hollow shaft
d_o	external diameter of the tube
D	destructive disassembly
D	shelf length
DinY	displacement in y-direction
E	engagement
E	module of elasticity, flexural modulus
E_s	secant modulus
f	unfastening effort index
L	length
L	beam length
L	finger length
MaxDis	maximum displacement
MPS	maximum principal stress
mm	millimeter
N	Newton
R	strength/stress ratio
P	force
P	mating or perpendicular force
P	transverse force
P_{remote}	transverse force in the case the fit is remote
P_{near}	transverse force in the case the fit is near
PC	polycarbonate
PCB	printed circuit board
PEI	polyetherimide
Q	deflection magnification factor
STE	strain energy
t	thickness
t	beam thickness

LIST OF SYMBOLS
(Continued)

t	wall thickness
T	torque
U	unfastening
U-Effort	unfastening effort
U-Force	unfastening force
UV	ultra-violet
VMS	von Mises stress
W	mating force, insertion force, or push-on force
W'	pull-off force, pull-out force, removal force, or retention force
W_a	accessibility weight factor
W_e	environmental exposure weight factor
W_i (W_1, W_2 ..)	fastener corresponding weight value
W_m	material weight factor
W_{near}	mating force in the case the fit is near
W_{remote}	mating force in the case the fit is remote
W_t	tool weight factor
X_N	geometric factor for rigid shaft and elastic tube
X_W	geometric factor for elastic shaft and rigid tube
y	offset, undercut, deflection
y_{max}	maximum deflection
y_{pm}	permissible undercut
α	insertion angle, lead angle, support angle
α'	return angle or retention angle
α'_{crit}	critical angle
$\alpha'_{P-W'}$	retention angle where forces P and W' are equal
β	retention angle
δ_{min}	distance from the end of the tube to the fit
ε	strain
ε_o	maximum strain in the base (allowable dynamic strain limit)
ε_{pm}	permissible strain
μ	coefficient of friction
ν	Poisson's ratio

CHAPTER 1

INTRODUCTION

In this research, a design methodology for unfastening (as one part of disassembly) is developed in order to support the demanufacturing process of products, which aims to reduce the amount of waste going to landfills. This design methodology for unfastening will assist in the design of new products, in the disassembly process of current products, and in the evaluation of fastening methods regarding unfastening and disassembly. Designers have to get a new awareness about design for disassembly and design for unfastening. This research should be helpful in the decision-making process of what kind of fastening methods to use in the multi-life-cycle design of products. Fasteners have to be determined individually for every design, but general guidelines and knowledge about the unfastening behavior and unfastening effort for commonly used fasteners will be beneficial towards an environmentally friendly design.

1.1 Motivation for Research

The motivation for this research comes from the need for a design for unfastening analysis, especially for integral attachments. Recently, there has been an increase in the demanufacturing of products. There is a need of assistance guidelines for disassembly processes in order to reduce costs and time required for assembly and disassembly. Plastic parts are becoming increasingly complex and consequently the disassembly process is getting very complicated. Generally, disassembly or more specifically for this research, unfastening, has many influencing factors, which determine how much effort is needed to unfasten a product. Therefore, it is necessary to understand these factors and their relationship to each other.

1.2 Thesis Objective

The overall objective of this thesis is to increase the knowledge base of unfastening and to raise awareness for the importance of unfastening for designers. There is a great need for more information on unfastening. More specifically, this thesis will:

1. Define unfastening and related parameters.

A standard nomenclature for defining unfastening related parameters and variables are introduced for the first time. It is significant to know how unfastening and related terms are defined in order to be able to consider unfastening in a design concept.

2. Develop a model to obtain unfastening effort values.

The U-Effort model is a detailed study about the unfastening effort and the design attributes of commonly used fasteners. The unfastening effort encompasses all effects that different influencing factors can have on an unfastening process. These influencing attributes for the unfastening effort regarding the geometry and shape of the fasteners are considered in the model in the first part of the study.

3. Analyze disassembly motions.

The U-Effort model is extended to include the effects of unfastening motions and hence estimate disassembly complexity. The unfastening or disassembly motions are set into relationship with influencing factors like material, end-of-life product condition, tools required, and fastener accessibility.

4. Develop a model to calculate removal forces for cantilever and cylindrical snap fits.

The unfastening effort depends on several influencing factors, whereas unfastening force is a more directly calculated value. As an addition to the U-Effort model, the U-Force model covers in detail the effects of unfastening forces. In the U-Force model, unfastening considerations are included in the design phase, mainly through the calculation of unfastening forces. As an example, the U-Force model is applied to the cantilever and cylindrical snap fit integral attachments.

1.3 Problem Statement

The goal of this research is to analyze unfastening and its related parameters and to develop models to estimate the unfastening effort and force for commonly used fasteners and integral attachments.

1.4 Scope of Research

Unfastening is a part of the demanufacturing process, which is itself just one part of multi-lifecycle engineering. Many research fields are joined together in the approach to find solutions for sustainability, protection of the environment, and economical multi-lifecycle engineering. However, this research focuses on the unfastening component of disassembly. The topic of destructive disassembly is only covered marginally. Unfastening can be applied to many fastening elements. There is a great variety of different fasteners. As an example, ten commonly used fastening elements are examined at regarding their unfastening effort. Because of the newness of integral attachments, the emphasis is on their design for ease of disassembly with consideration of unfastening forces for two types of snap fit fasteners.

1.5 Thesis Approach

To understand unfastening, it is important to fully define and analyze the unfastening process. Therefore, a literature review is conducted to estimate the technical foundation and state-of-the-art research regarding unfastening. An extensive analysis of unfastening includes a definition and the study of its influencing parameters. The relationship between these factors and the fastening elements has an impact on the unfastening effort. The model proposed enables an estimate of the relative unfastening effort for different fastener types. Since unfastening is mostly done manually, the effects of disassembly motions on the unfastening effort are added to the model. An additional approach to judge the unfastening process for different fasteners is to look at the removal forces. Therefore, to provide design guidelines for the ease of unfastening the force calculations and the determination of affecting parameters are studied in the last part of this research.

CHAPTER 2

LITERATURE REVIEW

This chapter introduces the research work done in the demanufacturing and disassembly area. It then turns to work done in the related fields of fastening and connections in order to understand the concept of unfastening. Information about unfastening is still very limited, so the author presents some of her own research to deepen the understanding of unfastening. Unfastening is defined and explained, and the scope why and where it is applied is presented.

2.1 Demanufacturing

Demanufacturing is the process of separating parts in a product at the end of its useful life through unfastening and destructive disassembly. Therefore, unfastening is only one part of the demanufacturing process. To completely grasp all aspects of unfastening, demanufacturing is described first. Even though the concept of assembly is well developed and understood, the research related to demanufacture and disassembly is still in a stage of infancy.

Maintenance and service purposes were, for a long time, the only reasons to unfasten a product. However, this has changed in recent years. Motivated by different reasons, more and more people have started to think about the effects of waste disposal on the environment. The recycling of used products is now an issue based on global competitiveness, societal equity, and environmental responsibility [Caudill, 1999]. Different concepts for recycling and pollution prevention have been developed. Recycling has different meanings to different people. For some, it is 'the ability to extend the life of the product' or for others 'the taking of materials from a product and then rendering it to a condition where it can be used again for another product'. It also could be the 'reduction of waste', or 'the reusing of products', and by all this to save energy, reduce costs, and avoid environment damage. Irrespective of the motivation for recycling, in order to be able to recycle, a product has to be demanufactured first.

Demanufacturing usually includes several processes, such as reuse, recycling, disassembly, refurbishment, cleaning, inspection and sorting, part upgrading or part renewal and reassembly, and incineration and/or disposal of products or product parts. Demanufacturing adds a new phase to the product life cycle, see Figure 2.1 [Sonnenberg, Sodhi, 1998].

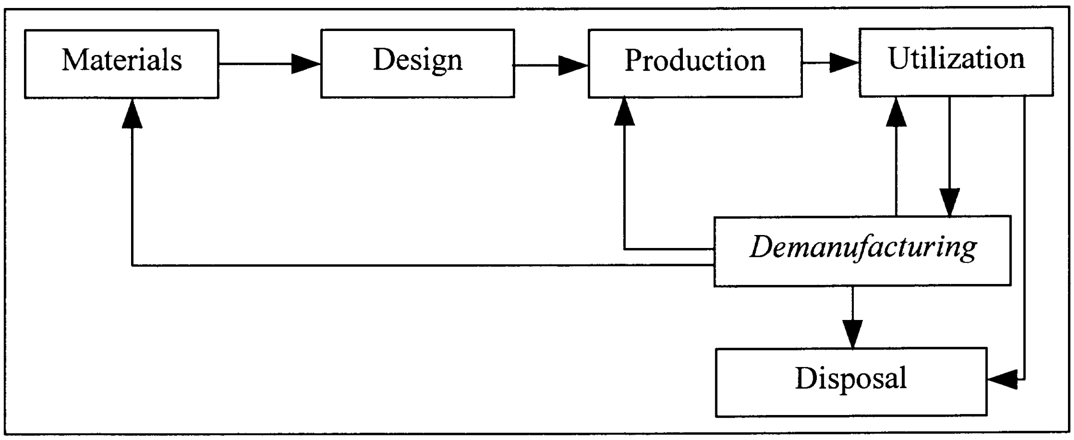


Figure 2.1 Demanufacturing in the Product Life Cycle

The demanufacturing process increasingly reclaims parts and subassemblies from used products. These reclaimed parts and subassemblies can then be used to build new versions of the original product or alternative products. Here, demanufacturing is defined as the process of collecting, dismantling, selling, and reusing the valuable components of end-of-life products. For these end-of-life options of reuse, recycling, or refurbishment and cleaning, it is necessary to disassemble the product. Issues related to demanufacturing are shown in Figure 2.2.

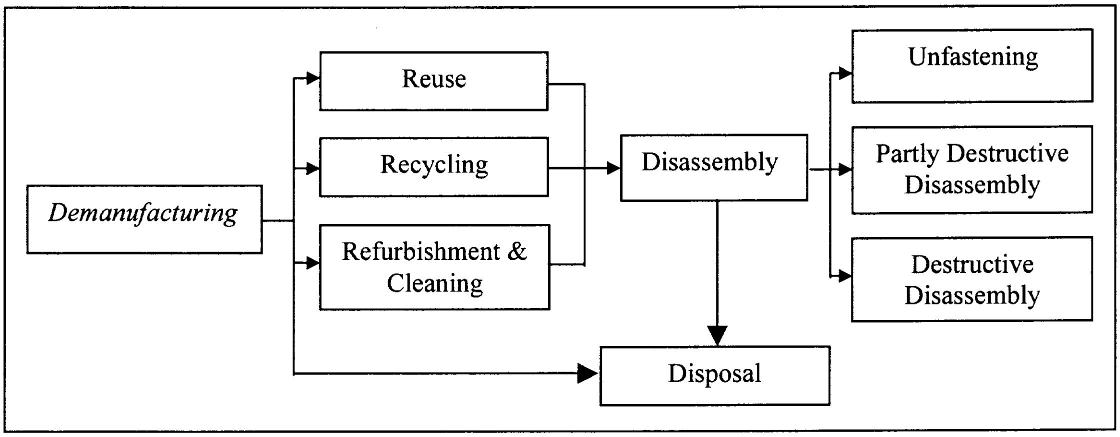


Figure 2.2 Demanufacturing Issues

A number of researchers have worked in the field of demanufacturing with different emphasis like remanufacturing, life-cycle assessment, end-of-life options, recycling, and disassembly. For example, Bras [1998] in his paper "Integrated Product and De- and Remanufacture Process Design," defines demanufacturing as the entire process involved in recycling, reuse, incineration and/or disposal of products after they have been taken back by one or more companies. He suggests that for an environmentally conscious design, de- and remanufacture processes have to be integrated into the design process. An important issue of demanufacturing for the corporate side is stated by Grenchus et al. [1997] from IBM (Endicott, NY) in their paper "Demanufacturing of Information Technology Equipment." They emphasize that for propriety parts, the demanufacturing process has to render the products so that they are unusable (impairment). IBM has also established an environmentally conscious product program in order to incorporate environmental attributes in designing their products. This program encompasses the use of recycled materials, reuse and recyclability of products, and design for disassembly [Brinkley, et al., 1997]. Specifically to the topic of life cycle assessment, Carnegie Mellon University for example developed a program, which has the goal to minimize and effectively manage the use of resources and to minimize toxic releases into the environment [Conway-Schempf, Lave, 1995/96]. Lankey et al. [1997] show a case study where a product life cycle is evaluated using an environmental attributes matrix.

Honeywell/Allied Signal [1996] sees recycling as an important global drive towards reducing contamination, landfill volume and saving of natural resources. Recycled plastic material can often be used for less-demanding applications. Beitz [1993] at the Technical University Berlin, Germany has introduced the Design for Ease of Recycling. He states that in the future a designer has to consider the utilization or the reusing of full products, subassemblies, and parts in connection with a recycling process. Ishii [1997] has developed a methodology to evaluate the modularity of product designs from the recyclability point of view. A recyclability map focuses on disassembly complexity and value recovery efficiency. Langerak [1997] analyzes the question if it is better to shred or to disassemble a product. The presence of (precious) metals traditionally has been the driving force for recycling. Besides that, the recovery of plastic materials can be problematic. Dismantling and plastic recovery has been compared with shredding and physical/mechanical material separation processes. Based on the existing research literature it can be concluded that the designer, in addition to fulfilling the design requirements such as function, safety, ergonomic, operation, manufacturing, and assembly must also consider the design for ease of recycling.

2.2 Disassembly

As a part of the demanufacturing process, disassembly is the process of physically separating parts of a product into its parts or subassembly pieces according to Sodhi and Sonnenberg [1999]. It is defined as the process of removing components from products. It contains all operations for the successful separation of a product. Disassembly is often the preferred technique, because different materials do not get mixed. Unlike the assembly process, which is highly automated and it deals with homogeneous products, disassembly in a demanufacturing facility involves a number of different product types having variable damage and is mostly carried out manually. The overall process of disassembly is still not well understood.

In disassembly, complete components can be recovered for reuse [TUB-Technical University Berlin, 1998]. The disassembly process includes unfastening and cutting, the handling and control tasks, and other special operations. The key aspects of disassembly include part separation through unfastening (non-destructive disassembly) and destructive disassembly, such as cutting or sawing. The fastening method determines if the product can be unfastened or destructively disassembled. There is a third type of disassembly – partly destructive or semi-destructive disassembly. Here, the fastener can be destroyed during the disassembly with no damage to the components. This is often a cost-effective disassembly procedure. Through disassembly, some components can be retrieved for reuse, some parts will be shredded for recycling, and some parts will be disposed. Special attention is needed during the disassembly for handling hazardous and toxic materials.

There have been a number of publications, which give guidelines about disassembly. Beitz [1993] has presented procedures for disassembly of manufacturing structures and for disassembly of joining points. Sonnenberg and Sodhi [1998] present common disassembly tools for various fastening methods and discuss the problems, which can occur.

The determination of the disassembly effort for a given product design is of interest for many people, because it constitutes the economic effort needed to disassemble a product. Recently there have been numerous studies to determine the value of recovered parts and disassembly costs. Most of these methods use a disassembly cost or effort value for each disassembly step. These include the Re-Star method by Navin-Chandra [1993], which provides an assessment of the recyclability and disassembly strategy for any given product design and composition. Mathematical models by Pnueli and Zussman [1997], and by Penev and de Ron [1996] use graphs to prescribe a disassembly plan and to compute the end-of-life value of a product. Gunger and Gupta [1997] propose a disassembly sequence generation heuristic, which aims for an optimum solution. Zussman, et al. [1998] developed a disassembly petri net approach to model and plan disassembly processes. Another example is the multi-factor model to obtain the disassembly effort on a prescribed scale to estimate the disassembly costs by Das et al. [2000]. Dowie and Kelly [1994] have experimentally obtained times for removal of screws and cutting etc. Hanft and Kroll [1996] and Kroll et al. [1996] have developed procedures for estimating the ease of disassembly using work measurement analysis of standard disassembly tasks. Vujosevic et al. [1995] have used work measurement procedures to estimate disassembly times. Some models assume a fixed cost of disassembly per step. These encompass Johnson and Wang [1995], and Gungar and Gupta [1997]. They try to minimize the number of steps needed to retrieve the usable parts. McGlothlin and Kroll [1995] describe a disassembly evaluation scheme that translates form properties of a design into quantitative scores and provide a means of identifying weaknesses in the design and comparing alternatives.

None of these researchers consider the fact that the disassembly process depends upon the type of the fastener, the type of the connection, geometrical shape, size and material of the fastener(s), variability of damage to the fastener, and the arrangement of the fasteners in any assembly. In addition, a large number of products are assembled using integral fasteners, which need to be detached for demanufacture. In the next section, research issues related to the unfastening component of disassembly will be presented.

2.3 Unfastening

In order to understand unfastening, fastening has to be understood first, because unfastening can be considered as a reversed fastening process. That means, only what was fastened or connected before can be unfastened now. Connected parts are all objects with two or more parts mated in a fixed connection or as a flexible joint. The purpose of connected parts is quite versatile. According to VDI Guideline 2232 [1990], connections, or joints are used to connect parts with each other, sometimes also to position them. Further, flexible parts can be joined together and move relative to each other on a certain track. The role of a fastener is very important in assembling parts. A fastener is a component employed between connected parts, which holds the mated parts together and establishes relative part location, alignment and orientation, transfers loads, and absorbs tolerances between the parts to prevent vibrations.

Due to the lack of any suitable definition of unfastening in literature, the author gives the following definition of unfastening. *Unfastening is the process of separating components or subassemblies from each other by removing fasteners or by detaching parts with integral attachments usually manually with or without the use of a tool.* Issues related to unfastening are depicted in Figure 2.3.

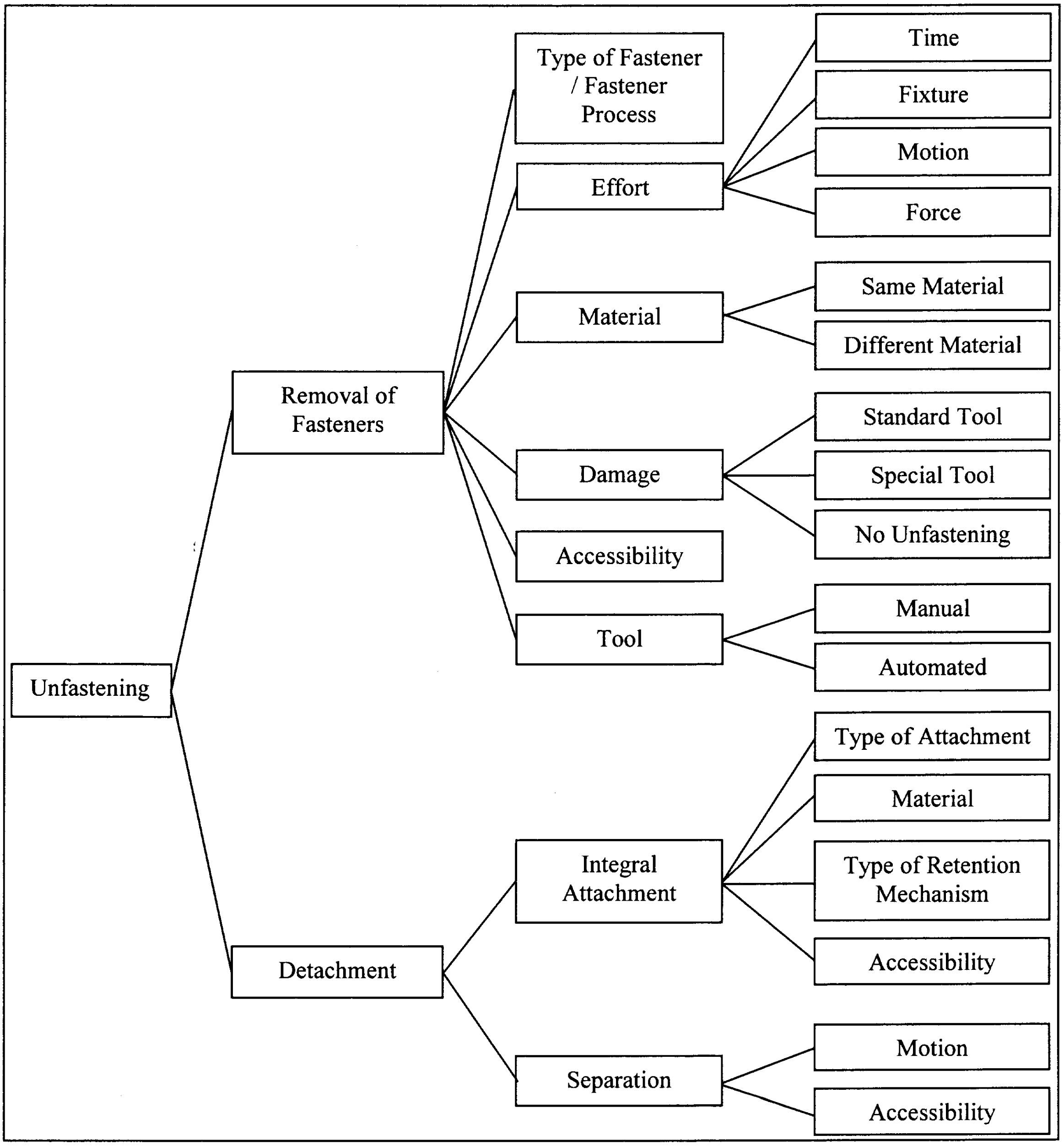


Figure 2.3 Unfastening Issues

Different fastening methods require different disassembly procedures. For example, screws are removable fasteners i.e. for maintenance purposes the screws can be removed through unfastening and later screwed in again. On the other side, there are permanent fastening methods, like welding. In these cases, a product can only be disassembled through destructive disassembly. In this research, however, the focus lies on unfastening processes only.

Now the next question is why to unfasten a product? There are different reasons to remove fasteners from a product. One reason is that a product has to be accessed for maintenance purposes. In this case, fasteners, which can easily be removed, are generally used. Disassembly and reassembly steps are performed to carry out servicing, maintenance, and upgrading tasks. Nowadays demanufacturing, that means disassembly for reuse and recycling has become an important reason for unfastening. The prime performance measures of any installed fastener or fastening system are strength, appearance, and reusability.

As seen above in Figure 2.3, unfastening can be done by two basic methods. One is the removal of discrete fasteners and the second is through detachment of components with built-in fastening elements. Considering the removal of discrete fasteners, issues such as the fastener type, fastener process, unfastening effort, component and fastener material, fastener damage during use, fastener accessibility, and unfastening tools are of significance. Literature on unfastening research is very limited. Even fastening is linked to design and assembly issues. Unfastening is grouped together with disassembly and demanufacturing, but there have been no in-depth studies done on unfastening processes. Chido, et al. [1999] uses shape memory polymers for active disassembly, but this method can only be used with certain material and specific applications. Then, Shu and Flowers [1995] put fastening methods in relation to remanufacture. In their paper, "Considering Remanufacture and other End-of-Life Options in Selection of Fastening and Joining Methods" they look at the effects of fastening and joining methods on remanufacturing. Remanufacturing here includes disassembly, sorting, cleaning, refurbishment, reassembly, and testing with the main goal of part reuse. However, one result stated is that 'design for remanufacturing' not necessarily means the same as 'design for recycling'.

Regarding the issue of type of fasteners, Scharff [1979] has described different fastening elements in 'Successful Putting It All Together'. Furthermore, Keeley [1974] deals almost entirely with a wide variety of non-threaded fasteners in 'Miscellaneous Fasteners', an engineering design guide. Speck [1997] looks into fastening properties in his book 'Mechanical Fastening, Joining and Assembly'. He also touches on the topic of reusability and its influencing factors. However, he looks at reusability as an aspect of reassembly, and not necessarily of demanufacturing and recycling. Furthermore, Lee and Hahn [1996] give some kind of fastener classification in their paper "A Survey of Integral Fit Joint Technologies for Composites". They identify new technologies for joining structural components and their potential uses and evaluate them in comparison to other joining methods. Similarly, Messler [1993] in his book 'Joining of Advanced Materials' classifies fastening methods. He has described advantages and disadvantages of mechanical fastening, integral attachments, and adhesive bonding. He also mentions disassembly, but for service and maintenance only. Furthermore, Ananthasuresh and Kota [1995] analyze compliant fastening methods, where elastic deformation generates a desired motion and force for applications in micromechanical systems. Considering the fact that disassembly has become an important industrial activity, the determination of the unfastening effort is vital in making disassembly economically efficient. As there is no research literature available about the unfastening effort, its determination and relationship with various factors are an important objective of this research.

Material properties of fasteners and components are another important issue of unfastening. Lately plastic materials have been used frequently for fastening elements (both integral and discrete fasteners), which need more considerations in design than commonly used metal fasteners. Hoechst Celanese [1991] provides a manual "Designing with Plastics – The Fundamentals", which gives information about the properties of plastic materials to assist designers in the use of plastic fastening elements in product design. Plastic fasteners in general are also discussed by Schuch [1989] in the article "Plastic Fasteners, Please." Schuch claims that the plastic materials are attractive, because they offer versatility and design advantages. They enable multifunctional components with a variety of shape, sizes, and finished conditions, which often combine two or three parts into one. In many cases, they can replace traditional fasteners. Plastic materials have high lubricity and moderately high temperature resistance. However, Schuch also emphasizes that not every fastening need can be met by plastics.

Fasteners are often damaged due to environmental exposure. Damage makes unfastening very difficult or even impossible. A listing of the fastening factors, which have an effect on the ability to reuse a specific style or type of fastening, would be difficult given the wide range of fasteners available [Speck, 1997]. According to Speck, some examples of damage to fasteners include the damage to a drive during installation, seizing of the threads from friction, damage from vibration, metallurgical fatigue of metals, ultraviolet degrading of plastics, thread wear, and service misplacement of fasteners. For many mechanical fasteners, the main problem is corrosion. Messler [1993] shows different types of corrosion and their effects on the connections. Further, he suggests ways to prevent corrosion. Deutschman et al. [1975] also study the effects of corrosion and methods to control it. The major problem in unfastening of damaged fasteners is the inability to use standard unfastening tools, e.g. the screw heads may have been damaged. Considering the various types of fasteners, materials, geometry and the amount of damage, it is almost impossible to quantify relationships for the analytical determination of the unfastening effort.

About tools and accessibility, these issues usually are closely related to each other and they depend on the type of fastening element. In his guidelines for design for ease of recycling, Beitz [1993] gives some examples of good and bad accessibility and tool use. Das et al. [2000] include in their disassembly effort model a score for how difficult a part can be accessed and what kind of tool has to be used.

The second method of unfastening is detaching. One aspect of this method is the separation of components. Once the fastening element is removed, several parts can be taken out of the disassembly. The separation then means, for example, the sliding of a part out of a slot, or the lifting of a component. The other use of detaching is for the unfastening of integral attachments. The most commonly used plastic fastening elements are integral attachments, which are becoming increasingly popular, as they reduce the number of parts inventory. That usually reduces the assembly time and costs. The use of integral attachments is growing rapidly and because of the newness of their design, special emphasis was placed on them in this research. However, often the design of integral attachments is still regarded more as a form of art than science. Until recently, not much information on integral attachments has been available. Integral attachments are not yet part of the discussion in machine design texts. This has started to change. Rensselaer Polytechnic Institute (RPI) in Troy, New York has an Integral Fastening Program under the direction of Dr. Gary A. Gabriele. In the "Integral Fastening Program Mission Statement" [1995] it is stated that their goal is to obtain design guidelines for improving the performance of snap-fit type fasteners. In addition, they have developed design guidelines for standardized integral attachment features, that can be readily and economically used in product design and advanced fastening concepts and evaluation methods. Mayer and Gabriele [1995] in their paper "A Design Tool Based on Integral Attachment Strategy Case Studies" present design guides or attachment strategies for integral attachments implemented as a software tool.

Similarly, Ohio State University has an Integral Attachment Program with Dr. Anthony F. Luscher as principal investigator. He did extensive research in the field of integral attachments, and especially on cantilever snap fits. In his Ph.D. thesis [Luscher, 1995] he investigates the performance of cantilever hook-type integral attachment features. A finite element model of the actual insertion and retention processes of hooks using contact and friction surface elements was developed. The results were compared to experimental data. In addition, he has developed equations for determination of insertion and retention forces for these types of fasteners.

Furthermore, Knapp et al. [1995] investigated the performance of in-plane cantilever hooks. He experimentally determined the effects on performance variations in the geometry of the in-plane cantilever hook and compared the results with a nonlinear finite element analysis procedure. Lewis [1996] has examined the compressive hook in his research using results of experiments and finite element methods to generate approximate second-order response surfaces. These are used to calculate the insertion and retention forces for the compressive hook integral attachment feature. Furthermore, Lewis et al. [1997] have also studied the bayonet-and-finger type integral attachments. A method similar to the one for the compressive hook has been applied here. The results are incorporated into feature design guidelines.

On the corporate side, Honeywell-Allied Signal developed a 'Snap-Fit Design Manual' [1998]. The manual has the purpose to assist in the basic snap fit design, and to help calculate the strength of the component and the amount of force needed for assembly. They introduce a deflection magnification factor, which reflects the length/thickness ratio of the cantilever beam and also the beam configuration. Furthermore, Hoechst [1991] has published a manual where snap fit design is discussed. Uniform and tapered cantilever snap fits are distinguished and a proportionality constant for the tapered beam is introduced. DuPont [1990] gives design considerations for cylindrical and cantilever snap fits in their technical report "Snap and Press-fits in Engineering Polymers." In addition, Bonenberger [1995] from GM in cooperation with RPI has defined assembly motions for integral attachments in his paper "A new design methodology for integral attachments." At the University of Erlangen-Nuremberg [1998], design guidelines for cantilever snap fits have been developed for assembly. Furthermore, they have introduced the term of disassembly suitability. Also the Technical University of Munich [Dobmeier, Pscheidt, 1997], in their research in the field of integral attachments, has developed design considerations for snap fits for the ease of assembly.

In most of the research on integral attachments and snap fits, the focus lies usually on the cantilever snap fit design. Information about cylindrical snap fits is limited. At the Rensselaer Polytechnic Institute in Troy, NY [1995], some work has been done on the determination of the forces for the cylindrical or annular snap fit fasteners. Similar equations are also provided by the University of Erlangen-Nuremberg [1998].

Polyplastics in Tokyo, Japan [1998] provide design techniques for press fits and snap fits. They compare cylindrical snap fits with press fits made from plastic material. Additionally, arc-shaped cantilever snap fits are also studied here. An application example of a cylindrical snap fit is given in the paper of Bowman and Pawlek [1993]. Rapid prototyping technologies have been used here to evaluate design approaches, and a Taguchi screening study has been used to determine the hierarchy of design parameters governing the required removal force.

Product designs using integral attachments are usually an iterative process due to their complexity. Several calculations may be needed before optimal parameters can be found. To simplify this process, some software solutions are available for the design of snap fit features. For example, 'Winsnap' from Rapra Technology [1997] provides a pre-design program for snap-fit elements. It is a computer tool for the design of snap-fit elements used in the assembly of plastic parts. 'Snap Design Software' from Closed Loop Solutions [1997] is another software for snap-fit attachment design and analysis. Eastman [1995] provides a 'Cantilever Snap-Fit Design Analysis – Snap-Fit Calculator' based on the snap length and the deflection. Then, RPI's research resulted in the 'IFP Snap-Fit

Design Tools' [Gabriele, et al., 1995, Oh, et al., 1999]. It is a Java-based design calculator for integral snap fits. This tool aids in designing snap fits to meet specific loading requirements. The procedure is useful in the design process and the most convenient way to estimate the performance of snap fits.

Summarizing it can be said that there is very limited information available on the unfastening process, although the issue of fastening and fastening methods of individual fasteners is a well-covered topic. In addition, there is extensive research material available about integral attachments, but the emphasis on their detachment or disassembly of parts with integral attachments is not there. Fasteners or fastening methods have been mentioned in many papers or books in one way or another, but usually only in relationship with design/assembly issues. Unfastening is seldom covered. In view of the existing literature, there is a need for a better understanding of the disassembly and unfastening processes and for a procedure to obtain the disassembly effort associated with the unfastening component. Therefore, this thesis is attempting to establish a fundamental knowledge of unfastening and for the first time to characterize the unfastening effort for commonly used fasteners. One objective of this research is to present a new multi-factor model to estimate accurately the unfastening effort for commonly used fasteners. In addition, for the cylindrical and cantilever snap fit type of integral attachments, removal forces and the parameters, which influence them, will be studied. The results will be included in an overall model for the design of integral attachments. Now concentrating solely on unfastening, influencing factors for the unfastening process will be studied in the next chapter.

CHAPTER 3

PARAMETERS FOR THE UNFASTENING PROCESS

The effort to unfasten components of an assembly affects the disassembly cost and therefore is an important issue in the product design for disassembly. Unfastening can be done in different ways and these influence the unfastening effort. The unfastening effort is different for different fastener types and depends on various factors. This study has evaluated most of these factors and concluded that the relevant factors are the six factors shown in Figure 3.1. These are studied separately in order to evaluate their impact on the unfastening process.

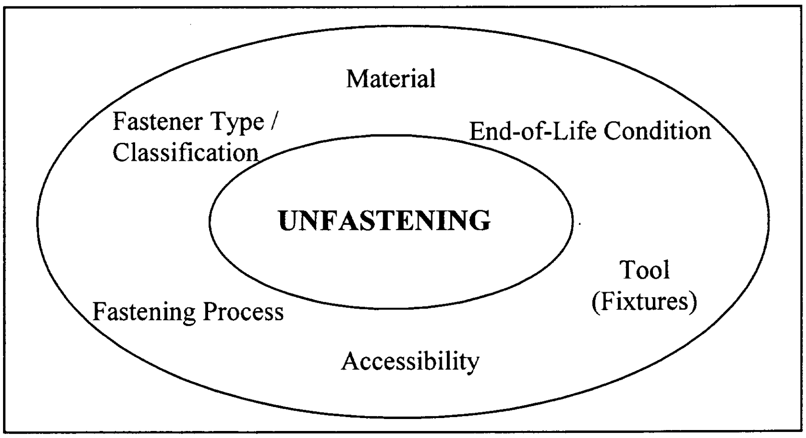


Figure 3.1 Unfastening Influence Factors

Because so many factors affect the unfastening effort, a designer has to consider these to obtain a suitable design. A design is suitable, if the product easily can be unfastened. Unfastening suitability is then determined by the design structure and the type of connection. That includes the unfastening influence factors shown in Figure 3.1. These factors are looked at first.

3.1 Fastener Type - Classification

The field of fastening methods is very wide [Scharff, 1979, Das et al., 1997]. There are many kinds of different fasteners or attachments or bonding. Each fastening method must perform its intended function adequately to specific conditions. Here, a classification of fastening methods is introduced as shown in Figure 3.2.

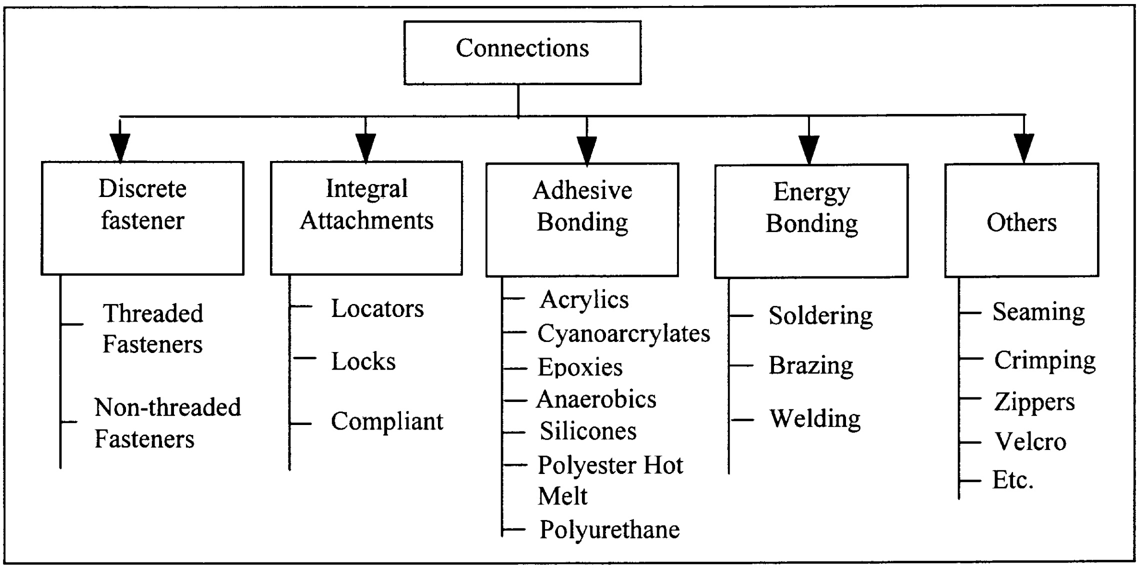


Figure 3.2 General Classification of Fastening Methods

The group of discrete fasteners can be divided into two groups, threaded fasteners or non-threaded fasteners. Discrete fasteners are separate fasteners, which connect two or more parts with each other. They are independent of the parts and can be removed. Sometimes they are also called mechanical fasteners [Messler, 1993]. Generally, mechanical fastening allows simple and practical disassembly without any damage to the components. Further, they also permit relative motion between parts while providing mechanical alignment, which can be very important in certain applications. Discrete fasteners cause no change to the chemical composition, and they give the opportunity to join dissimilar materials together. On the other side, they can create stress concentrations at the point of fastening (hole in part). Another disadvantage is that joints can loosen through vibrations, thermal changes, or fastener relaxation.

Integral attachments are fastening units integrated into the parts. They are belonging to the parts so that an assembly without separate fastener is possible and often during assembly, a multiple joining takes place. Integral attachment is advantageous because the number of parts can be reduced, the assembly time decreases, and less tools are required for the assembly. Here, three groups can be distinguished, locators, locks, and compliant.

In the general field of fasteners, adhesives are becoming more popular. They have been used for many years in various areas of manufacturing. Originally, glues and cements were used for bonding purposes where little strength was required. The ever increasing number of new adhesive compounds, together with all the variations in each basic adhesive material, make the selection of a suitable formulation for a given application seem very difficult.

Energy bonding is a method where the joint is melted or plasticized in order to form a bond using an external energy source such as ultrasound or inductive heating. Soldering, brazing, and welding are all popular processes. Soldering is a joining operation that may be good for electrical connections, or it can be used for sealing out fluids under low pressures. Brazing is a process that is somewhat more complicated than soldering; it does a better sealing job, but the joint will not withstand much load. The class of welding processes would include types as forge, gas, Thermit, induction, resistance, and arc welding.

A great variety of different fastening methods can be used as connections. Some connections do not fit into one of the classified groups, so they have been classified as a group of others. This is true, because for many applications special purpose fasteners are designed and used. That concludes the information given here about fastener types. The next section considers the influence material properties have on the unfastening process.

3.2 Material Properties of Mating Components or Fasteners

When designing a fastening connection, fundamental knowledge about material properties is essential. It is essential for the proper functionality to understand the properties in different environmental situations, and it is important to understand the different kinds of properties and their influence on a product design. Material properties provide quantitative information about the response of a material to external triggers, which can be mechanical, thermal, electrical, optical, chemical, etc., or combinations of these. However, not all properties have to be considered in every case. It is dependent on the environment in which a certain fastener will be used. In the design process, following topics should be considered in the selection of material according to Davis, et al. [1982]:

- Types of material available
- Properties of various materials
- Service requirements for materials
- Relative economical value of various materials and of various forms of a particular material (different grades)
- Methods of preparation or manufacturing of various materials or products and the influence of processing on their properties (e.g. injection molding, blow molding)
- Methods of testing and inspection and their significance with respect to the measure of its desired properties.

In order to select the most suitable material, the designer should consider the usage of the desired product and try to clarify static and dynamic forces, which are going to be applied to the component. The simplest shape and form is usually preferred. Furthermore, the magnitude of stresses and strains in the part resulting from the forces applied should be estimated and the approximate part shape and dimension then be chosen. The estimated stresses and strains are significant, because they can be compared with material data to help find the right material [Brown, 1980].

Material properties can change when subjected to excessive loads and/or certain environmental conditions over long periods. Metals usually behave conveniently according to the equations of stress analysis in a linear way, however, problems like corrosion can occur. For plastic materials, creep and stress relaxation have to be considered so that they do not effect the functionality of the product. Even at room temperature, creep can occur, which means the plastic part will gradually change under load. Generally, many different properties can influence the material selection. It is necessary to understand these in order to obtain the performance and reliability needed in a part or product. In addition, due to the increasing importance of recycling issues, it is essential to have knowledge about the used materials in a product. This includes in addition to properties for processing and usage end-of-life properties as well.

In the disassembly or unfastening process it should be clear what will be done with every component. Some parts might be useful for reuse, but a major portion will go to recycling or disposal. Knowledge about materials can help to decide what to do during the demanufacturing process. For example, a plastic or metal can be shredded and then added to the process of the same product or the recycled material can be rendered to another product and used for other purposes. Material data determine how large the ratio between virgin and recycled material can be. A major concern is what will happen at the end-of-life, what kind of impact will the material have on the environment if it would be disposed (landfill), incinerated, or recycled. In some cases, it could be economical to have parts made from the same material. For recycling they do not have to be disassembled, which saves time and costs. Companies have to face these considerations regarding recycling and demanufacturing, because more and more consumers demand it. Therefore, it is important to select suitable materials for the ease of recycling and disassembly. Further, it is easier to separate fewer different materials than a large variety. The material selection should be minimized. In any case, materials, which can be recycled, should be used preferably. Another issue is that toxic or hazardous materials require special attention in the demanufacturing process. The product should be designed in a way that provides easy access to these hazardous parts so that they can be removed without difficulty.

Fundamental mechanical properties are strength, stiffness, elasticity, plasticity, and energy capacity. Mechanical properties are crucial since virtually all end-use applications involve some degree of mechanical loading. Material selection for a variety of applications is often based on mechanical properties such as tensile strength, modulus, elongation, and impact strength. These values are normally available in the marketing data sheets provided by material suppliers. In practice, materials are rarely subjected to a single, steady deformation without the presence of other adverse factors such as the environment and temperature. Furthermore, friction is an important factor for unfastening or removal forces. Stresses, strains, and elasticity have effects on the unfastening process.

Parts made of flexible materials have to be unfastened in a different way than brittle parts. Products under constant load during their usage stage often show effects of fatigue, which also can influence the ease or difficulty of the unfastening process. The influence caused by the end-of-life conditions is closely related to the material properties.

3.3 End-of-Life Condition

In-use properties or end-of-life conditions are here regarded as factors, which are related to changes through environmental exposure. Fastening methods are used everywhere, therefore, fasteners can be found in all kinds of conditions. The result can be that the access might decrease and a possible performance loss of the fastener occurs with time [Speck, 1997, Deutschman, Michels, Wilson, 1975]. Effects of environmental exposure during usage can be corrosion, wear, debris buildup in drives, temperature caused deformations, vibrations, or UV degradation. During use, the subassembly and/or the fasteners may have been damaged. For a damaged fastener, unfastening may not be feasible and only destructive disassembly is possible.

The most common environmental influence is corrosion. Corrosion can make the unfastening process very difficult, because the strength properties of the material will change and ultimately it can cause a reduction of the cross-section or lead to complete failure of the fastening joint. There are different types of corrosion [Messler, 1993]. Corrosion can take place uniformly over a surface, or it can occur localized. In most cases related to metal fasteners, the problem is oxidation or rust. The oxidation rate increases with rising temperature. Other types of corrosion are galvanic corrosion, stress corrosion, and pitting corrosion. Galvanic corrosion occurs when a combination of two dissimilar metals is used together in a fastening joint with an electrolyte. The electrolyte can be acid rain, ocean salt spray, or even rain, dew, snow, or high humidity. Stress corrosion occurs when cracks appear and then propagate under the stress in a corrosive environment. Pitting or concentration-cell corrosion can occur in some metals, when they are exposed to certain corrosive agents (electrochemically corrosion). To prevent corrosion through oxidation, stress, or pitting corrosion, it is important to use protective coatings or finishes. Wear is damage to a surface caused by the effect of one or more surfaces moving past each other while in contact. Different types of wear can be distinguished. One type is galling; it is also called scuffing, scoring, and seizing. Then there is abrasion, pitting and fretting, and cavitation erosion. Wear and corrosion are very close, and often both effect a product.

The problem in unfastening is that it may be difficult to remove a damaged fastener with the use of standard tools. For example, a corroded screw can cause major problems to be removed with a standard screwdriver, because the drive might be damaged. Any fastener, which has a drive can have grit, grease or lubricants build up in it. This might cause a reduction of the torque transmitting ability. To minimize the buildup of debris in fastener drives, fasteners are to be located in such a way that they are shielded from direct discharge and accumulation of debris materials.

Large temperature deviations have a marked affect on the fatigue strength of metals. Elevated temperature can cause problems in threaded fasteners. Thread seizing can occur, especially in fastening connections with dissimilar materials with different thermal expansion coefficients. Jamming might then prevent nondestructive unfastening. Therefore, it is important to have enough clearance and to use temperature appropriate lubricants in these applications. Another factor can be vibration. Fasteners can be loosened, which might cause reusability problems ranging from the hammering and fatigue of parts, to fasteners which shake out completely and are either lost or fall into working components with the possibility of damage. For plastics, UV degradation is a very important factor to consider. If a product is exposed to sunlight for a longer time, some material properties might change. It is important to know the UV resistance of the used plastic material and if necessary to apply appropriate shielding. Also, organic compounds can be severely damaged and even destroyed by a little exposure to radiation. At the end of its useful life, a product can show signs of exposure to one or even more than one of the mentioned factors. The selection of the disassembly tool is dependent on the condition of the fastening element.

3.4 Tools

Generally, a look to the assembly techniques and procedures can be very helpful in the unfastening process. Tool access, grasping and fixturing is usually very similar for assembly and disassembly, just the motions are reversed. Under the aspect of accessibility, it is necessary to consider that tools need a certain access field. Some tools require a precise position in a certain orientation to do the task. For example, a higher accuracy is needed to fit a screwdriver blade in a screw head than a simple gripping and removing movement. In some applications, special assembly tools and fixtures are used, therefore, for the disassembly the same special tools might be needed. It is especially economically disadvantageous if OEM (original equipment manufacturer) tools are needed in the unfastening process. For example, just for unscrewing, several different tools can be used, like a Phillips screwdriver, a flathead screwdriver, a nut driver, a fixed-end wrench, an adjustable wrench, a socket with ratchet, an Allen key or a power wrench. There are also different gripping or fixturing tools, like a vise, pliers, standard grippers, long-nose grippers, expanding grippers, or large grippers. In addition, for the disassembly pry bars, hammers, chisels, wire cutters, drills, or special tools can be necessary. The tools depend on the fastening method and the condition of the fastener.

3.5 Accessibility

Accessibility is the ease or difficulty, with which a fastener or part can be accessed, that means the positioning of the tool on the fastener interface, and then the use of an unlocking motion. However, accessibility of a fastener might be something, which is often overlooked in the design process. Accessibility depends on different factors e.g. how the fastener is designed, where the fasteners are located, and the type of fasteners used. It also depends on if any corrosion protection was used and the value of the proper approach angle for access to assemble or disassemble fasteners. Another aspect of the accessibility is that access can decrease with time and service environment exposure. This can happen due to corrosion effects on fastener drives and mating fastening clamping surfaces such as threads and heads. The disassembly motion is dependent on the accessibility. Since most unfastening is done manually, the issue of disassembly motion is also an important factor. More information about disassembly motions will be given in Chapter 5.

Fastener accessibility can be from one to five directions. Usually, access from the z-axis, which means directly from above, is preferred. This is ergonomically the best case for manual disassembly. Under the aspect of accessibility, it is necessary to consider that different tools need a certain access field. Beitz [1993] shows in "Design for Ease of Recycling" an example of an engineer's wrench or open-ended spanner, see Figure 3.3.

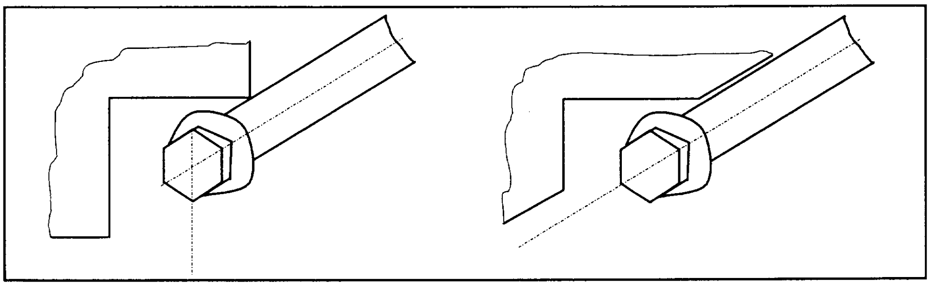


Figure 3.3 Design for Good Accessibility for Disassembly Tools

3.6 Fastening Process

A product generally consists of several subassemblies or components, which have been assembled together. In order to proceed with the disassembly, it is important to determine the type and number of fasteners holding the particular subassembly to the product. The more fasteners were used, the longer is the disassembly time and the higher the disassembly cost. The location of the fasteners plays an important role in determining the unfastening effort. Another issue to consider is if the components were assembled in a proper manner, for example, was a screw tightened with the appropriate torque, or was it over-tightened and plastic deformation took place? The decisions made in the design and production phases have a major effect on the unfastening process.

3.7 Unfastening Suitability

Until recently, designers almost never thought about the ability to disassemble a product at the end-of-life of the product. As discussed, many factors influence the unfastening process. If these factors have not been considered in the design, disassembly can be quite difficult. A good design has to combine ease for assembly and disassembly and still provide a good functionality. D. E. Lee and H. T. Hahn [1996] developed a classification scheme. Table 3.1 shows an extended classification, which also includes the unfastening aspect, besides the general factors to consider for the four basic fastening groups Assembly and functionality aspects are usually already integrated in the design process. Therefore, to facilitate unfastening it is important to include disassembly or unfastening suitability in a product design.

The KTmfk center of the University of Erlangen-Nuremberg, Germany [1998] did some research in the field of assembly and disassembly suitability. Some criteria for unfastening suitability are presented in Figure 3.4. Two components influence the unfastening suitability: the design structure and connections/joints. Every product is designed with a certain purpose. To fulfill a specific functionality is the main goal in the design process. Depending on the design structure and the used fastening elements, a product can be easy to disassemble or unfasten (suitable) or it can be difficult (not suitable). In the next section, the different factors for the unfastening suitability will be looked at.

Table 3.1 Classification of Fastening Elements under the Aspect of Fastening, Functionality, and Unfastening

	Discrete Fastener	*Integral Attachments*	*Adhesive Bonding*	*Energy Bonding*
Fastening:				
Design issues	localized stress concentration, fastener spacing, fastener weight, shear and tension joint loads	elastic assembly deformation, local friction forces, macroscale material interference	large joint areas, uniform stress distribution, damping of shock loads	thermal and electrical conductivity in bond area
# of fastening elements	one to multiple	one to multiple (but integral)	none to one	none
Assembly motion	twist (threaded), push, pull, impact, slide (non-threaded)	push, slide, tip	chemical	external energy
Assembly steps	few to many (depending on fastener)	few	few to many	few
Necessary tools	standard/special (depending on fastener)	usually no tools (simple tools if necessary)	simple tools necessary in most cases	torch
Breakage/damage during assembly	low	high to low (depending on α')	medium	low
Cost for changes	medium	high	low	low
Prototype testing	good	difficult, expensive	often expensive	good
Automated assembly	possible (depending on fastener)	possible	possible	possible
Functionality:				
Joint Efficiency	high (R=1)	tailorable (R>1)	low (R<1)	low (R<1)
Unfastening:				
Disassembly	non-destructive & semi-destructive	non-destructive or destructive (depends on angle)	destructive	destructive
Disassembly motion	twist (threaded), pull, pry, lift, ... (non-threaded)	pull, lift, pry, push, slide,...	destructive	destructive
Disassembly tool	standard: pliers, screwdriver, ...	standard: pliers, screwdriver, ...	cutters, shears, saws, ...	cutters, shears, saws, ...
Possible problems	corrosion, accessibility, damage	accessibility, multiple latches joining	permanent	permanent
# of estimated reassemblies	none to several times	none to several times (depends on retention system)	none	none

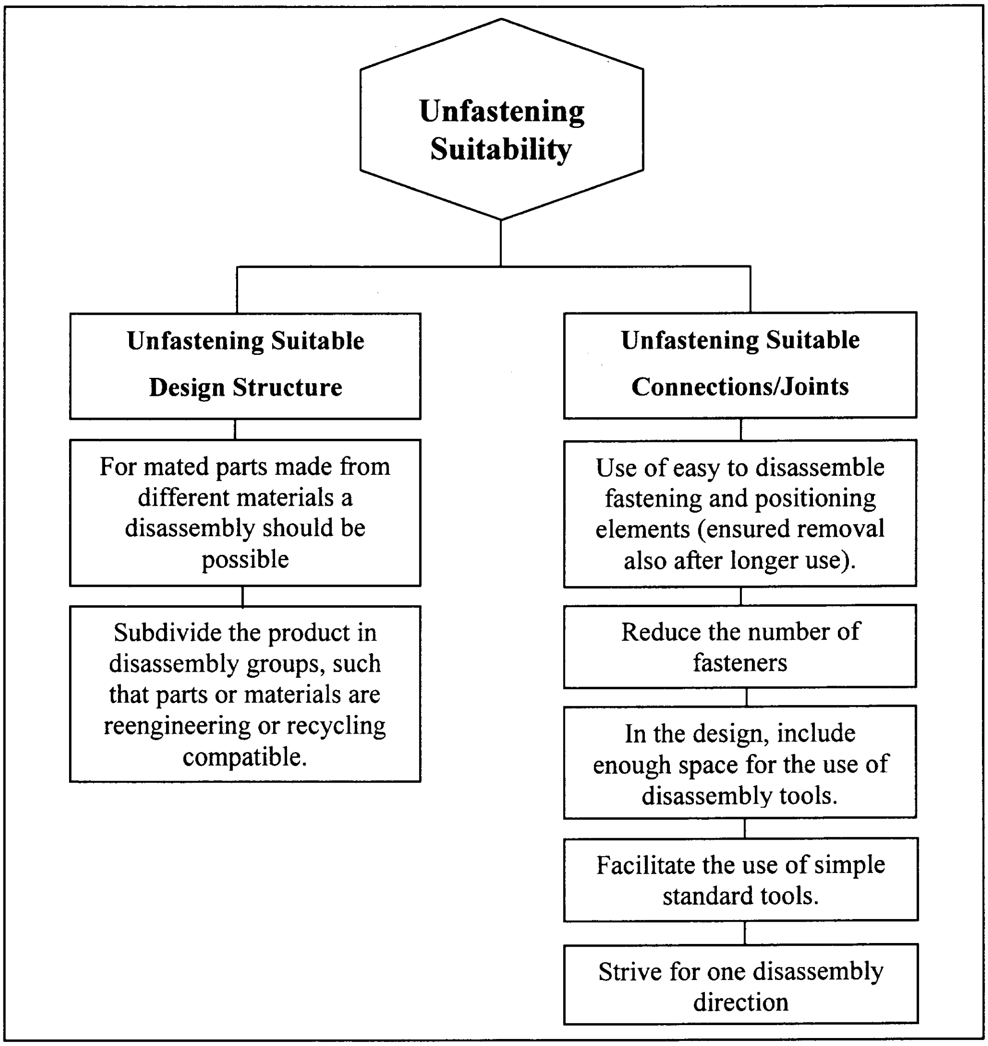


Figure 3.4 Disassembly Suitability

3.7.1 Unfastening Suitable Design Structure

♦ It has to be possible to disassemble mating parts made out of different materials.

If it is not possible to use a single material, then at least fastening elements, which enable unfastening, should be used in the design, so that for recycling reasons the materials can be separated. Hereby, it is important to consider rigidity and weight (usually manual disassembly), corrosion resistance (damaged fastening elements are more difficult to unfasten), etc. of each component to obtain a good and suitable design.

- Subdivide the product in disassembly groups, such that parts or materials are reengineering or recycling compatible.

A clear structure of the product simplifies the disassembly planning. That means manufacturing structures influence the disassembly. Schmidt-Kretschmer and Beitz, [1991], state that it is better to have a tree structure than a centralized structure, as shown in Figure 3.5. In a tree structure, it is easier to identify the material of each component, and therefore, to plan how many disassembly/recycling bins are necessary.

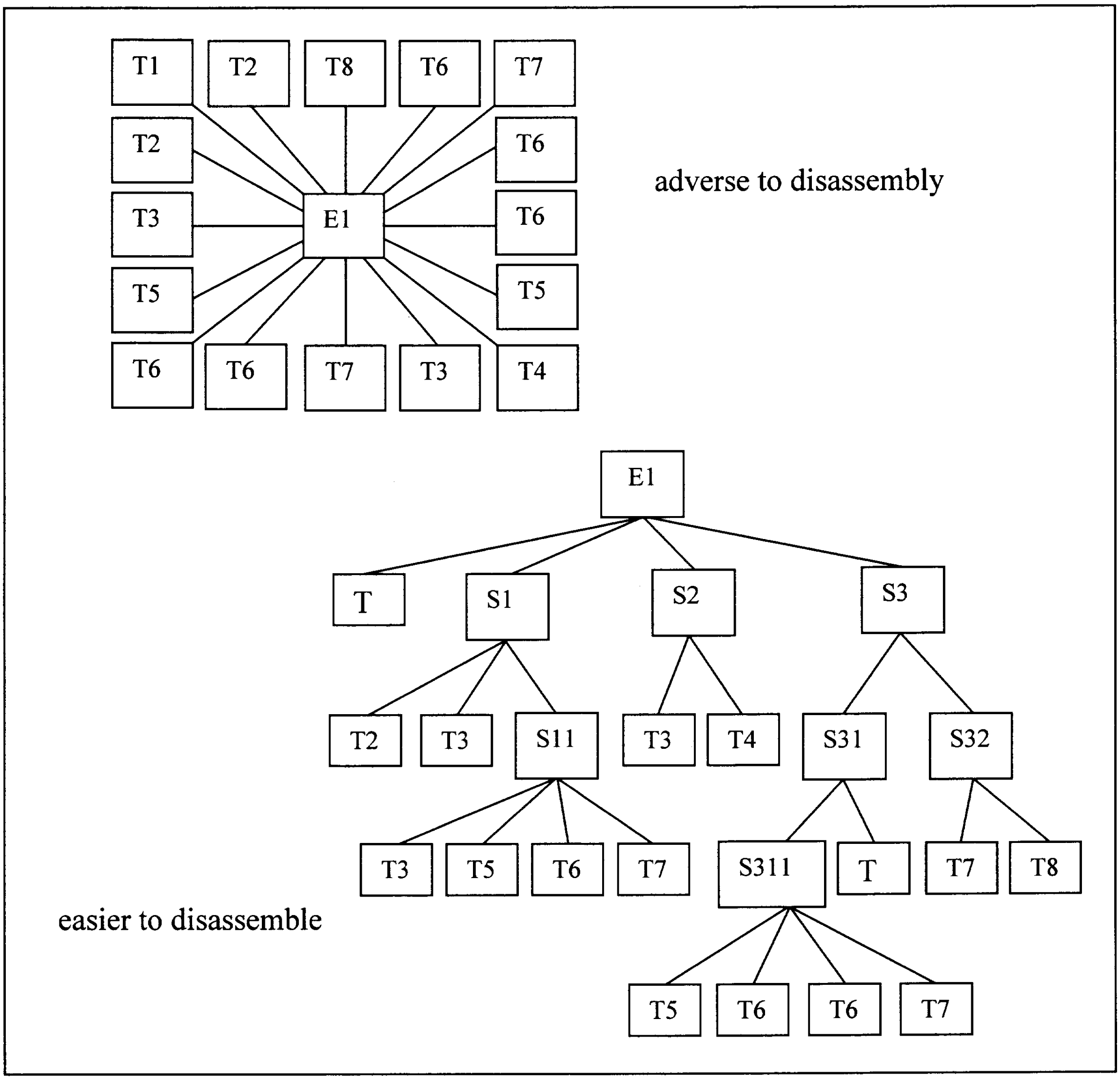


Figure 3.5 Disassembly Structures

3.7.2 Unfastening Suitable Connections/Joints

- Use of easy to disassemble fastening and positioning elements (ensured removal also after longer use)

Not all fastening elements are suitable to be unfastened or disassembled. Therefore, it is important to use fastening elements, which can be unfastened, in order to accommodate unfastening suitability in the design. Later in this research, different fastening elements are analyzed in more detail. The required unfastening effort is estimated for them depending on the different influencing factors.

- Reduce the number of fasteners

Through the aggregation of separate connections and replacing them with one, the number of fasteners can be reduced. Fewer connections mean less disassembly steps. Therefore, the disassembly time can be minimized. Another issue is the use of fewer different connections. If, for example, two different kind of screws in a product can be replaced by one screw size, only one disassembly tool has to be used, and no tool change is necessary. Preferably, common sizes or dimensions of fasteners should be use.

- In the design, include enough space for the use of disassembly tools

Accessibility for the use of disassembly tools is an important aspect; see section 3.5 for details.

- Facilitate the use of simple standard tools

To limit the cost for the disassembly, it is recommended to use standard tools. They have the advantage of a wide range of use and the acquisition cost is low compared to OEM or special tools, (see section 3.4).

- Strive for one disassembly direction

When a uniform disassembly direction can be implemented, the process is easier to accomplish and the disassembly time and costs will be lower.

It is important for a designer to decide how to implement these criteria in a design and how to choose the right type of fastening element for each application. Therefore, in the next chapter, the unfastening effort will be looked at and a model will be proposed, which will enable a relative comparison of different fasteners.

CHAPTER 4

THE U-EFFORT MODEL

The U-Effort or unfastening effort is depending on different factors, as was shown in Chapter 3. The unfastening effort is the total effort required to execute the action of removal of a fastener. In the next step, the unfastening effort for different commonly used fasteners is determined and a U-Effort model is developed.

4.1 Introduction to the U-Effort Model

Product designers have traditionally only been concerned with optimizing the functionality of a product. Subsequently, the objectives of reducing the manufacturing effort and cost became increasingly important. The emergent need for product disassembly has generated the need for tools, which permit designers to evaluate the ease with which their proposed design can be disassembled.

Disassembly, the process of removing components from products at the end of their useful life is complex due to a variety of fasteners and variability in damage to the connections during its use. Because of this, the economics of the disassembly process have not been well established yet. Here, the commonly used fasteners are studied to determine the unfastening effort for widely used fasteners. Each fastener is studied in detail and the factors, which affect the unfastening process for different fastener types are determined. These factors are related to the geometry and shape of the fastener and also to the condition of their use in the product. A model, the U-Effort or Unfastening-Effort model, is presented in which the relative difficulty for unfastening is determined as a relative scaled score. The cumulative value of the score for an assembled product could be used to gauge the relative difficulty of unfastening the assemblies and subassemblies for the product. Using this approach, an estimate can be made to determine the economic cost for physically separating a product for maintenance or demanufacturing at the end of its useful life.

Current trends in environmental protection legislation indicate that manufacturer may soon be responsible, or at least share responsibility, for recycling components at the end of their useful life. In addition, the environmental conscious consumers demand products that are easy to dismantle and recycle due to limited natural resoures and limited landfill space. Under the overall concept of design for environment (DFE), the emphasis now lies on designing products, which facilitate different steps for recycling such as unfastening, disassembly, parts cleaning and refurbishment for eventual reuse. Most products are assembled from several components with the help of various types of fasteners or through bonding. Discrete fasteners and integral attachments, can usually be unfastened or detached manually. The objective of this research is to present a procedure to estimate the unfastening effort of commonly used fasteners, which will assist in obtaining an estimate of the disassembly effort and to develop guidelines for the disassembly process planning and for design for disassembly/unfastening. Using this approach, an estimate can be made of the economic cost for physically separating a product for maintenance reasons or demanufacturing at the end of its useful life.

Almost all product disassembly involves the removal of one or more fasteners. This removal process includes accessing the fastener, unlocking or releasing it, and finally extracting it. The total effort required to execute this removal action is therefore a function of (a) the type of fastener and (b) several situation specific attributes. From a design perspective therefore, if several equivalent fastener options were avaiable, and product disassembly was an end-of-life option, then a designer would select the fastener with least removal effort. In this study, several fasteners, which are commonly used in industry, have been analyzed. For each fastener an attribute driven model for estimation the fastener removal or unfastening effort is provided.

4.2 The Unfastening Process

The process of disassembly is generally manual and the economics of disassembly are still not well understood. As defined in Chapter 3, unfastening is the process of separating components or subassemblies from each other by removing fasteners or by detaching parts with integral attachments manually with or without the use of a tool. This is the reverse of the fastening process, which is defined as the process of connecting together one or more parts with the aid of external fasteners such as screws, bolts, etc. or through integral attachments embedded in the parts themselves. Unfastening can be done in different ways. The unfastening effort is different for different fastener types and is dependent on various factors. The cost of unfastening is the key to determine the recovery of reusable parts.

There are several types of fasteners or attachments commonly used in product assembly. Fasteners may be discrete fasteners, which are separate fasteners used to connect two or more parts with each other or integral attachments, where the fastener is a part of the component itself. Discrete fasteners can be unfastened to separate the parts and the integral attachments can be detached to separate the components. As shown in Chapter 3, there are two groups of discrete fasteners, threaded fasteners and non-threaded fasteners. The threaded fasteners could be screws, nuts & bolts, studs, hooks, spring toggle bolts, turnbuckles, etc. The non-threaded fasteners include nails, tacks, rivets, keys, pins, staples, clips, retaining rings, snap-type fasteners, quick release fasteners, etc. In this study the following widely used fastening elements will be considered: bolt, cantilever snap fit, cyclindrical snap fit, nail, nut and bolt, releasable clip, retaining ring, screw, staple and Velcro/zipper.

4.3 The U-Effort Model

There are considerable differences between fasteners, and even between the attributes or factors of each fastener. This research, therefore, indicates that a common model cannot be developed for all fasteners. Rather, the U-Effort model uses a common effort scale but a fastener specific effort calculator. Each calculator was experimentally derived and involved (i) isolating the causal factors, which have a significant impact on the unfastening effort for that fastener type, (ii) the use of experimentation and simulation to formulate the effort calculator, and (iii) a validation process.

In this research, it has been determined that the unfastening effort for each fastener depends upon several factors, which are mostly related to fastener size, shape or operational attributes. For each fastener the model has been limited to a maximum of four factors. The effectiveness of this model depends upon how well these factors influence the determination of the unfastening effort for each fastener. In this model each unfastening step is evaluated independently. The causal factors were isolated from a simulation and experimentation process. This involved using mock setups in which each fastener was repeatedly analyzed in the context of the tools commonly used to remove them. The results of this process are shown in Table 4.1.

Table 4.1 Factors for Different Fasteners

Fastener	Effecting Factors
bolt	bolt head shape, bolt length, bolt diameter, use of washers
cantilever snap fit	beam length, rentention angle, multiple joining
cylindrical snap fit	joint diameter, retention angle, wall thickness
nail	length, diameter, head type
nut and bolt	nut shape, nut size, washers or other auxiliary devices, unfastening torque
releasable clips	access, size
retaining rings	access, diameter, tools
screw	screw head shape, length, diameter, washers or auxiliary devices
staple	access/tools, length, hold
Velcro/zippers	access/tools, size

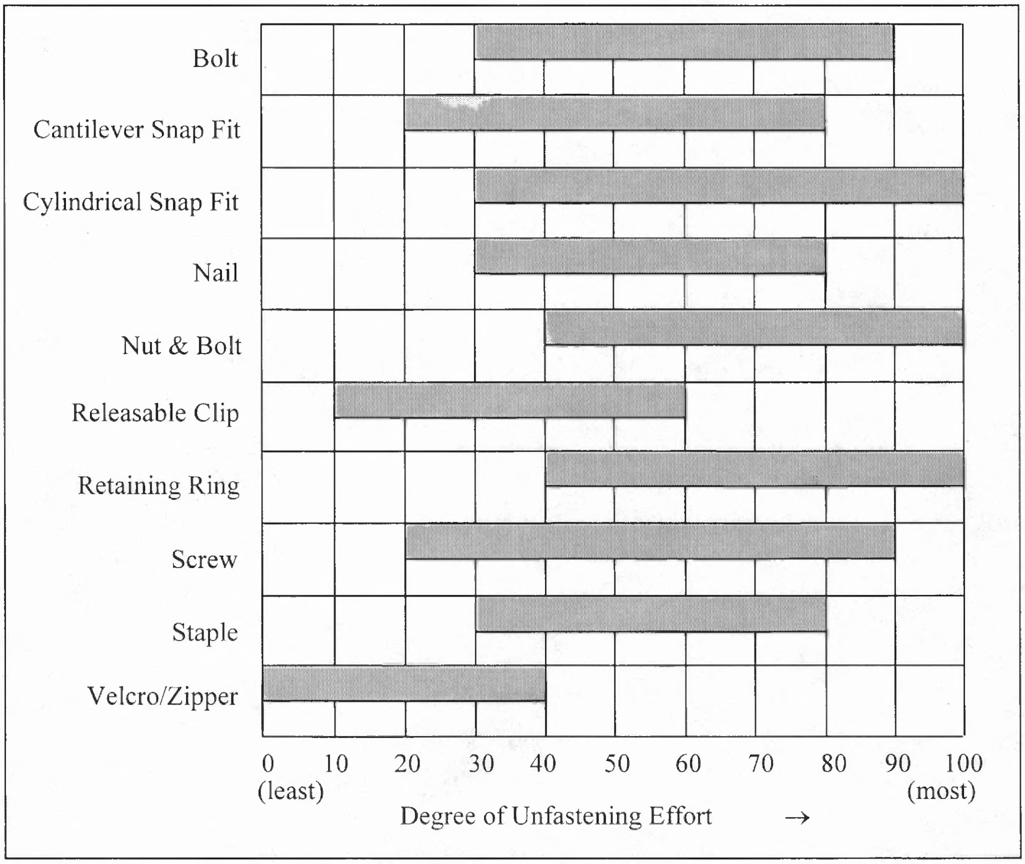


Figure 4.1 Degree of Unfastening Effort for Different Fasteners

Field tests were used to confirm that good and reliable estimates are generated for the various factors. This model can be easily adapted by other users who want to have their own weightage for various factors. The unfastening effort index scale is defined in the 0 to 100 range with 0 representing the case when no effort is required. An example for this is the removal of a Velco. 100 represents the upper bound effort of when an equivalent of 15 minutes of labor time is required for the unfastening. Field surveys indicate that times beyond this bound would make the disassembly uneconomical. For each fastener, there is a minimum and maximum value assigned to the effort as illustrated in Figure 4.1. It can be easily seen that the Velcro/zipper fastener needs the least amount of unfastening effort and the maximum possible effort is 40 on the scale. The bolt, on the other hand, needs a minimum effort of 30 due to the requirement of a tool for unfastening and the maximum unfastening effort may be 90 on the same scale. The minimum and maximum values of unfastening effort for different fasteners have been established after considerable research collaboration with industry involved with disassembly of electronic products.

4.4 Estimating the Unfastening Effort for Each Fastener

There are many factors involved, which influence the unfastening process of fasteners. To determine the unfastening effort of different fasteners, one issue to be considered is the assemble and disassembly motion, see more in Chapter 5. Certain types and directions of motions are used to disengage or remove specific fasteners in order to take a product apart. The access available to impart such motion greatly influences the degree of difficulty and hence access is one of the most important factors in this study. It is usually possible to unfasten discrete fasteners and integral attachments. However, there are cases where unfastening should be possible, but due to certain reasons it becomes very difficult (e.g. environmental exposure). The environmental exposure may cause damage to the screw and bolt heads creating difficulty in the unfastening of such fasteners. Thus the shape of the fastener heads is a leading factor affecting the unfastening effort. The other important factor to consider is the requirement of a tool. The selection of the tool needed depends on the fastener type and on the condition of the fastener. Usually, if standard tools are used for the fastening process, the same standard tools would often be sufficient for unfastening. Sometimes regular tools are not able to access the fasteners and special tools for unfastening are needed. The location of the fasteners also plays an important role in determining the unfastening effort.

A product generally consists of several subassemblies or components, which have been assembled together. In order to proceed with the disassembly, it is important to determine the type and number of fasteners holding the particular subassembly to the product. During use, the subassembly and/or the fasteners may have been damaged. For a damaged fastener, unfastening may not be feasible and only destructive disassembly is possible.

Table 4.2 Unfastening Tools and Problems in Unfastening

Type of Fastener	Unfastening Tools	Problems in Unfastening
bolt	wrenches, pliers, spanners	accessibility, corrosion, damage to head
cantilever snap fit	screwdriver, punches, pliers	accessibility, difficult because of muliple latches joining
cylindrical snap fit	pliers, screwdrivers, punches	accessibility, corrosion
nail	pliers, hammers, or hacksaw	corrosion, accessibility
nut and bolt	screwdriver, ratchets, spanners, wrenches, Allen keys, pliers	accessibility, corrosion, damage to the nuts, missing nuts or screw heads
releasable clip	manual, pliers	accessibility
retaining ring	ring pliers	accessibility
screw	screwdriver	corrosion, damage, accessibility
staple	staple pliers, flat tipped screw driver	accessibility, corrosion
Velcro/zippers	manual, pliers	accessibility

Table 4.2 lists the problems, which may occur with different types of fasteners. These problems have been studied and influenced the selection of factors considered in the determination of the unfastening effort for each fastener. The factors, which affect unfastening for each type of fastener, are listed in Table 4.1. For example, in the case of a bolt, there are four factors affecting the unfastening effort and these are the shapes of the bolt head, bolt length, bolt diameter and the use of washers. Each factor contributes towards the unfastening effort index and carries a weight, which has been assigned based on their relative difficulty in unfastening. As explained in the next section, the shape of the bolt head can contribute up to a maximum of 20 on the unfastening effort index scale. Factor C_1 has been given values based on the shape of the bolt head. As shown in Table 4.3, a countersunk bolt carries a maximum value of 15 whereas a hook bolt obviously easiest to unfasten (does not need any tool) has a zero value for this factor. Factor C_2 is dependent on the length of the bolt. It has been determined that the value of this factor is directly proportional to the bolt length as the time for unfastening increases with the increase of the bolt length. Generally, following equation for the unfastening effort index f can be formed:

$$f = B_{min} + W_1 \cdot C_1 + W_2 \cdot C_2 + W_3 \cdot C_3 + W_4 \cdot C_4 = B_{min} + \sum W_i \cdot C_i \qquad (4.1)$$

In this equation, B_{min} is the lower margin value of the basic unfastening effort for the specific fastening element (Figure 4.1). C_i is the factor for the different fasteners (usually three or four parameters, see Table 4.1), and W_i gives the corresponding weight value. According to the importance of the single factors the weight factor can be distributed, so that,

$$B_{min} + \sum W_i = B_{max} \qquad (4.2)$$

B_{max} is the upper margin value of the basic unfastening effort for the specific fastening element (Figure 4.1). In the next section, the ten individual fastener types are considered in detail and the factors affecting each of these fasteners are explained.

4.4.1 Bolt

It is assumed that bolts are being used to assemble metal parts where additional strength is required than a screw can provide. In this study, it is has been established that four factors have an effect on the unfastening effort. These factors are the shape of the bolt head, the length of the bolt, the bolt size, and washers. The proposed unfastening effort model for an unfastening index f then would be:

$$f_{bolt} = 30 + 15 \cdot C_1 + 15 \cdot C_2 + 15 \cdot C_3 + 15 \cdot C_4 \qquad (4.3)$$

In this equation, coefficient C_1, C_2, C_3, and C_4, respectively, are dependent on the four factors stated above, which are described next.

- Shape of bolt head

Basically, two major groups of bolts have been standardized: roundhead bolts and machine bolts (sometimes called "wrench-head bolts"). The most common head shapes are listed in Table 4.3.

Table 4.3 Factor C_1 for the Bolt Depending on the Head Shape

Head shape	C_1
square	0.2
hexagonal	0.2
round (carriage)	0.8
countersunk	1.0
T-head	0.5
askew	0.6
eyebolt	0.0
hook bolt	0.0

♦ Length of bolt

For each bolt size, there are different bolt lengths available. The length increments can vary from 1/4, 1/2, 1 to 2 inches. A classification of the bolt length is given in Figure 4.2.

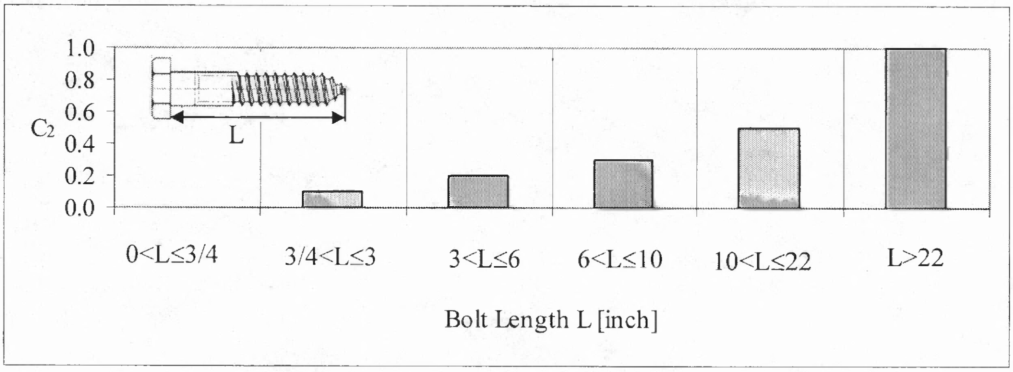


Figure 4.2 Factor C_2 for the Bolt Depending on the Bolt Length L

♦ Bolt size

Bolts come in a certain number of different sizes. Usually they are based on their diameter. Four bolt diameter groups are shown in Figure 4.3.

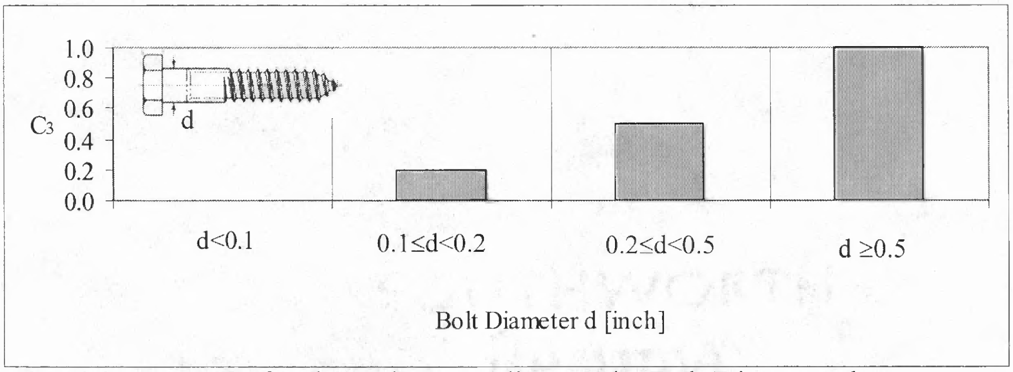
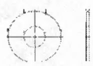
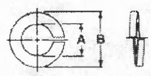
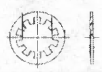


Figure 4.3 Factor C_3 for the Bolt Depending on the Bolt Diameter d

♦ Use of washer or other auxiliary devices

If washers or other auxiliary devices are used the unfastening process becomes more difficult, see Table 4.4.

Table 4.4 Factor C_4 for the Bolt Depending on the Use of a Washer or Other Auxiliary Device

Device		C_4
no washer		0.0
flat washer		0.2
split lock washer		0.8
shakeproof lock washer		1.0

4.4.2 Cantilever Snap Fit

It is assumed that the cantilever snap fit is used as an integral attachment in plastic products. In that case, the three factors most likely to affect unfastening are the beam length, the retention angle, and multiple joining (not regarding the material properties, which play an important role).

A suggested equation for an unfastening index f then would be:

$$f_{cant} = 20 + 20 \cdot C_1 + 20 \cdot C_2 + 20 \cdot C_3 \tag{4.4}$$

- Beam length

The longer the cantilever beam, the more flexible it is and the easier it can be released as shown in Figure 4.4.

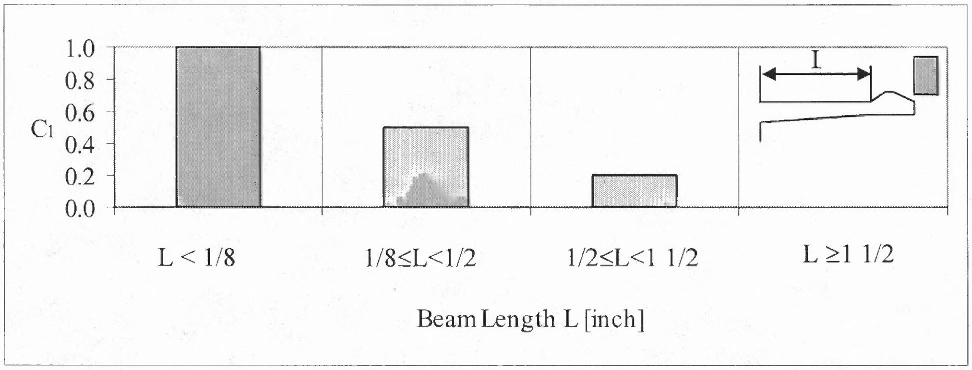


Figure 4.4 Factor C_1 for the Cantilever Snap Fit Depending on the Beam Length L

- Retention angle

The retention angle determines if a cantilever snap fit can be unfastened or not. Figure 4.5 shows that the bigger the angle the more difficult is the unfastening.

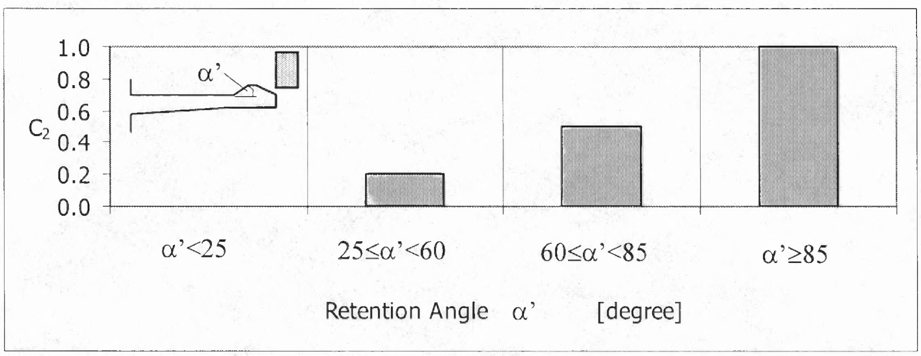


Figure 4.5 Factor C_2 for the Cantilever Snap Fit Depending on the Retention Angle α'

♦ Multiple joining

Cantilever snap fits are integral attachments and it is quite common that a connection has more than one cantilever to hold parts together, see Figure 4.6.

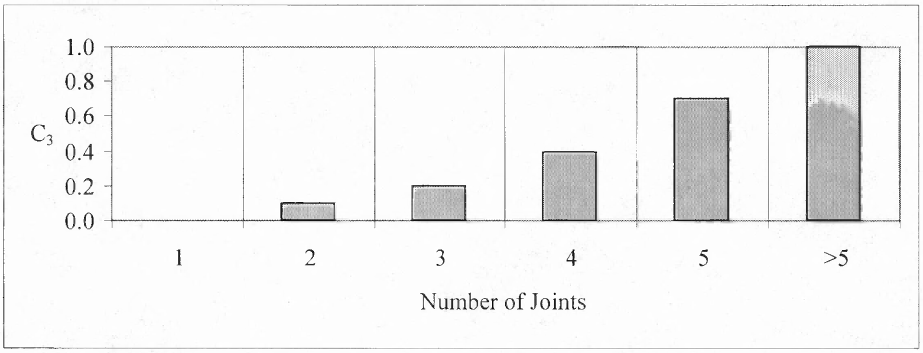


Figure 4.6 Factor C_3 for the Cantilever Snap Fit Depending on the Number of Joints

4.4.3 Cylindrical Snap Fit

It is assumed that the cylindrical snap fit (annular snap fit) is used as an integral attachment in plastic products. The three factors most likely to affect unfastening are the joint diameter, the retention angle, and the wall thickness (not regarding the material properties, which are very important as well). A suggested equation for an unfastening index f then would be:

$$f_{cyl} = 30 + 25 \cdot C_1 + 25 \cdot C_2 + 20 \cdot C_3 \qquad (4.5)$$

♦ Joint diameter

The bigger the joint diameter is the more effort is necessary to unfasten the parts, as shown in Figure 4.7.

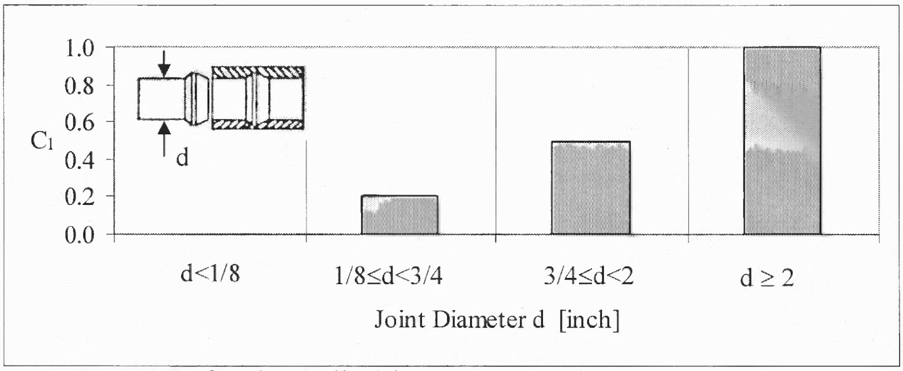


Figure 4.7 Factor C_1 for the Cylindrical Snap Fit Depending on the Joint Diameter d

♦ Retention angle

The retention angle determines if a cylindrical snap fit can be unfastened or not. Figure 4.8 shows that the bigger the angle is the more difficult the unfastening gets.

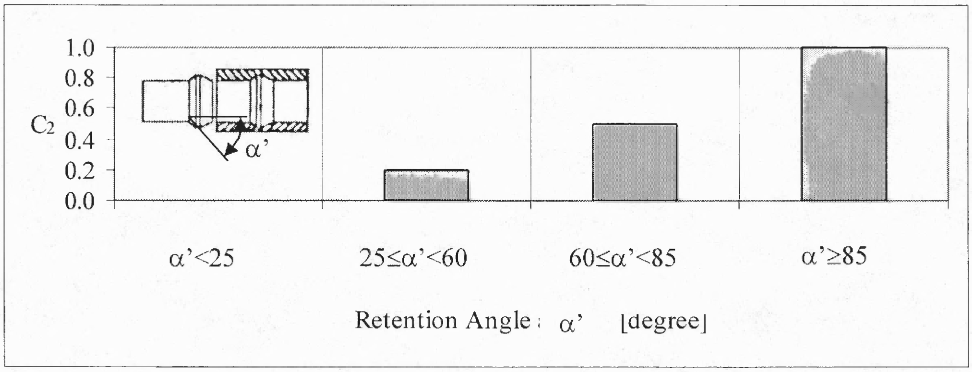


Figure 4.8 Factor C_2 for the Cylindrical Snap Fit Depending on the Retention Angle α'

♦ Wall thickness

The thinner the wall thickness of the outside cylinder (tube) is the more flexible the tube becomes and the easier it can be released, see Figure 4.9.

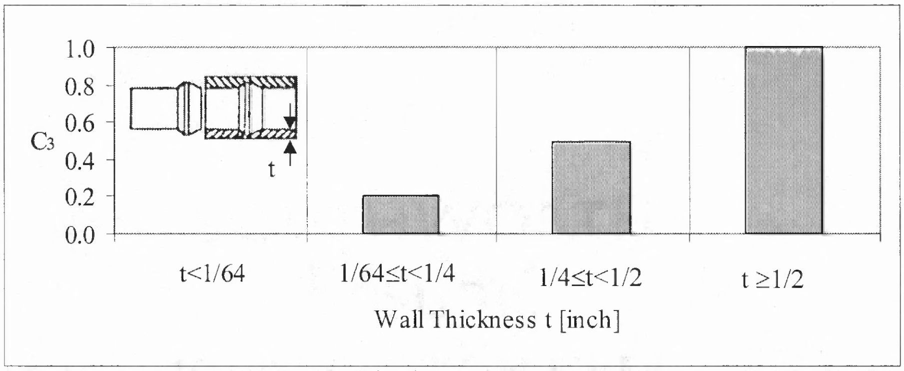


Figure 4.9 Factor C_3 for the Cylindrical Snap Fit Depending on the Wall Thickness t

4.4.4 Nail

It is assumed that nails are being used to assemble wood to wood or metal sheet to wood or fabric to wood. For this situation, the three factors most likely to affect the unfastening effort are the length, the diameter, and the head type.

A suggested equation for an unfastening index f then would be:

$$f_{nail} = 30 + 10 \cdot C_1 + 20 \cdot C_2 + 20 \cdot C_3 \tag{4.6}$$

♦ Length

It is assumed that the longer the nail is the more effort it takes to unfasten it. The nail length is usually measured in penny size (abbreviated d). Figure 4.10 shows the factor C_1 in dependency of the nail length.

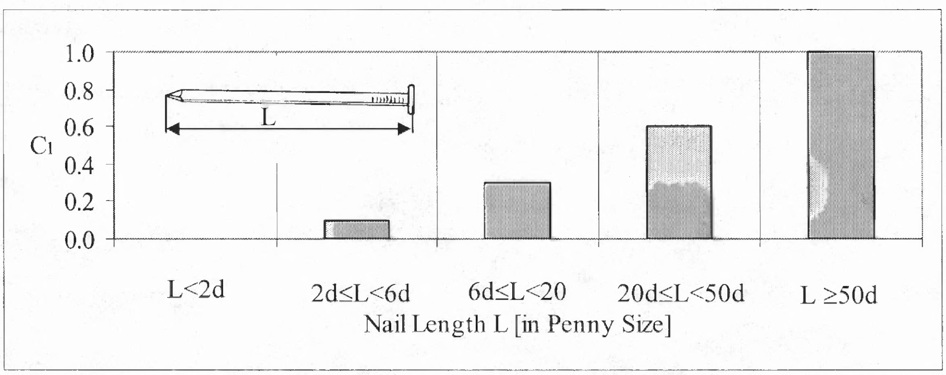


Figure 4.10 Factor C_1 for the Nail Depending on the Nail Length L

- Diameter of nail

The resistance of a nail to withdrawal increases almost directly with its diameter; if the diameter of the nail is doubled, the holding strength is doubled. This influences the unfastening effort. Further, the head diameter relates to the nail diameter. Nail diameters are measured in wire gauge numbers. As the wire gauge numbers go down, the diameter of the nail goes up, see Figure 4.11.

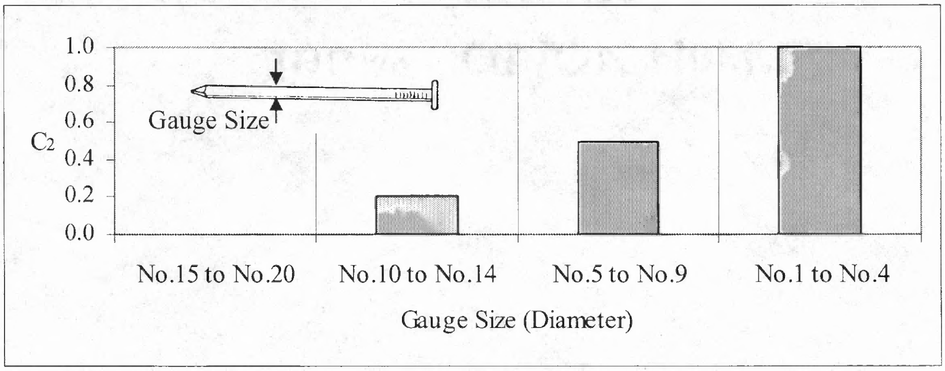


Figure 4.11 Factor C_2 for the Nail Depending on the Gauge Size

- Head type

The head type of a nail influences the clearance of the head and with it the access to use a tool. It is assumed that if the head size is very small the unfastening is more difficult than with a bigger nail head, because with a bigger head the nail can be grabbed easier with pliers or hammer claws. If the head of a nail is sunk in the unfastening is more difficult than if it is not sunk in.

Table 4.5 Factor C_3 for the Nail Depending on the Nail Head Type

shapes of nail heads	C_3	shapes of nail heads	C_3
flat	0.1	round	0.2
double-headed	0.0	casing	0.9
button	0.2	oval	0.3
curved	0.6	cup head	0.8
sinker	0.9	projection	0.4
cupped oval	0.3	oval countersink	0.9
countersink	0.9	headless	1.0
hook head	0.1	slotted	0.5

4.4.5 Nut & Bolt

It is assumed that nuts and bolts are used to assemble metal parts. A bolt, with an integral head on one end and a thread on the other end, is passed through clearance holes in two parts and draws them together by means of a nut screwed on the threaded end. Nuts are used with bolts to develop clamping action on a joint by moving up the threaded shaft of the fastener to oppose the force applied by the head upon tightening. For this situation, the following four factors have effects on the unfastening: the shape of the nut, the nut size, the use of washers, and the unfastening torque.

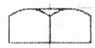
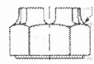
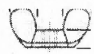
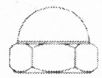
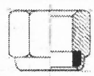
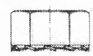
A suggested equation for an unfastening index f then would be:

$$f_{n\&b} = 40 + 15 \cdot C_1 + 15 \cdot C_2 + 15 \cdot C_3 + 15 \cdot C_4 \tag{4.7}$$

- Nut shape

There are many different types of nuts. The nut shape determines if standard tools or special tools have to be used.

Table 4.6 Factor C_1 for the Nut & Bolt Depending on the Nut Type

Nut type		C_1
square		0.3
hexagonal		0.2
jam		1.0
castellated		0.9
wing		0.1
cap or acorn		0.5
thumb		0.0
stop		0.9
lock		1.0

- Nut size

Every nut type has a standard set of dimensions. The diameter determines the thickness of the nut, and with this the thread length and how many turns have to be made to remove the nut from the bolt, as can be seen in Figure 4.12.

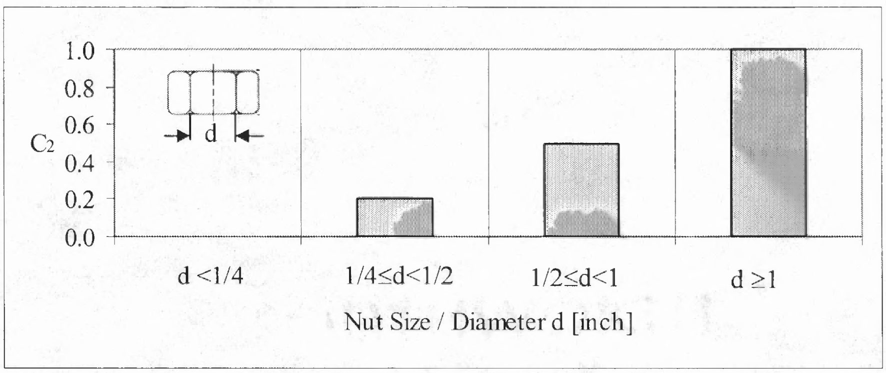
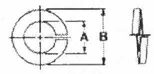
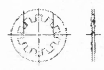


Figure 4.12 Factor C_2 for the Nut & Bolt Depending on the Nut Diameter d

♦ Use of washer or other auxiliary devices

If washers or other auxiliary devices are used the unfastening process becomes more difficult, shown in Table 4.7.

Table 4.7 Factor C_3 for the Nut & Bolt Depending on the Use of a Washer or Other Auxiliary Device

Device	C_3
no washer	0.0
flat washer 	0.2
split lock washer 	0.8
shakeproof lock washer 	1.0

♦ Unfastening torque

If a great torque is necessary to tighten a nut-bolt connection, then the torque to unfasten it again is still greater than that. If the torque is too high, the bolt is shearing before it would unfasten. That means, the lower the tightening torque the lower the torque to loose the nut, and the lower the unfastening effort. This relationship is shown in Figure 4.13.

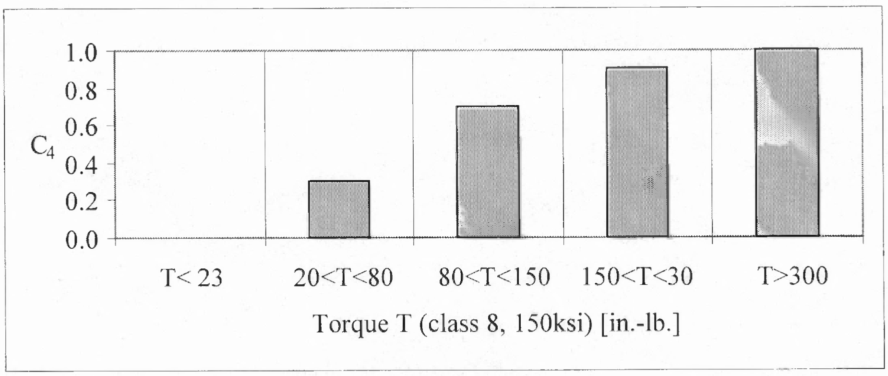


Figure 4.13 Factor C_4 for the Nut & Bolt Depending on the Torque T

4.4.6 Releasable Clip

It is assumed that these clips snap into an annular groove on a shaft or pin with ends projecting beyond shaft surface. The unfastening effort is effected by the access and size of the clips.

The suggested equation for an unfastening index f then would be:

$$f_{rclip} = 10 + 25 \cdot C_1 + 25 \cdot C_2 \tag{4.8}$$

- Access

Usually the clip can easily be unfastened or released, mostly by hand. The more access is given the easier it is to unfasten it, too(see Table 4.8).

Table 4.8 Factor C_1 for the Releasable Clip Depending on the Access and Tool

Access / tool to unfastening	C_1
Manual removal from top	0.0
Manual removal from side	0.3
Pliers necessary	1.0

- Size

The bigger the clip is the easier it is to grasp it as shown in Figure 4.14.

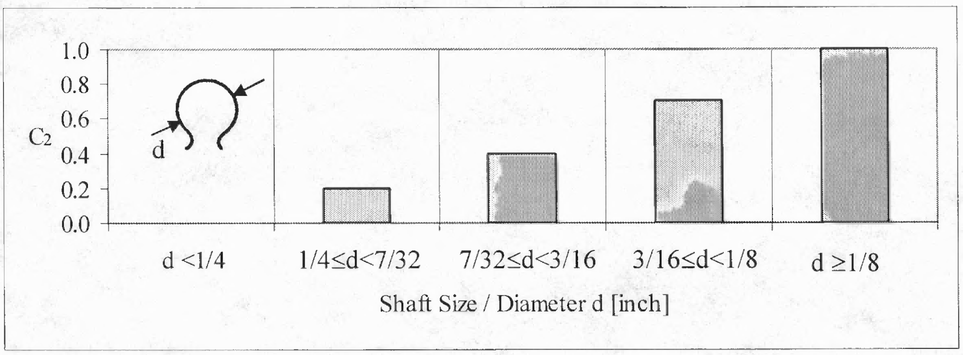


Figure 4.14 Factor C_2 for the Releasable Clip Depending on the Shaft Diameter d

4.4.7 Retaining Ring

It is assumed that retaining rings are used for locking and retaining components on shafts or in housings and bores. The rings generally are made of resilient materials, so the fasteners may be deformed elastically to a considerable degree and still spring back to their original shape. The unfastening effort is then based on the access or position of the ring, the diameter and need of special tools.

A suggested equation for an unfastening index f then would be:

$$f_{rring} = 40 + 20 \cdot C_1 + 20 \cdot C_2 + 20 \cdot C_3 \tag{4.9}$$

- Access, position of the ring

There are usually two possibilities: axial (external and internal) rings or radial assembled ones as shown in Table 4.9.

Table 4.9 Factor C_1 for the Retaining Ring Depending on the Access

Access	C_1
Radial retaining rings – radial access	0.0
Axial retaining rings – axial access, parallel to shaft or bore	1.0

- Diameter

Retaining rings are available in a very wide range of sizes, see Figure 4.15.

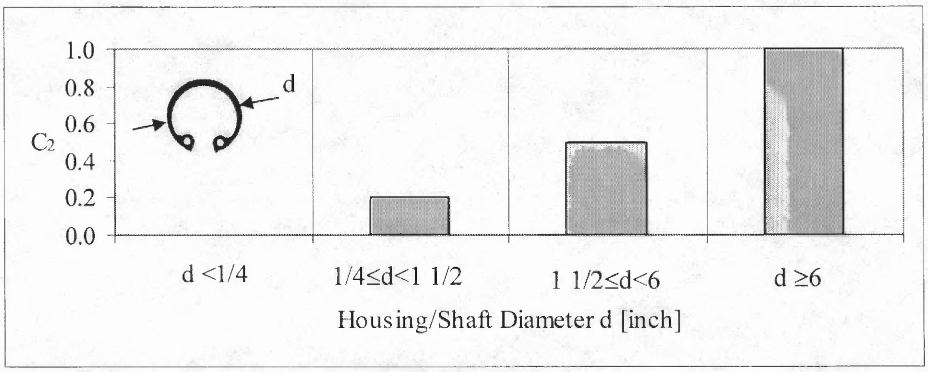
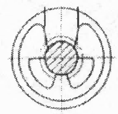
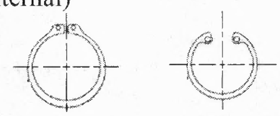


Figure 4.15 Factor C_2 for the Retaining Ring Depending on the Housing or Shaft Diameter d

- Special tool (ring pliers)

In the most cases, it is necessary to use special tools to unfasten the retaining rings, see Table 4.10.

Table 4.10 Factor C_3 for the Retaining Ring Depending on the Use of Unfastening Tools

Types	Unfastening Tools	C_3
Radial rings (E ring, spring clip, crescent ring, bowed ring, C ring, Klipring) 	Manual or Pliers	0.0
Axial rings (external and internal) 	Pliers necessary	1.0

4.4.8 Screw

It is assumed that screws are being used to assemble metal parts. (Wood screws are not considered here.) For this situation, the following four factors have effects on the unfastening: the shape of the screw head, the length of the screw, the screw size, and the use of washers.

A suggested equation for an unfastening index f then would be:

$$f_{screw} = 20 + 20 \cdot C_1 + 20 \cdot C_2 + 15 \cdot C_3 + 15 \cdot C_4 \qquad (4.10)$$

- Shape of screw head

Screws come in many types, and include machine, cap, set, thumb, socket, lag, miniature, and self-tapping types. Like bolts, screws are classified by their head type, see Table 4.11.

Table 4.11 Factor C_1 for the Screw Depending on the Head Shape

Head shape	C_1
round, slotted	0.2
countersunk, slotted	0.3
Phillips	0.3
Torx drive	0.6
hexagonal	0.7
square	1.0
fillister, slotted	0.2
hex-socket	0.8
spline socket	1.0
thumb	0.0

♦ Length of screw

For each screw size, there are different screw lengths available. The length increments can vary from 1/4, 1/2, 1 to 2 inches. A classification of the screw length is given in Figure 4.16.

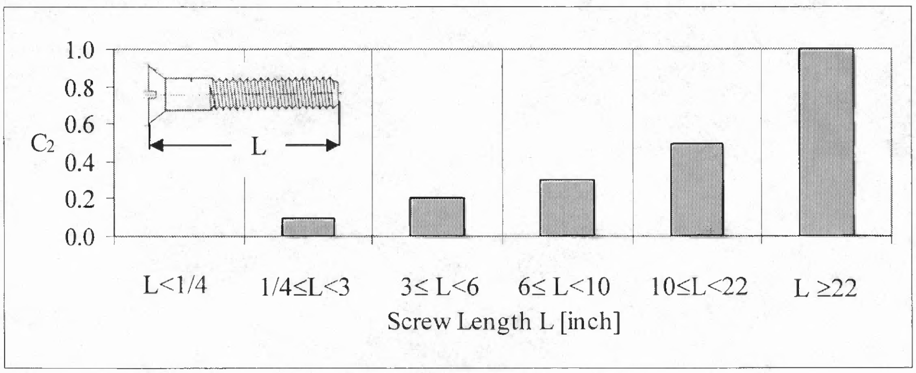


Figure 4.16 Factor C_2 for the Screw Depending on the Screw Length L

♦ Screw size

Screws are available in many different sizes, see a classification in Figure 4.17. Usually they are based on their diameter.

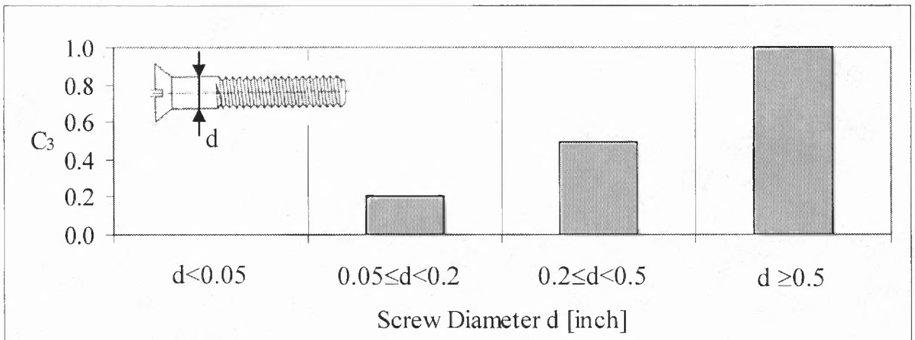
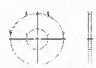
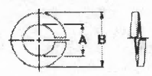
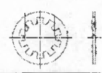


Figure 4.17 Factor C_3 for the Screw Depending on the Screw Diameter d

♦ Use of washer or other auxiliary devices

If washers or other auxiliary devices are used the unfastening process becomes more difficult, see Table 4.12.

Table 4.12 Factor C_4 for the Screw Depending on the Use of a Washer or Other Auxiliary Device

Device		C_4
no washer		0.0
flat washer		0.2
split lock washer		0.8
shakeproof lock washer		1.0

4.4.9 Staple

It is assumed that staples are used to attach a thin layered material to wood, cork, or similar materials. The unfastening is influenced mainly by the access, the length of the staple and the staple hold.

A suggested equation for an unfastening index f then would be:

$$f_{staple} = 30 + 10 \cdot C_1 + 20 \cdot C_2 + 20 \cdot C_3 \tag{4.11}$$

♦ Access / special tool (staple pliers)

To unfasten a staple it is necessary to have enough space to use staple pliers. Some staples can be removed from the same direction they were inserted. Sometimes it is necessary to also have access from the backside in order to avoid tearing of the parts. See the dependency of C_1 in Table 4.13.

Table 4.13 Factor C_1 for the Staple Depending on the Access and the Need for Tools

Access / tool to unfastening	C_1
Manual removal	0.0
Staple lifter to use from top	0.3
Staple lifter to use from bottom	1.0

♦ Length of staple

Today, most staples are driven by mechanical staplers. Therefore, staples are produced in five standard leg lengths for average use. The shorter the staple leg, the easier it is to remove the staple as shown in Figure 4.18.

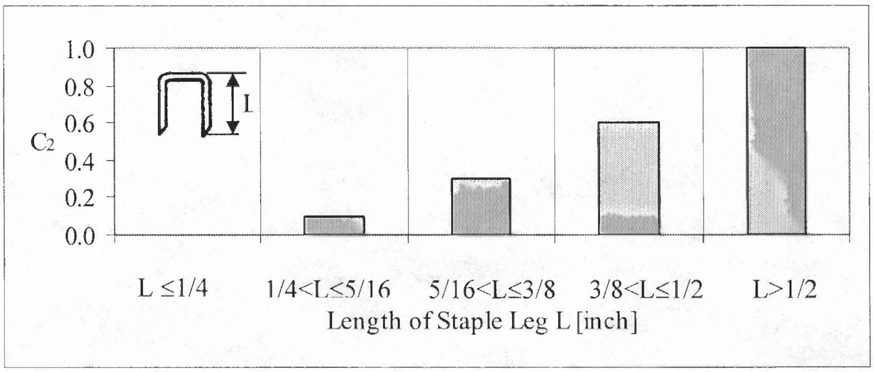


Figure 4.18 Factor C_2 for the Staple Depending on the Length of the Staple Leg L

♦ Staple hold

With different mechanical staplers, like staple gun, hammer tacker, or pliers stapler, the final shape of the staple to hold the parts together can vary. Depending on the hold is the degree of how much effort is needed to remove a staple, see Table 4.14.

Table 4.14 Factor C_3 for the Staple Depending on the Stable Hold

Staple hold	C_3
straight	0.0
outward-pointing	0.5
inward-pointing	1.0

4.4.10 Velcro/Zipper

It is assumed that Velcro is made of either nylon or polyester. Further, it is assumed that Velcro is separated in two pieces during the unfastening process, the same for the zipper. Unfastening here is not the removing of the glued Velcro tape or the sewed in zippers, only the separation of them. The unfastening effort in this case is influenced by the access and necessary tools and the size.

A suggested equation for an unfastening index f would be:

$$f_{Vel/Zip} = 20 \cdot C_1 + 20 \cdot C_2 \qquad (4.12)$$

♦ Access/tools

The more access is given the easier it is to unfasten, too (see Table 4.15). Velcro and zippers usually can be separated manually, but there might be cases where the use of pliers becomes necessary.

Table 4.15 Factor C_1 for the Velcro/Zipper Depending on the Access and the Use of Tools

Access/tools to unfasten	C_1
Manual removal from top	0.0
Manual removal from side	0.3
Pliers necessary	1.0

♦ Size

Velcro tapes are usually available in two widths. The amount of tape that is required for any given job is directly proportional to the strength needed. Therefore, the larger the Velcro surface, the more strength is needed to unfasten it, see Figure 4.19. Zippers are available in a number of different lengths for different purposes. The longer the zipper is the more difficult it gets to unfasten it.

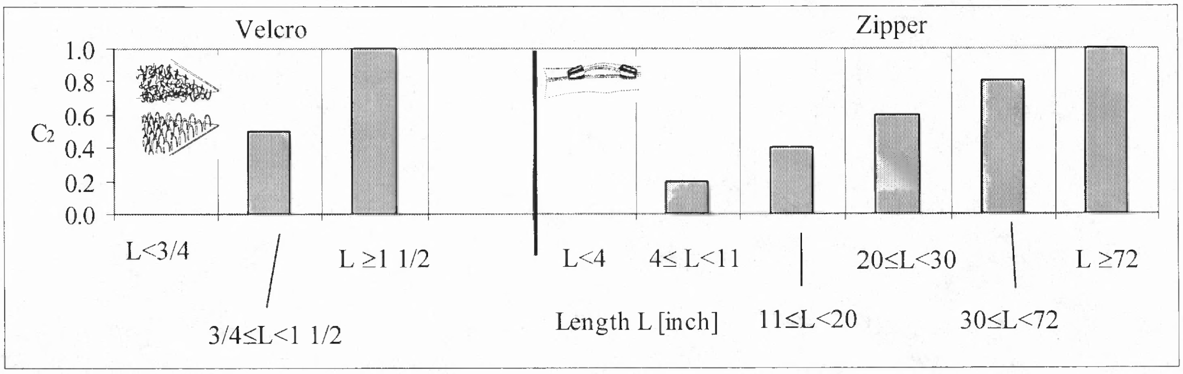


Figure 4.19 Factor C_2 for the Velcro/Zipper Depending on the Length L

4.5 Example

To show how to use the U-effort scores, an example for the unfastening of a Walkman is given here. As shown above, the equation for the unfastening index f for the cantilever snap fit is

$$f_{cant} = 20 + 20 \cdot C_1 + 20 \cdot C_2 + 20 \cdot C_3 \tag{4.4}$$

and the equation for the screw is

$$f_{screw} = 20 + 20 \cdot C_1 + 20 \cdot C_2 + 15 \cdot C_3 + 15 \cdot C_4 \tag{4.10}$$

These equations are necessary for the calculation of the U-Effort of the unfastening of the Walkman, because there are two cantilever snap fits to unfasten and four different screws to remove.

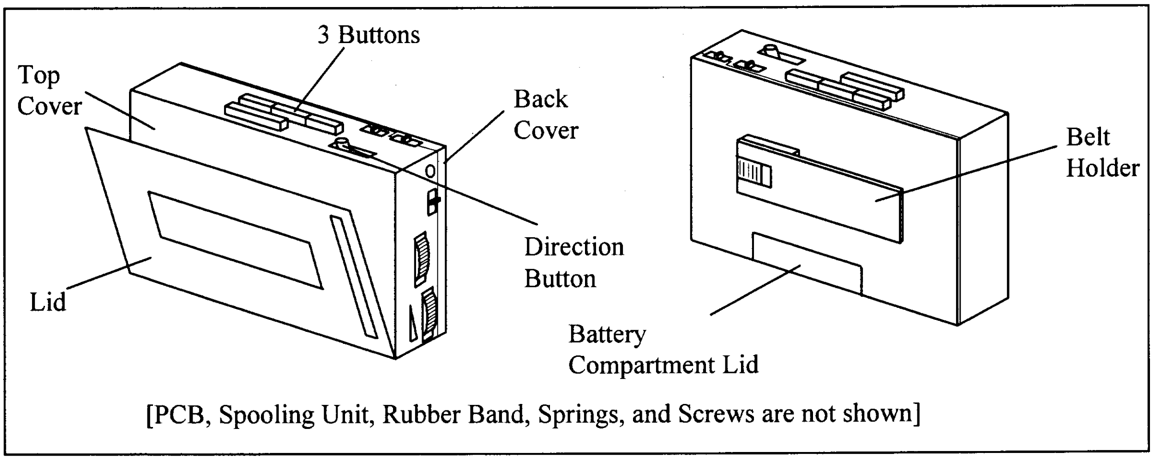


Figure 4.20 Components of a Walkman

Table 4.16 Example U-Effort Model for a Walkman

Fastener ID	Type	C_1	C_2	C_3	C_4	Score f	#	Sum
1	cantilever	1.0 (1/8")	0.5 (60-85)	0.1 (2)		52	1	52
2	cantilever	0.5 (3/16")	0.5 (60-85)	0.0 (1)		40	1	40
3	screw	0.3 (Phillips)	0.1 (5/16)	0.2 (~0.06)	0.0 (no w.)	31	4	124
4	screw	0.3 (Phillips)	0.0 (3/16)	0.2 (~0.07)	0.0 (no w.)	29	1	29
5	screw	0.3 (Phillips)	0.0 (3/16)	0.0 (~0.045)	0.0 (no w.)	26	2	52
6	screw	0.3 (Phillips)	0.0 (3/16)	0.2 (~0.07)	0.0 (no w.)	29	2	58
							SUM:	355

For the case shown in Table 4.16, the minimum unfastening score is 220 (with all $C_i = 0$). That means at this value the effort to unfasten the product is the lowest. With a score of 355, about average effort is needed to unfasten the Walkman. This value could be transferred to costs if the values for the different components and/or materials are known. In addition, a study to compare the unfastening effort with the values of different product brands could be done.

CHAPTER 5

DISASSEMBLY MOTION AND THE U-EFFORT MODEL

The unfastening effort for the different fasteners as determined in the previous chapter was dependent on factors such as fastener type, geometry, shape, and accessibility. In addition, disassembly motion also plays an important role in the unfastening process. In this chapter, it is described what disassembly motions are and in which way they effect the unfastening effort.

5.1 Introduction to Disassembly Motion

Cutting down the time needed to disassemble products is vital to encouraging recycling. Therefore, it is important to analyze the disassembly process. To find out, what kind of fastener can be unfastened, and which fastening elements need destructive disassembly through cutting or sawing, it is necessary to look at the assembly and disassembly motions. Certain types and directions of motion are used to disengage specific fasteners in order to take a product apart. The assembly direction is the direction of motion required to locate and lock the mating part relative to the base part. Furthermore, the assembly motion is a set of simple movements that describes the last motion the fastener/part makes as it is attached to the base/other component. Assembly direction can consist of motion in a single direction such as a push, pull, or a twist, or motion in two directions such as a push followed by a twist. The part separation is usually accomplished by reversing one of the simple assembly motions. For integral attachments, the retention direction is the direction in which a lock feature eliminates motion and therefore takes service load. For a lock, the retention direction is almost always the exact opposite of the insertion direction. Bonenberger [1995] developed a methodology of assembly/ disassembly symbols for integral attachments. This concept was here enhanced to a general model, which also includes other fastening methods besides integral attachments.

5.2 Types of Motion

- Push: To push can be defined as to press against something with force, to drive or impel by pressure, to push an object without striking, to apply pressure against for the purpose of moving. A push is a linear movement with contact shortly before final nesting and locking. It is the simplest, most used motion and therefore the least expensive. Furthermore, a push motion is very good for automated assembly.

  Example: Push in a pin or key in a slot (also see Figure 5.1).

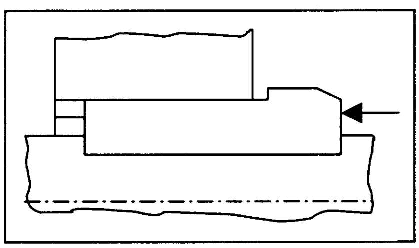


Figure 5.1 Push Motion – Inserting a Key in a Slot

⊙ Pull: To pull can be defined as to draw or attempt to draw towards one forcefully, to move or operate by the motion of drawing towards one, to remove from a fixed position; extract or to tug at. Pull is the reversed movement of push.

  Example: Remove a pulley from a shaft (see Figure 5.2).

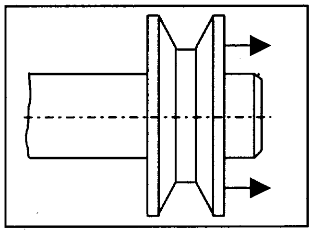


Figure 5.2 Pull Motion – Removing a Pulley from a Shaft

Twist: To twist can be defined as to distort, as a solid body by turning one part relatively to another around an axis passing through both; to turn or open by turning. A twist is a rotational movement with a part with axisymmetric locators and locks rotating around its axis to engage the base. For integral attachments, a twist motion is rarely used alone. It is usually used as the last portion of a compound assembly motion where it is very effective.

Example: Turning a screwdriver in a Screw (see Figure 5.3).

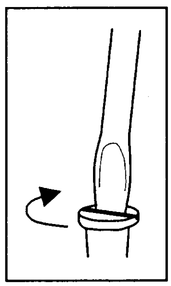


Figure 5.3 Twist Motion – Turning a Screwdriver in a Screw

Spin: To spin can be defined as to cause to turn round rapidly, to whirl, or to twirl as to spin a top. It is also a rotational movement.

Example: Removing a knob or a loosened nut (see Figure 5.4)

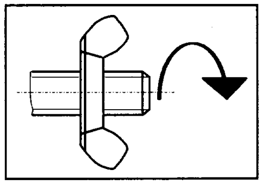


Figure 5.4 Spin Motion – Removing a Loosened Nut

⬆ Lift: To lift is defined as to move in a direction opposite to that of gravitation, to raise, to elevate, to bring from a lower place to a higher, sometimes implying a continued support or holding in the higher place.

Example: Lifting an unfastened component out of an assembly (see Figure 5.5).

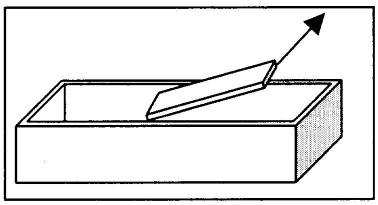


Figure 5.5 Lift Motion – Lifting an Unfastened Component out

► Slide: To slide can be defined as to move along the surface of a body by slipping, to move gently without friction or hindrance, like a cover which opens by sliding, e.g. clasp or brooch for a belt. It is also to move over a surface while maintaining smooth continuous contact, to glide. A slide is a linear movement where early contact and additional relative movement describes it. For example, the integral attachment lug uses this motion.

Example: Close or open a cover by sliding (see Figure 5.6).

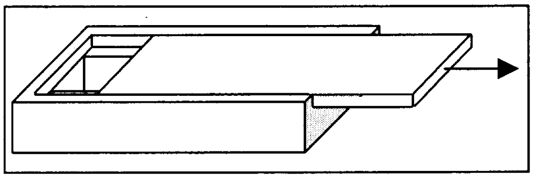


Figure 5.6 Slide Motion – Sliding a Cover Open

▼ Tip: To tip can be defined as to strike slightly, to tap, to tilt, to lower one end of something with a light touch, to move to a slanting position. It is a rotational movement with one end of the part engaged to the base followed by part rotation towards base into locked position. This motion is for example a common movement for lugs.

Example: To separate a glass screen after the holding cover has been unfastened (see Figure 5.7).

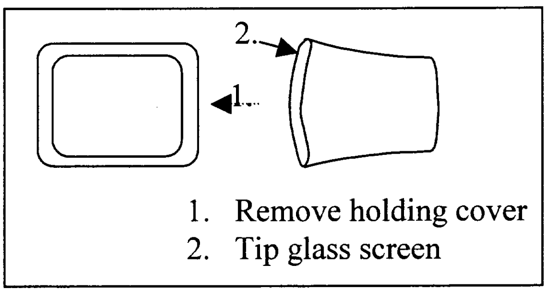


Figure 5.7 Tip Motion – Separating a Glass Screen of a Monitor

Impact: To impact can be defined as to drive close, to press firmly together or contact by forcible touch, collision.

Example: Hitting a nail with hammer or applying force suddenly on a spanner to loosen a nut (see Figure 5.8).

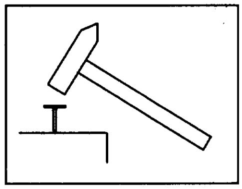


Figure 5.8 Impact Motion – Hitting a Nail with a Hammer

Pry: To pry can be defined as to raise, move or force open with a lever, to obtain with effort or difficulty.

Example: To open a housing of a remote control, which is fastened by multiple snap fits (see Figure 5.9).

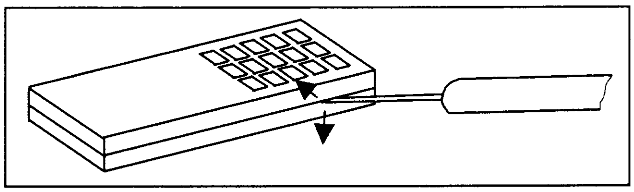


Figure 5.9 Pry Motion – Opening a Housing of a Remote Control

Bend: To bend can be defined as to strain, turn or deflect something from a normal position or out of shape.

Example: To bend a cantilever snap for removal (see Figure 5.10).

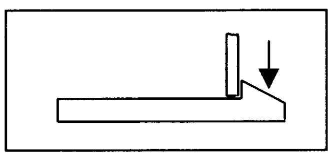


Figure 5.10 Bend Motion – Removing a Cantilever Snap Fit

Motions not considered are to fold, to turn (same as to twist or spin), to hold, to grip, e.g. cutting wire with scissors, to grasp, to cut, to flip - some of these are not unfastening. These motions are manual and can be applied in direction

- parallel to the fastening axis
- perpendicular to the fastening axis
- eccentric to the fastening axis

Furthermore, motions can be applied:

- directly on the component
- on the fastener
- on the tool

Some of the motions need less force and effort than other motions. For each type of fastener, there is a particular motion or a combination of motions. Some motions can be applied with one hand, others with two hands. In some cases there might be even a fixture needed. For different fastening elements or methods, the disassembly motion or motion combinations with their visual symbols are presented in Table 5.1. In addition, for every fastening element the appropriate disassembly tools and possible problems, which can occur during the unfastening, are listed here. Furthermore, from Table 5.1, it can be seen, that generally adhesive bonding and energy bonding can only be disassembled in a destructive way, like cutting, shearing, sawing, etc. That means, if recycling and demanufacturing is an important issue for a product then it is not recommended to use adhesive bonding or energy bonding as fastening method.

Table 5.1 Disassembly Motion for Different Fastening Methods

	Disassembly Motion		**Disassembly Tools**	**Possible Problems**
Discrete Fasteners / Nonintegral Attachments:				
Threaded Fasteners				
Screw	Twist, Spin	↻	Screwdriver	Access., Corr., Dam.
Nut & Bolt	Twist, Spin	↻	Ratchet, Spanner, Wrench	Access., Corr., Dam.
Stud	Twist, Spin	↻	Spanner, Wrench	Access., Corr., Dam.
Hook	Twist, Spin	↻	Spanner, Wrench	Access., Corr., Dam.
Spring Toggle Bolt	Twist, Spin	↻	Screwdriver, Pliers	Access., Corr., Dam.
Turnbuckles	Twist, Spin	↻	Spanner, Wrench	Access., Corr., Dam.
Non-threaded Fasteners				
Nails	Pull	⊙	Pliers, Hammers, Hacksaw	Corrosion, Access.
Tacks	Pull	⊙	Pliers, Screwdriver	Access., Corrosion
Rivets	Destructive	⊛	Chisel, Hammer, Grinder	Difficult, Perm. Joint
Keys	Push/Pull(+Impact)	●/⊙ (+↓)	Hammer, Pliers	Accessiblity
Pins	Pull	⊙	Pliers	Access., Corrosion
Staples	Pull (+ Bending)	⊙(+↷)	Staple Pliers, Screwdriver	Access., Corrosion
Clips	Pull, Lift or Pry	⊙/▲	Manual, Pliers	Accessibility
Retaining Rings	Push, Pull	●/⊙	Ring Pliers	Accessibility
Snap-Type Fasteners	Push then Pull	! ● ⊙	Pliers, Punches	Access., Corrosion
Quick Release Fast.	Spin or Twist	↻	Screwdriver	Access., Corrosion
Integral Attachments:				
Locators				
Stop	Pull, Lift, or Pry	! ⊙/▲	Screwdriver, Pliers, Punch	Access.,Locat.,Mult.J.
Lug	Pull, Lift, or Pry	! ⊙/▲	Screwdriver, Pliers, Punch	Access.,Locat.,Mult.J.
Pin-in-hole	Pull, Lift, or Pry	! ⊙/▲	Screwdriver, Pliers, Punch	Access.,Locat.,Mult.J.
Wedge-in-slot	Pull, Lift, or Pry	! ⊙/▲	Screwdriver, Pliers, Punch	Access.,Locat.,Mult.J.
Locks				
Cantilever Hook	Push + Slide	! ● +▶	Screwdriver, Pliers, Punch	Access.,Locat.,Mult.J.
Trap	Push/Pull+Slide	! ●/⊙ +▶	Screwdriver, Pliers, Punch	Access.,Locat.,Mult.J.
Cylindrical Snap Fit	Pull + Tip	! ⊙ +▼	Screwdriver, Pliers, Punch	Access.,Locat.,Mult.J.
Ball & Socket	Pull + Tip	! ⊙ +▼	Screwdriver, Pliers, Punch	Access.,Locat.,Mult.J.
Compliant				
Cantilever Spring Fea	Pull, Slide	! ⊙/ ▶	Screwdriver, Pliers, Punch	Access.,Locat.,Mult.J.
Crush Rib Feature	Pull + Tip	! ⊙ +▼	Screwdriver, Pliers, Punch	Access.,Locat.,Mult.J.
Adhesive Bonding:				
Acrylics	Destructive	⊛	Cutters, Shears, Saw, etc.	Difficult, Perm. Joint
Cyanoacrylates	Destructive	⊛	Cutters, Shears, Saw, etc.	Difficult, Perm. Joint
Epoxies	Destructive	⊛	Cutters, Shears, Saw, etc.	Difficult, Perm. Joint
Anaerobics	Destructive	⊛	Cutters, Shears, Saw, etc.	Difficult, Perm. Joint
Silicones	Destructive	⊛	Cutters, Shears, Saw, etc.	Difficult, Perm. Joint
Polyester Hot Melt	Destructive	⊛	Cutters, Shears, Saw, etc.	Difficult, Perm. Joint
Polyurethane	Destructive	⊛	Cutters, Shears, Saw, etc.	Difficult, Perm. Joint

Table 5.1 Disassembly Motion for Different Fastening Methods (Continued)

Energy Bonding:				
Soldering	Destructive	⊛	Cutters, Shears, Saw, etc.	Difficult, Perm. Joint
Brazing	Destructive	⊛	Cutters, Shears, Saw, etc.	Difficult, Perm. Joint
Welding	Destructive	⊛	Cutters, Shears, Saw, etc.	Difficult, Perm. Joint
Others:				
Seaming	Destructive	⊛	Cutters, Shears, Hacksaw	Difficult, Perm. Joint
Crimping	Pull or Destructive	⊛	Vice, Pliers, Hammer, etc.	Easy Damage to Parts
Zippers	Slide, Pull	▶ / ⊙	Manual, Pliers	Wedging of Parts
Velcro	Pull	⊙	Manual, Pliers	Accessibility, Dirt

With the exception of rivets, discrete fasteners are usually possible to unfasten. The same is valid for integral attachments. However, there can be cases where unfastening should be possible, but through certain factors, like environmental exposure, it might become impossible. Concerning the integral attachments, a disengagement force should be applied to the lock features to actuate the deflection mechanism. To avoid unintended disengagement of lock features, it is quite important to select the disassembly force direction very carefully. The assembly direction is influenced by the basic part geometry, the severity of the service loads, and the design for assembly (and disassembly) methodologies. A visual indication of the release direction and motion can be very helpful. Especially for new products, it would be useful, if symbols for disassembly instruction would be integrated in the design. Closely related to the issue of disassembly motion is the use of tools and the accessibility to use the tools to perform the disassembly motions.

5.3 Unfastening Effort in Relation to Disassembly Motion

In Chapter 4, a U-Effort model has been introduced. Mainly, it is based on geometric fastener parameters and the condition of their use in the product. Every fastener has a specific equation usually with three to four weighting factors, mostly based on the geometry and shape of the fastener.

$$f = B_{min} + W_1 \cdot C_1 + C_2 \cdot W_2 + W_3 \cdot C_3 + W_4 \cdot C_4 = B_{min} + \sum W_i \cdot C_i \tag{4.1}$$

$$B_{min} + \sum W_i = B_{max} \tag{4.2}$$

However, as mentioned above another important factor is the disassembly motion. An estimate how the disassembly motion effects other unfastening influence factors is shown here. This equation can be extended here. A suggested equation for an unfastening index f then would be:

$$f = B_{min} + \sum W_i \cdot C_i + W_m \cdot C_m + W_e \cdot C_e + W_t \cdot C_t + W_a \cdot C_a \tag{5.1}$$

$$B_{min} + \sum W_i + W_m + W_e + W_t + W_a = B_{max} \tag{5.2}$$

C_m is a factor based on material properties, similarly C_e represents the environmental influence, C_t covers the effects of tools and C_a accessibility, and these factors are impacting the disassembly. W_m, W_e, W_t, and W_a are corresponding weight factors. The designer/disassembler has to decide how much weight each factor should have. How the C factors can be estimated, is determined in the next section.

- Material Properties

Material properties for metals might require a different unfastening motion than the ones for plastic materials. Specific properties are important for different motions, as shown in Table 5.2.

Table 5.2 Material Properties for Different Disassembly Motions

Disassembly Motion:	bend	impact	lift	pry	pull	push	slide	spin	tip	twist
Material types:										
metal										
modulus of elasticity	x	x		x	x					x
impact strength		x								x
Poisson's ratio	x				x					
coefficient of friction						x	x	x		x
specific weight / density			x			x	x	x	x	
plastics										
modulus of elasticity	x	x		x	x					x
Poisson's ratio	x				x					
coefficient of friction						x	x	x		x
strain limit	x			x						
specific weight / density			x			x	x	x	x	

Table 5.3 Factor C_m for Material Properties Depending on Disassembly Motions

Sum of x in Material Property Table (Table 5.2) [for some fasteners motion combinations are possible]	C_m
1x	0.2
2x	0.4
3x	0.6
4x	0.8
5x	1.0

To obtain a C_m value, the disassembly motion or a combination of motions has to be determined first. Then, in Table 5.2, under the specific motion the marked fields have to be counted for metal or plastic, respectively. The C_m value can be obtained with the number of marks from Table 5.3.

♦ End-of-Life Condition / Environmental Exposure

An estimate how much environmental exposure increases the U-Effort for the different disassembly motions is shown in Figure 5.11. The unfastening effort to unfasten a relatively new screw is less than that of unfastening a corroded one. That means, disassembly motions are hindered in some way or another, if the fastening element shows effects of end-of-life conditions.

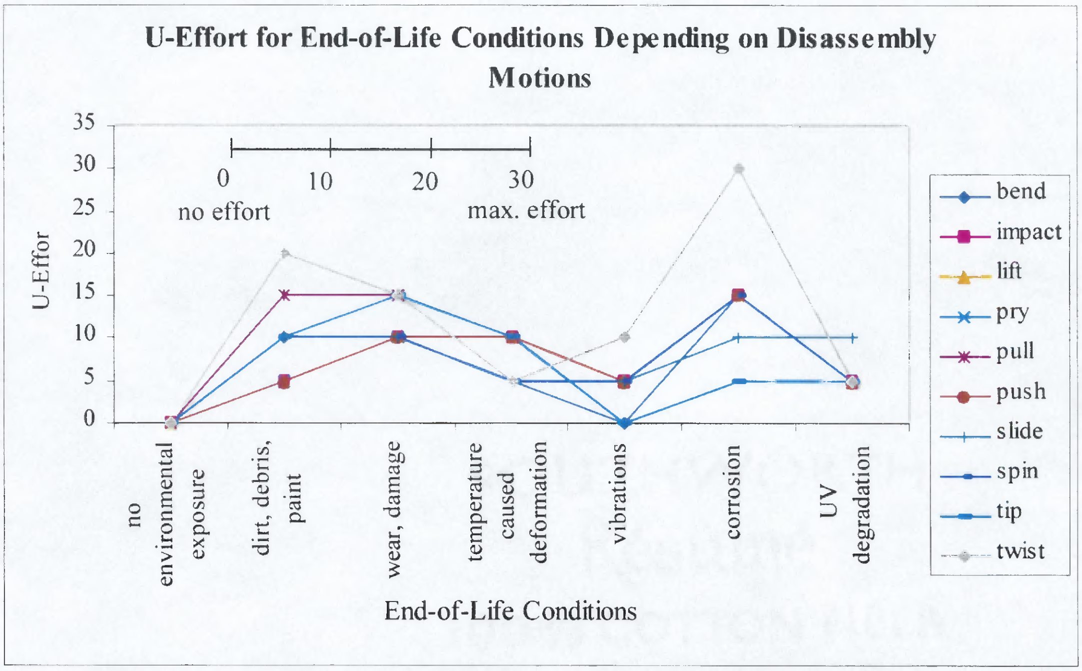


Figure 5.11 U-Effort of Different Disassembly Motions for End-of-Life Conditions

Similarly to the material factor, a C_e value has to be obtained. For that, a disassembly motion or a combination of motions has to be determined. In Figure 5.11 or in Table 5.4, respectively, under the specific motion the scores have to be counted and summed up. From Table 5.5, the C_e value can then be obtained.

Table 5.4 Scores for Disassembly Motions for Different Cases of End-of-Life Conditions

	bend	impact	lift	pry	pull	push	slide	spin	tip	twist
no environmental exposure	0	0	0	0	0	0	0	0	0	0
dirt, debris, paint	10	10	5	10	15	5	10	10	10	20
wear, damage	10	10	10	10	15	10	10	10	15	15
temperature caused deformation	5	5	10	5	10	10	5	5	10	5
vibrations	0	0	5	5	5	5	5	5	0	10
corrosion	15	10	15	10	15	15	10	15	5	30
UV degradation	5	0	5	5	5	5	10	5	5	5

Table 5.5 Factor C_e for Environmental Exposure Depending on Disassembly Motions

Sum of Score for Environmentally Influenced Disassembly Motions (Table 5.4 / Figure 5.11)	C_e
0	0.0
$\Sigma \leq 30$	0.3
$30 < \Sigma \leq 50$	0.5
$50 < \Sigma \leq 70$	0.8
$\Sigma > 70$	1.0

- Tools

The use of every tool requires a specific motion, and therefore, influences the unfastening effort. The effort increases extremely, if OEM or special tools have to be used. Not all tools can be used for unfastening, some can only be used for destructive disassembly, and some for either one, see Table 5.6. A value for C_t can be obtained from Table 5.7.

Table 5.6 Tools for Unfastening and Destructive Disassembly and their Activities (Motions) and Forces

Tools	Activity	Resulting Force	U = Unfastening, D=Destr.Disassembly
Hammer	hammering	impact force, compression, bending	U/D
Screwdriver	turning	twisting moment / torque	U
Pliers	twisting, turning, pulling	twisting moment / torque, bending moment	U/D
Allen key	turning, twisting	torque	U
Hacksaw	sawing, ripping, cross-cutting	shear	D
Chisel	chipping, cutting	shear	D
Grinder	grinding, cutting	shear	D
Punch	punching, cutting, blanking	shear	U/D
Ratchet	turning, twisting	torque	U
Spanner	turning, twisting	torque	U
Wrench	turning, twisting, pulling	torque, bending moment	U/D
Blow torch	melting, cutting (torch-cutting)		D
Cutter	cutting	shear	D
Shear	cutting	shear	D
Vice*	holding	compression	Fixture

Table 5.7 Factor C_t for Tools

Unfastening / Disassembly Tool	Ct
manual / no tool	0.0
simple / standard tool	0.5
OEM tool	0.9
Special tool	1.0

♦ Accessibility

The accessibility basically depends on which tool is used to unfasten the product and how large the work envelope of that tool is, which is dependent on the disassembly motion, too. Depending on the fastener, an unfastening tool has to be selected, which determines the unfastening effort based on accessibility, and then, a value for C_a can be obtained from Table 5.8.

Table 5.8 Factor C_a for Accessibility

Accessibility	C_a
No Tools Required	0.0
Screwdriver	0.2
Allen key	0.4
Pliers	0.6
Hammer	0.8
Ratchet / Spanner / Wrench	1.0

Adding all the effort values up according to the Equation (5.1), the overall unfastening effort can be determined. To show how this extended model can be applied, the example of Chapter 4, the disassembly of a Walkman, will be extended.

Then Equation (4.4) becomes

$$\begin{aligned} f_{cant} &= 20 + W_1 \cdot C_1 + W_2 \cdot C_2 + W_3 \cdot C_3 + W_m \cdot C_m + W_e \cdot C_e + W_t \cdot C_t + W_a \cdot C_a \\ &= 20 + 10 \cdot C_1 + 10 \cdot C_2 + 10 \cdot C_3 + 15 \cdot C_m + 5 \cdot C_e + 5 \cdot C_t + 5 \cdot C_a \end{aligned} \tag{5.3}$$

and Equation (4.10) becomes

$$\begin{aligned} f_{screw} &= 20 + W_1 \cdot C_1 + W_2 \cdot C_2 + W_3 \cdot C_3 + W_4 \cdot C_4 + W_m \cdot C_m + W_e \cdot C_e + W_t \cdot C_t + W_a \cdot C_a \\ &= 20 + 10 \cdot C_1 + 10 \cdot C_2 + 10 \cdot C_3 + 6 \cdot C_4 + 10 \cdot C_m + 10 \cdot C_e + 7 \cdot C_t + 7 \cdot C_a \end{aligned} \quad (5.4)$$

Table 5.9 Extended Example for U-Effort Model for a Walkman

Fast. ID	Type	B_{min}	B_{max}	W_1C_1	W_2C_2	W_3C_3	W_4C_4	W_mC_m	W_eC_e	W_tC_t	W_aC_a	Score f	#	Sum
1	cant.	20	80	10	5	1		15	0	1	0	52.0	1	52.0
2	cant.	20	80	5	5	0		15	0	1	0	46.0	1	46.0
3	screw	20	90	3	1	2	0	8	3	3.5	1.4	41.9	4	167.6
4	screw	20	90	3	0	2	0	8	3	3.5	1.4	40.9	1	40.9
5	screw	20	90	3	0	0	0	8	3	3.5	1.4	38.9	2	77.8
6	screw	20	90	3	0	2	0	8	3	3.5	1.4	40.9	2	81.8
													SUM:	466.1

For this case, shown in Table 5.9, the overall unfastening score is 466.1. That means this is the necessary effort to unfasten the Walkman. The U-Effort score increases when the influence through material, environmental exposure, tools, and accessibility is included (compared to the result in Chapter 4, where the overall U-Effort score is 355). Again, this value could be transferred to costs if the values for the different components and/or materials are known. Similarly, a comparing study with other brands could be done for the extended model, too. To improve the design for the ease of unfastening, changes in the design should result in a lowering of the U-Effort value. This model gives just an estimate about the unfastening effort, but it can be used as a guideline to figure out what changes would make the unfastening easier.

CHAPTER 6

THE U-FORCE MODEL FOR SNAP FITS

So far, the emphasis has been on the factors, which influence the unfastening process of commonly used fasteners and on how much effort is needed to unfasten them. In this chapter, a further step will be taken. The question how integral attachments behave during unfastening or disassembly in general is of concern here. The objective is to create a model for the design of integral attachments suitable for unfastening, the U-Force model. Therefore, for two fastening elements a model for the design process for ease of unfastening will be introduced, which is based on obtaining the unfastening or removal forces. The cantilever snap fit and cylindrical or annular snap fit have been selected, because they are the most known integral attachments fastening elements.

First, integral attachments are introduced, classified and issues for their design are discussed. Then a model for the design of each of these snap fits is introduced. In the design procedure the emphasis lies on unfastening in order to provide unfastening suitability. Further, the design process for integral attachments is usually an iterative one. To simplify this procedure, a parameter study has been performed.

6.1 Introduction to Integral Attachments

Integral attachments are features belonging to the parts or components. This means that an assembly without separate fasteners is possible and often during assembly, a multiple joining takes place, so that several integral features, for example snap fits, are joined together simultaneously. The essential attributes of an integral attachment feature are that it is integral to a part and that its primary purpose is to provide some attachment functionality. No extra material, fasteners, or external energy sources are needed. Integral attachment features have several functions [Gabriele, et. al., 1995; Luscher, et. al., 1995; Luscher, et. al., 1995]:

- to provide attachment between parts
- to establish part location, alignment, and orientation
- to transfer service loads
- to eliminate degrees of freedom
- to absorb tolerance between parts

Integral attachments are becoming increasingly popular because the number of parts is reduced, thereby reducing assembly time, and usually fewer tools are required for their assembly. The complexity and the costs of assembling structures using integral attachments can be simplified and reduced. In the past, integral attachments were used in less critical areas but recently even in areas of critical stresses. The number of integral attachments in product design is increasing. Snap-fit-type integral attachments have become quite popular. They not only make assembly easier; but also provide a finished and attractive look to the consumer. Integral attachments may also reduce the risk of loose fasteners floating free inside products and they become more cost attractive as the product volume increases. They can be applied to any combination of materials.

The unfastening aspect has been ignored mostly in their design. Disassembly is an important issue for the recycling of products, which is of increased concern for companies and customers. Because the number of products with integral attachments has increased so much, the unfastening process needs to be considered in their design. However, the use of integral attachments is quite new and a scientific basis is still being developed. Especially for the unfastening of integral attachments there is almost no information available in the literature yet. With the development of the U-Force model, a tool will be provided, which should implement unfastening suitability into the design process.

6.2 Attributes for the Unfastening of Integral Attachments

6.2.1 Classification of Integral Attachments

Integral attachment features can be divided into three groups: locators, locks, and compliants. Locators give location of the parts relative to each other. This includes a removal of the degree of freedom and the transfer of service loads. In order for two parts to remain together as an assembly, their relative location, alignment, and orientation must be fixed at all time. Locators ensure relative location by having surface contact from both parts, thereby eliminating the degrees of freedom normal to these surfaces [Luscher, et. al, 1998]. Some examples of locators are shown in Figure 6.1; others are ribs and bosses.

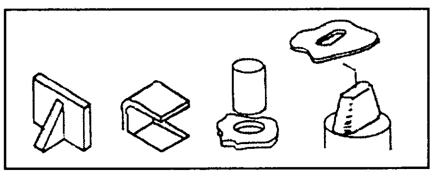


Figure 6.1 Locators (Stop, Lug, Pin-in-Hole, Wedge-in-Slot)

Compliant features absorb tolerance stack-up, misalignments, and manufacturing variability between parts through the built-in compliance or flexibility of these features [Luscher, et. al, 1998]. Variability for plastic parts often means warpage, which can be a great concern. Tolerance stack-up can cause a gap between the parts resulting in rattle or looseness or it can cause unintended interference resulting in high stresses in the part and high assembly forces. The compliant feature eliminates any gap or interference through their flexibility or compliance. There are two types of compliants. First, there are elastic compliants, which are designed so that they provide a minimum amount of preload to balance the tolerance stack-up. On the other side, there are inelastic or plastic compliant features, which are designed to permanently deform during initial assembly to eliminate any gaps. Typical examples of compliants are the cantilever spring feature which functions elastically with a preload and the crush rib feature which works inelastically and deforms permanently when a metal shaft is inserted into the cylinder, shown in Figure 6.2. Other compliant features are guides, darts, tapered features, limiters, and assists.

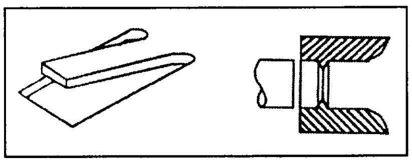


Figure 6.2 Compliants (Cantilever Spring Feature, Crush Rib Feature)

Locks provide the final locking together of the parts during assembly through their elastic deflection and recovery. They present the most common known integral attachments. Usually they have a structure, which elastically deforms during engagement of the lock and a structure, which contains an offset to entrap the parts after engagement. These two necessary structures are also called the deflection mechanism and the retention mechanism.

The deflection mechanism is the part that provides the elastic deflection needed for the engagement of the lock. The deflection can be bending or torsional or axial elongation. The retention mechanism is the part, which provides the latch to insure retention. Usually a catch is used as retention mechanism. A catch has two planar faces and it is molded unto the end of the deflection mechanism. Lock features should not take any load in the direction needed to deflect them. Typical locks are cantilever hooks, traps, cylindrical snaps, ball-and-socket features, see Figure 6.3, and bayonet fingers and compressive beams. Cantilever hooks are widely used in the plastic part design.

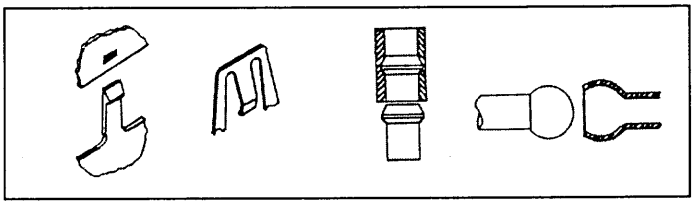


Figure 6.3 Locks (Cantilever Hook, Trap, Cylindrical Snap, Ball & Socket Feature)

Hook-typed integral attachments are bayonet-and-finger, cantilever-hole fastener, cantilever hook, compressive hook, L-shaped hook, and U-shaped hook [Oh, et. al, 1999], shown in Figure 6.4 and 6.5. For example, U-shaped and L-shaped cantilevers are often used when no configuration with a strain value below the allowable value can be found with the standard shape.

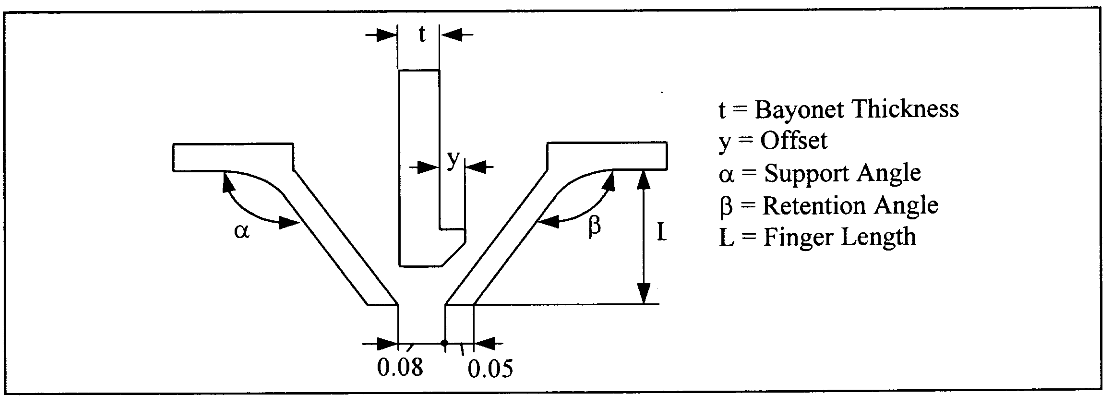


Figure 6.4 Bayonet-and-Finger

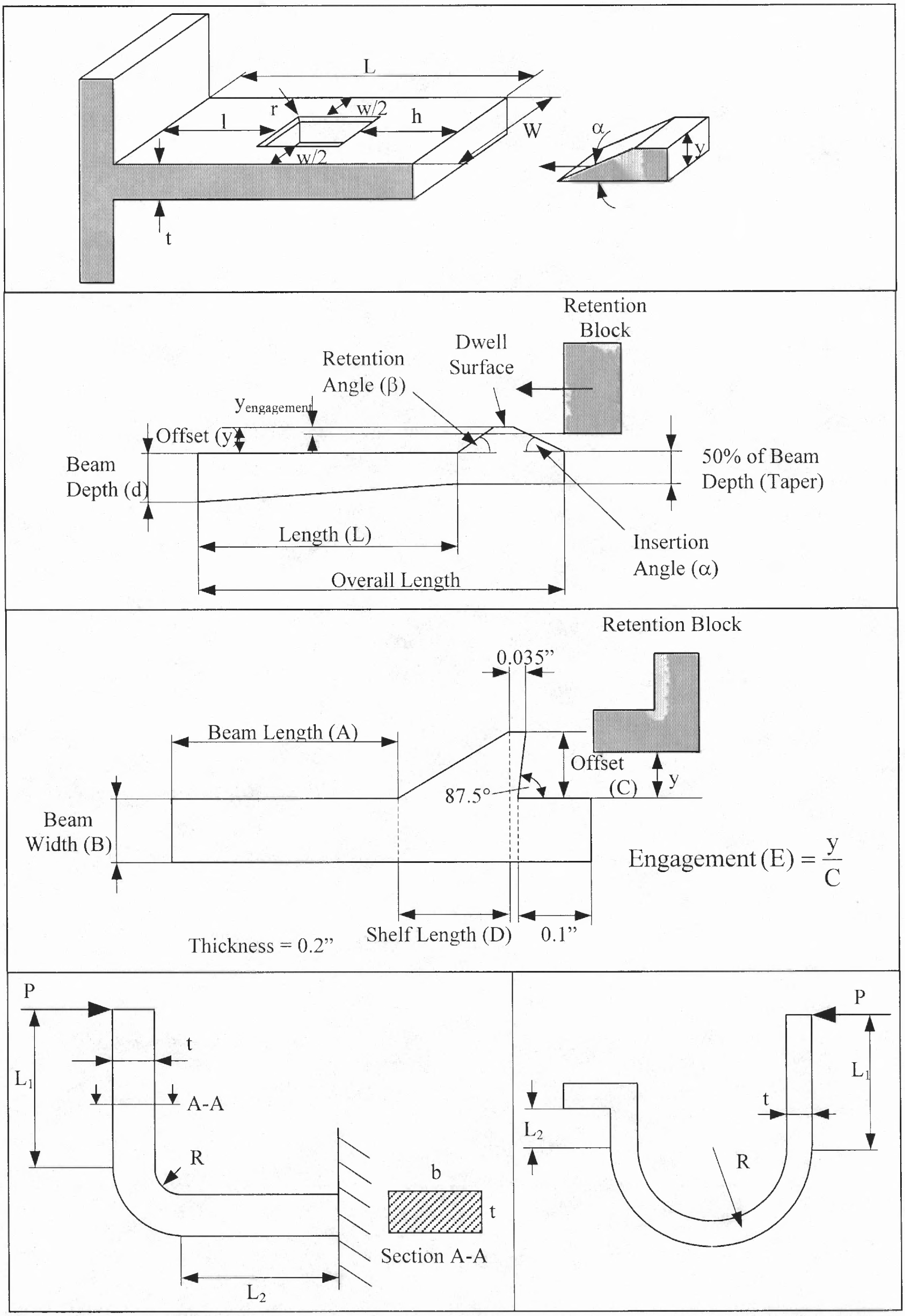


Figure 6.5 Cantilever-Hole Fastener, Cantilever Hook, Compressive Hook, L-Shaped Hook, and U-Shaped Hook

Another group of integral attachments is cylindrical snap fits. Interference fits are usually used for the joining of metals, where two parts can be assembled by press-fitting them together. However, this is more critical for thermoplastics. They can be and are used in many applications. However, the designer must consider creep or stress relaxation and consider a large reduction of the initial clamping force. Therefore, for plastic materials, cylindrical snap fits are preferred, see Figure 6.6.

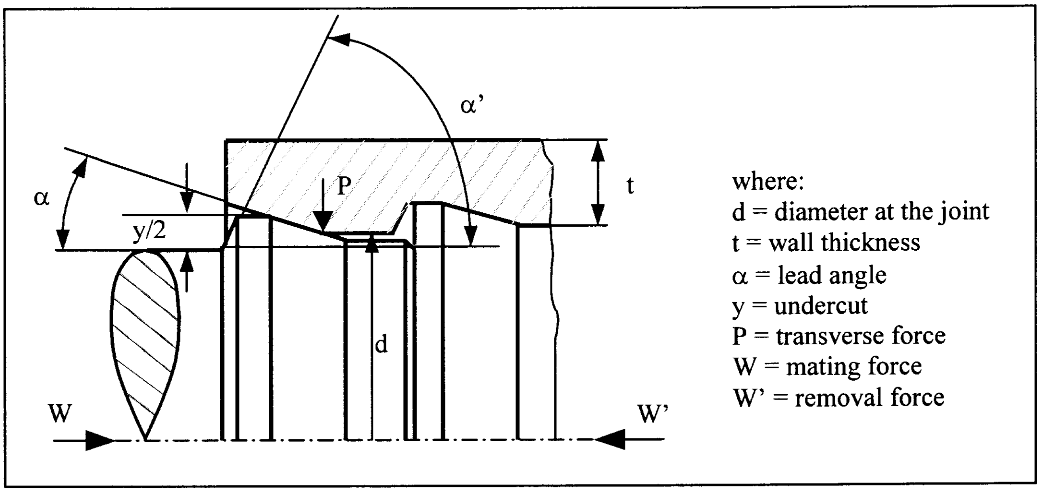


Figure 6.6 Cylindrical Snap Fit

A cylindrical snap or snap fit belongs to the group of locks in the integral attachments. They are also called annular snap fits based on their shape and they belong to the group of transition fits. However, they do not have smooth cylindrical surfaces, but recesses or grooves. They are useful when two parts with circular geometry have to be connected. Common applications are medicine bottles, but also for example for gelatin capsules the snap fit principle is used [Pelco International, 1997]

Cylindrical snap fits provide both the location and locking function. The outer round surface of the snap and the inner surface of the boss give the location, and the locking is supplied by the molded-in catch surfaces [Luscher, 1995]. Cylindrical snap fits can have an elastic shaft and a rigid tube or boss or the shaft can be rigid and the tube is elastic, or both are elastic. This is important in order to provide a deflection mechanism. A retention mechanism is then given by the undercut. The undercut is necessary for the engagement of the snap and ensures the locking of the parts. In addition, depending on the return or retention angle the parts can be disengaged under retention force. In a cylindrical snap fit, the degree of motion is decreased by removing two rotations and two translations.

6.2.2 Material Concerns for Integral Attachments

Mostly, integral attachments are made from thermoplastic materials because of their flexibility, resilience, dynamic strain, and low coefficient of friction, but with sufficient strength and rigidity. However, because integral attachments are usually manufactured through injection molding a certain production size is necessary to outweigh the costs for the tooling. Snap fits are very economical and efficient in mass production, but for a small number of units, the tooling costs can be very high. One of the advantages of injection molding is that features such as ribs, posts, and springs, which add functionality, can be readily molded into the part at little or no increase of the costs. By considering the parting line of the mold, integral attachment features can be inexpensively added to the mold. In all snap fit designs, some portion of the molded part must flex like a spring, usually past a designed-in interference, and quickly return, or nearly return, to its unflexed position to create an assembly between two or more parts. For a successful snap fit design, it is important to have sufficient holding power without exceeding the elastic or fatigue limits of the material. By nature, integral attachments have an insertion direction that causes the elastic deformation of the snap feature, followed by the elastic recovery and entrapment of the two parts. This leads to assembly in which the parts are brought together and secured in a simple linear motion versus the more complex helical insertion motion for threaded fasteners [Gabriele, et. al., 1995; Hoechst Celanese, 1991].

Using the beam equation, the maximum stress during assembly can be calculated. If it stays below the yield point of the material, the flexing finger returns to its original position. However, certain designs have not enough holding power due to low forces or small deflections. With many plastic materials, the calculated bending stress can far exceed the yield point stress if the assembly occurs rapidly. In other words, the flexing finger just momentarily passes through its maximum deflection or strain, and the material does not respond as if the yield stress has been greatly exceeded. Thus, a common way to evaluate snap fits is by calculating strain rather than stress. As mentioned before, it could be economical to have parts made from the same material. For recycling they do not have to be disassembled, which saves time and costs. With integral attachments, this becomes much easier.

Adequate mechanical properties are a prerequisite in most applications of plastics. When most plastics are subjected to a load, the relationship is non-linear. For design purposes, plastics are often treated as linearly elastic, homogeneous, isotropic materials. For example, for integral attachments, it is important to know the maximum permissible strain, because during insertion (and retention) the values often get close to the limits. In order to avoid breakage problems during assembly, especially when the assembly will be automated, strain limits have to be considered in the design phase. How brittle or how flexible a part is, depends on material properties like the modulus of elasticity. That value itself can change for a material for different temperatures. The higher the modulus of elasticity the more a part can be stressed without damage. Another important material parameter is the coefficient of friction. This material dependent parameter has great influence on how high the unfastening force for a snap fit can be.

6.2.3 End-of-Life Condition Regarding Integral Attachments

Environmental influence is the change through environmental exposure. For plastic materials, UV degradation and thermal deformation can cause trouble in the unfastening process. If a product is exposed for a longer period of time to sunlight, some material properties might change. It is important to know the UV resistance of the used plastic material and if necessary to apply appropriate shielding. Because the fastening elements are integrated, there is no danger of loosing parts. Snap fits have less problems with creep and stress relaxation than plastic press fits, a decrease in the holding force due to relaxation of stress does not occur. But sometimes there can be rattling and squeaking caused by vibration, which might influence the unfastening process.

6.2.4 Tools for Unfastening of Integral Attachments

Cantilever snap fits often eliminate tools required for assembly and disassembly, e.g., a battery compartment of a remote control or calculator. However, especially when multiple joining is given, the disassembly might be very difficult then and tools like screwdrivers are necessary to separate the components.

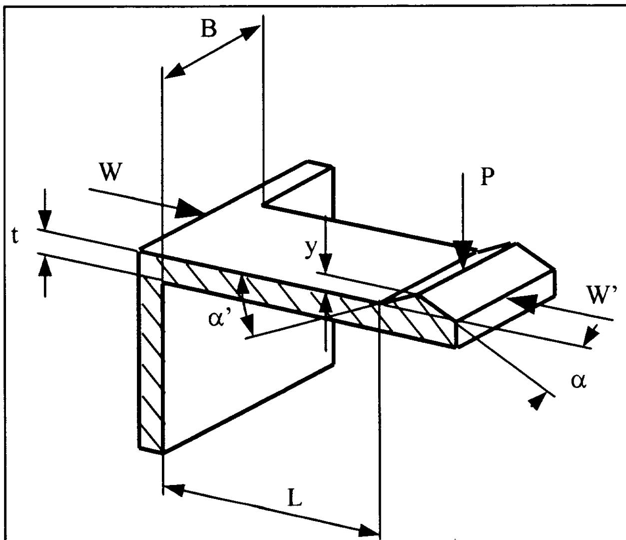
6.2.5 Accessibility for Integral Attachments

For single snap fits, the disassembly is often quite easy and it is no problem to access the snap fit. However, in some cases the joining places cannot even be seen from the outside (no visibility of attachment location) – one benefit of integral attachments is an attractive look for the consumer, but this can be a disadvantage for the disassembler. Here, the unfastening can be very difficult, because the disassembler does not know where to access and where to apply forces. Visuals are one way to overcome this problem.

6.3 Unfastening of Cantilever Snap Fits

6.3.1 Basis for Cantilever Snap Fits

The design of a cantilever snap fit is an iterative process. Sometimes it is necessary to change the geometric parameters several times before a snap fit with a strain below the permissible strain of the material can be obtained. The parameters of a cantilever snap fit are shown in Figure 6.7.



L	=	length of beam
B	=	width of beam
t	=	beam thickness
α	=	lead angle
α'	=	return angle
y	=	deflection
P	=	mating or perpendicular force
W	=	push-on force
W'	=	pull-out force

Figure 6.7 Cantilever Snap Fit

Cantilever hooks can have a uniform or a tapered cross-section. The deflection mechanism can be varied by using either a straight or a tapered beam. Beams can be tapered by different amounts in either depth or width or both dimensions at once. In most cases, a uniform cross-section would be sufficient. A tapered section beam is desirable if additional deflection is desired. It provides a uniform stress distribution and aids in part release during molding.

To unfasten a cantilever snap fit usually means to apply a force in a specific direction, and through the force, the parts will be disengaged. If the snap fit has to be removed without any permanent deflection (plastic deformation), the retention or pull-off force must be below a maximum value, the elastic strain limit, but high enough to retain engagement under normal service load. The push-on or assembly force is defined as

$$W = P \cdot \frac{\mu + \tan\alpha}{1 - \mu \tan\alpha} \tag{6.1}$$

Similar to the push-on or assembly force, the pullout or pull-off force is defined by

$$W' = P \cdot \frac{\mu + \tan\alpha'}{1 - \mu \tan\alpha'} \tag{6.2}$$

P is the mating or perpendicular force. α is the insertion angle, α' is the return or retention angle and μ is the coefficient of friction (values for μ see Appendix A). In some cases, there might also be another way to unfasten a cantilever snap fit, through applying a force in direction of P (perpendicular force).

$$P = \frac{B \cdot t^2 \cdot E \cdot \varepsilon}{6 \cdot L \cdot Q} \tag{6.3}$$

B is the beam width, L is the beam length, and t is the beam thickness. E is the flexural modulus and ε is the strain. Q is the deflection magnification factor (more information to

Q see Appendix B). Assuming, there is access to apply a force in direction P, the question is what takes less effort to unfasten the cantilever snap fit, P or W'. In order to find an answer to this question, different factors are looked at in more detail. First, the critical angle is defined as

$$\alpha'_{crit} = \arctan\left(\frac{1}{\mu}\right) \tag{6.4}$$

W' cannot be applied without damaging the parts if the retention angle is greater than the critical angle, e.g. for a coefficient of friction of 0.5 this would be 63.43°. The next question is at what retention angle the unfastening is easier with P instead of W'. It is assumed that unfastening from direction P is possible, which might not be the case every time. P and W' would need an equal unfastening effort, if:

$$\frac{\mu + \tan\alpha'}{1 - \mu\tan\alpha'} = 1 \tag{6.5}$$

From this equation the following result can be obtained:

$$\alpha'_{P-W'} = \arctan\left(\frac{1-\mu}{1+\mu}\right) \tag{6.6}$$

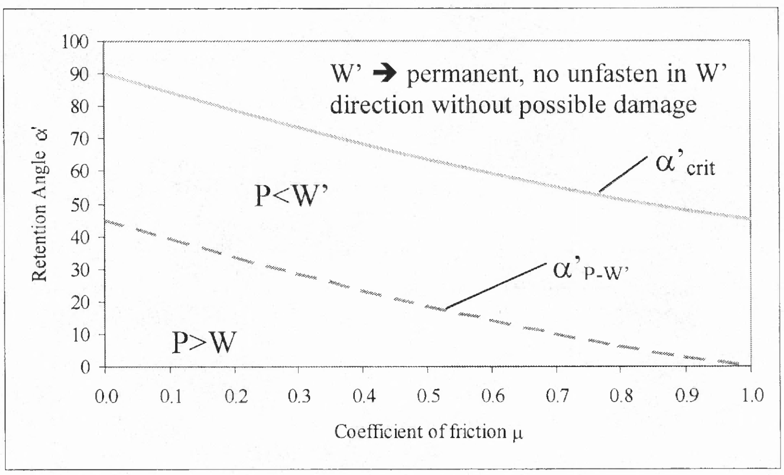


Figure 6.8 Retention Angle α' versus Coefficient of Friction μ

For example, for $\mu = 0.5$ it would mean that the retention angle $\alpha'_{P\text{-}W'}$ is 18.43°. If $\alpha'>$ 18.43°, then P needs less unfastening force than W'. The relationship between the retention angle and the coefficient of friction is shown in Figure 6.8.

6.3.2 Design Procedure of U-Force Model

The unfastening process for cantilever snap fits is shown as a flowchart (U-Force Model) in Figure 6.9. It can be applied, when a designer wants to ensure that a cantilever snap fit design provides unfastening suitability, that means in the process of a new design and in the evaluation of exiting ones. The design procedure is then as follows:

Step 1: Determine the material of the parts and input values for modulus of elasticity E, maximum strain ε_0, and coefficient of friction μ.

Step 2: Check assumptions. For the calculations, it is assumed that are no effects of environmental exposure, that accessibility is given and that for the unfastening process no tools are needed. If that is not the case, then destructive disassembly might be necessary, and the equations cannot be applied.

Step 3: Input geometric parameters. Often not all values are known, therefore, appropriate assumptions have to be made. The design process is usually an iterative one that means it can be necessary to apply the model more than once. The required geometric parameters are the beam length L, the beam width B, the beam thickness t, the insertion angle α, and the retention angle α'.

Step 4: Decide beam configuration. Based on Figure B.1 in Appendix B, with the ratio of beam length to beam thickness and the position of the cantilever to the part (beam configuration) the deflection magnification factor Q can be determined.

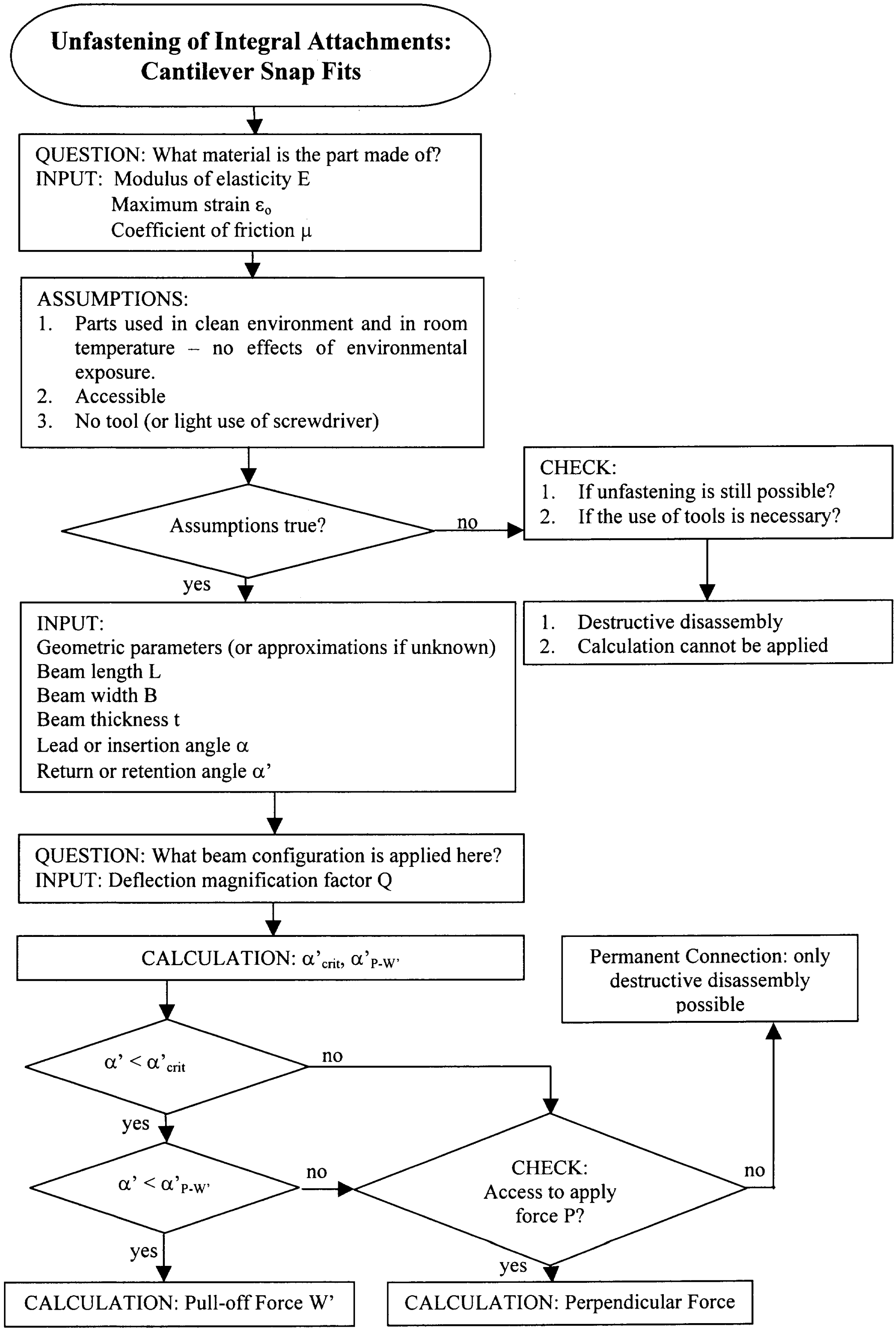


Figure 6.9 U-Force Model for Cantilever Snap Fit

Step 5: With Equation 6.4, determine the critical angle. If the retention angle is smaller than the critical angle, then proceed to Step 8. If the retention is larger, it should be continued with Step 6.

Step 6: Check if access is given to apply a perpendicular force P. If yes, than go to Step 7. If not, then two things can happen. If the retention angle was smaller than the critical angle, the cantilever snap fit still can be unfastened with the pull-off force W' (go to Step 9), but the required force to unfasten could be very high. That also means the holding force is quite high. In the other case, if the retention angle is bigger than the critical angle, unfastening of the cantilever snap fit will not be possible, that means the connection will be permanent, or destructive disassembly has to be used.

Step 7: With Equation 6.3, calculate the perpendicular force P as the unfastening or removal force.

Step 8: With Equation 6.6, the angle $\alpha'_{P\text{-}W'}$ is to determine and the value has to be compared with the retention angle α'. If the retention angle is smaller, proceed to Step 9. If the retention angle is larger, then proceed with Step 6.

Step 9: Determine the pull-off force W' according to Equation 6.2 as unfastening or removal force.

6.3.3 Example for Application of U-Force Model for Cantilever Snap Fit

Suppose a cantilever snap fit of ABS material is chosen with the dimensions as shown in Figure 6.10. The cantilever is on the edge of the molded part continuing in the same plane. It is assumed that the product is used in a clean environment, that the snap fit is accessible and that it can be unfastened without the use of tools. The design is to be evaluated and it is to determine if the design is unfastening suitable.

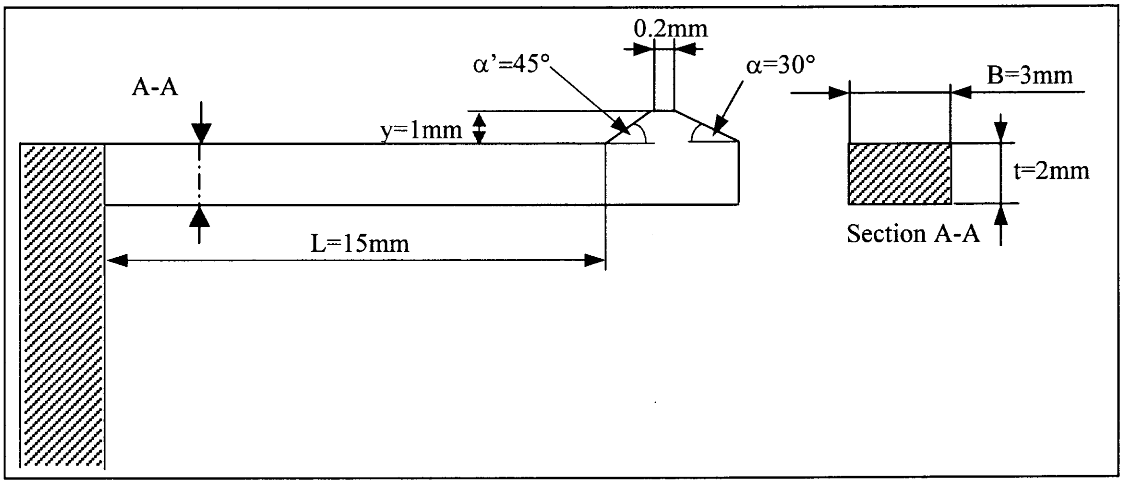


Figure 6.10 Application Example for Cantilever Snap Fit

1. The material is ABS, that means E =2100N/mm^2, $\varepsilon_o = 0.06$, and $\mu = 0.5$

2. Assumptions are true.

3. The geometric parameters are L = 15mm, B = 3mm, t = 2mm, $\alpha = 30°$, and $\alpha' = 45°$.

4. From curve 4 with L/t = 7.5 it follows that Q = 1.6 (Appendix B).

5. The critical angle is $\alpha'_{crit} = 63.43°$ (Eq. 6.6). The retention angle is 45°, and therefore smaller.

8. The unfastening or removal force is then W' = 31.5N (Eq. 6.2).

Result: Under the given circumstances, the design is suitable for unfastening.

Depending on the requirements for the cantilever, different paths can be followed. In order to find suitable design parameters, in section 6.5 the relevant parameters are analyzed. But first, the U-Force model for the cylindrical snap fit will be looked at.

6.4 Unfastening of Cylindrical Snap Fits

6.4.1 Basis for Cylindrical Snap Fits

The general geometric parameters of a cylindrical snap fit are shown in Figure 6.11. Similar to the cantilever snap fit, to unfasten a cylindrical snap fit usually means to apply a force in a specific direction and through this force the parts are disengaged. If the snap fit has to be removed without any permanent deflection, the retention or pull-off force must be below a maximum value, the elastic strain limit, but high enough to retain engagement under an average service load.

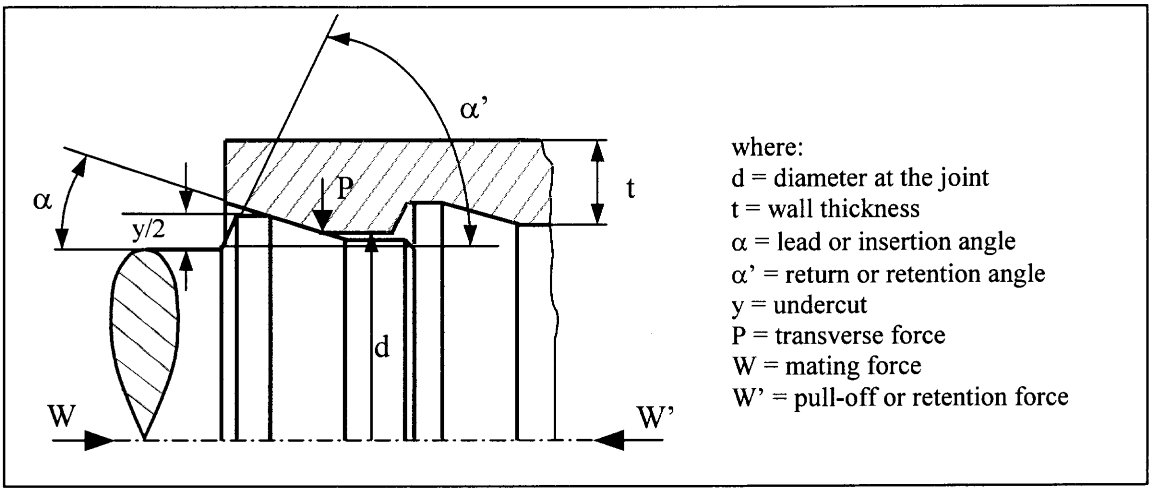


Figure 6.11 Geometric Parameters of a Cylindrical Snap Fit

The calculation of the mating force P is a little bit more difficult for cylindrical snap fits than for cantilever snap fits. This is because the snap-fitting bead on the shaft expends a relatively large portion of the tube. Accordingly, the stress is also distributed over a large area of the material surrounding the bead [Rensselaer Polytechnic Institute, 1995].

According to RPI, two cases can be distinguished, the force is applied at the end of the beam or the force is applied a long distance from the end of the beam. Considering the first case, where the force is applied at the end of the beam, and, if the shaft is rigid and the outer tube (hub) elastic, a geometric factor X_N has to be calculated,

$$X_N = 0.62 \frac{\sqrt{\left(\frac{d_o}{d} - 1\right) \Big/ \left(\frac{d_o}{d} + 1\right)}}{\left[\left(\frac{d_o}{d}\right)^2 + 1\right] \Big/ \left[\left(\frac{d_o}{d}\right)^2 - 1\right] + \nu} \tag{6.7}$$

where d is the diameter at the joint, d_o is the external diameter of the tube, and ν is Poisson's ratio. However, if the tube is rigid and the hollow shaft is elastic, then the geometric factor X_W is

$$X_W = 0.62 \frac{\sqrt{\left(\frac{d}{d_i} - 1\right) \Big/ \left(\frac{d}{d_i} + 1\right)}}{\left[\left(\frac{d}{d_i}\right)^2 + 1\right] \Big/ \left[\left(\frac{d}{d_i}\right)^2 - 1\right] - \nu} \tag{6.8}$$

where d_i is the internal diameter of the hollow shaft. Now, the transverse force P can be calculated, for the case of a rigid shaft and elastic tube,

$$P = y \cdot d \cdot E_s \cdot X_N \tag{6.9}$$

or for a rigid tube and elastic shaft

$$P = y \cdot d \cdot E_s \cdot X_W \tag{6.10}$$

respectively. y is the undercut. E_s is the secant modulus. Usually it is quite difficult to get material data for the secant modulus. Therefore, as an approximation for the case of room temperature the Young's modulus can be used instead. The mating force, W, is then,

$$W = P \frac{\mu + \tan \alpha}{1 - \mu \tan \alpha} \tag{6.11}$$

where μ is the coefficient of friction and α is the lead angle. The pull-off force, W', is very similar determined,

$$W' = P\frac{\mu + \tan\alpha'}{1 - \mu\tan\alpha'} \tag{6.12}$$

Here, α' is the return or retention angle.

In the second case, the force P is applied at a long distance from the end of the beam. The cylindrical snap fit is considered remote or at a long distance, if the distance, δ_{min}, from the end of the tube is at least

$$\delta_{min} \approx 1.8 \cdot \sqrt{d \cdot t} \tag{6.13}$$

where d is the joint diameter and t the wall thickness. If that is the case, the forces are theoretically four times greater as if the joint is located at the end of the beam or tube [Rensselaer Polytechnic Institute, 1995]. However, based on tests, the actual mating forces rarely exceed factor 3.

$$P_{remote} \approx 3\,P_{near} \tag{6.14}$$

$$W_{remote} \approx 3\,W_{near} \tag{6.15}$$

This means that if a joint lies between zero and the minimum values for δ, the factor is somewhere between one and three.

Unfastening of cylindrical snap fits can be difficult, especially if the joint is damaged or if it has changed due to environmental influence, e.g. UV degradation of plastic or long usage at high temperature. Through deformation, it might be jammed, and therefore, quite difficult to pull apart the two mating parts. Generally, the goal is to design a cylindrical snap fit so that the releasing force is greater than the insertion force, in order to ensure that it will snap in easily and provide a secure fit.

$$\max(W') \tag{6.16}$$

The retention force has to be greater than the insertion force, but still in such a way, that easy disassembly is possible.

$$W < W' \tag{6.17}$$

This has to be accomplished while maintaining all constraints and preventing failure due to permanent plastic deformation. Material properties as permissible stresses and strains are not to be exceeded.

$$\varepsilon \leq \varepsilon_{pm} \tag{6.18}$$

If α' is close or equal to 90°, the connection becomes inseparable or permanent. In order to get a removable design; the retention angle has to be smaller than the critical angle.

$$\alpha' < \alpha'_{crit} \tag{6.19}$$

A graph of the relationship between insertion and retention angle is given in Figure 6.12. Different materials are considered through the coefficient of friction μ, which is used to calculate the critical angle α_{crit}. Consequently, if insertion and retention angles are selected according to Figure 6.12, the cylindrical snap fit design should result in a feasible solution.

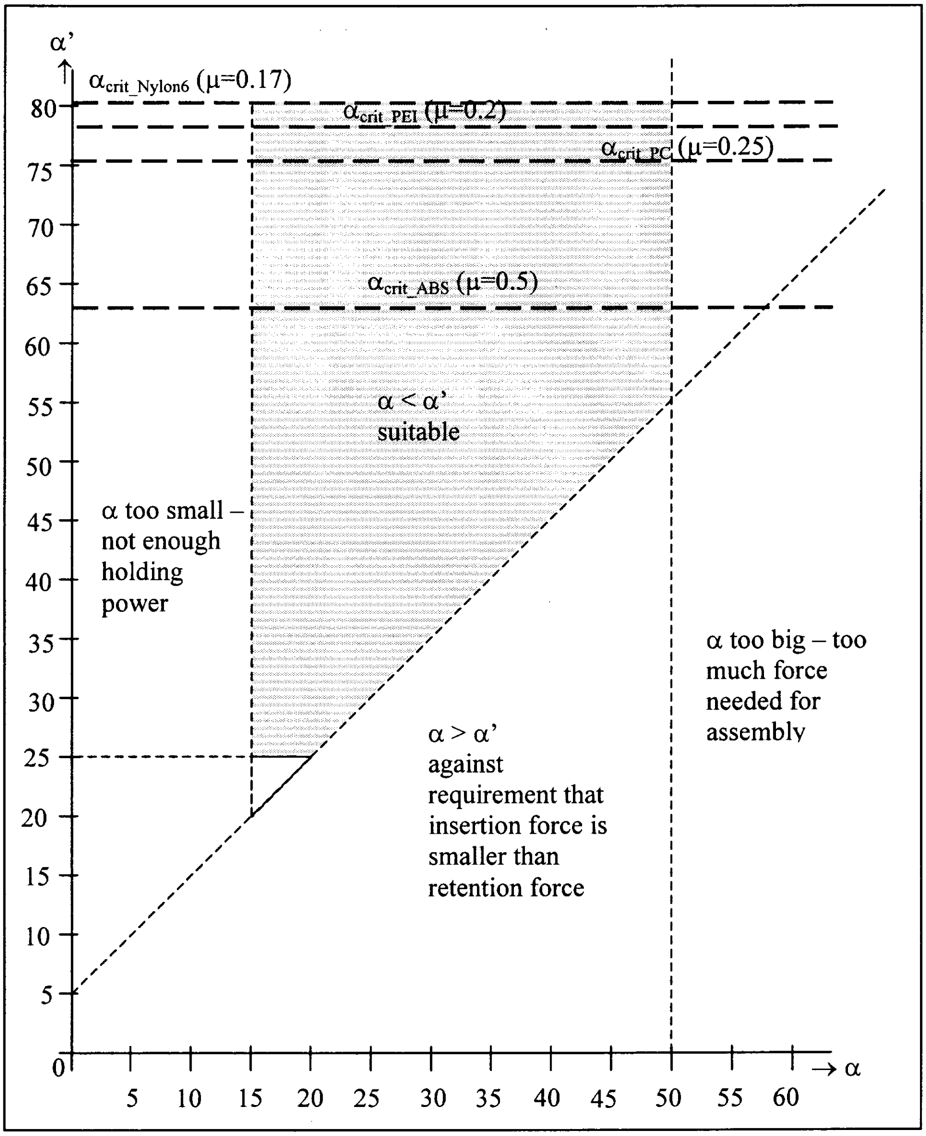


Figure 6.12 Array of Suitable Values for Insertion and Retention Angle

6.4.2 Design Procedure of U-Force Model

The unfastening process for cylindrical snap fits is shown as a flowchart (U-Force Model) in Figure 6.13. Again, it can be applied, when a designer wants to ensure that a cantilever snap fit design provides unfastening suitability. The design procedure is then as follows:

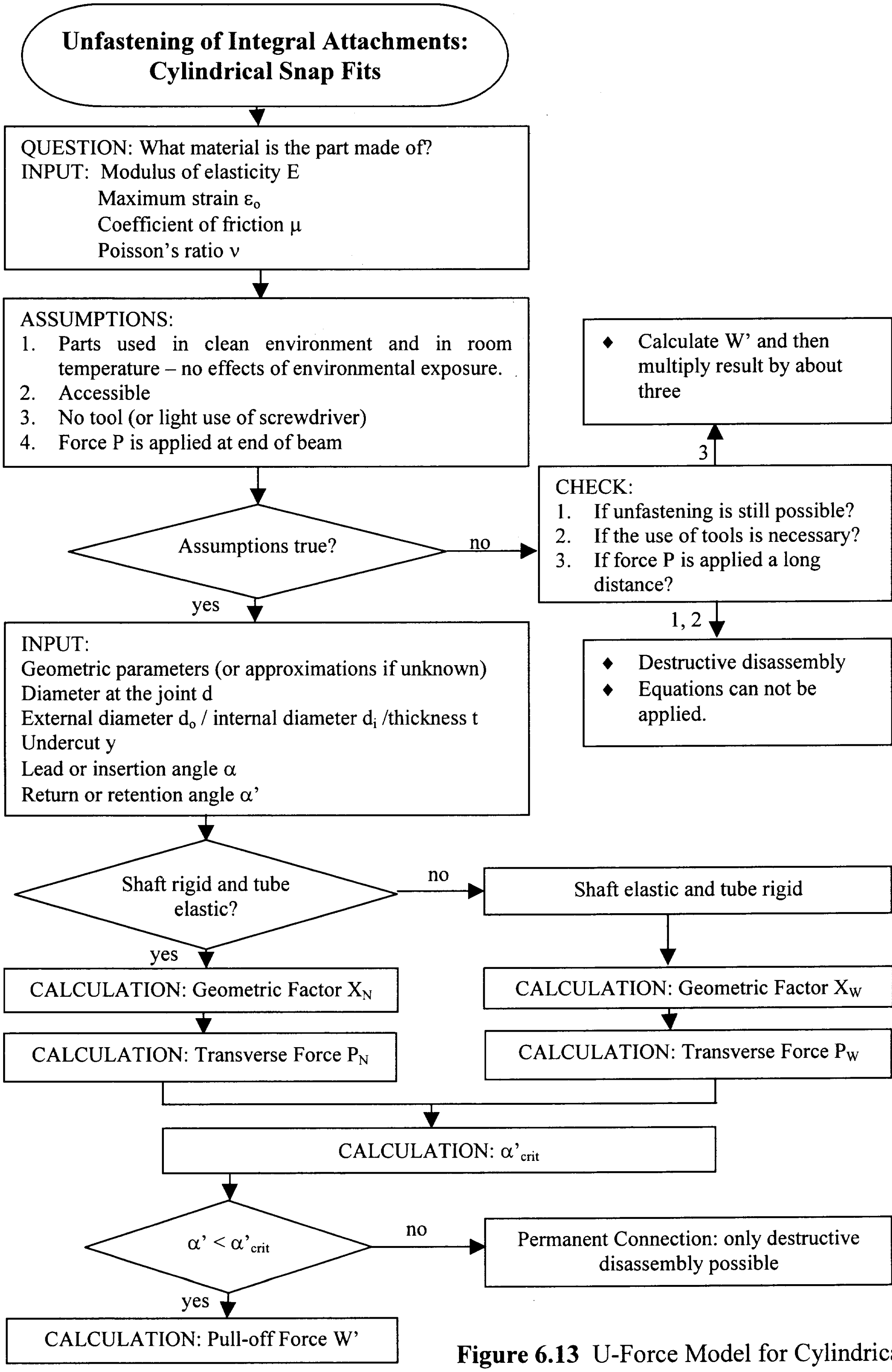


Figure 6.13 U-Force Model for Cylindrical Snap Fit

Step 1: Determine the material of the parts and input values for modulus of elasticity E, maximum strain ε_o, coefficient of friction μ, and Poisson's ratio ν.

Step 2: Check assumptions. For the calculations, it is assumed that are no effects of environmental exposure, that accessibility is given and that for the unfastening process no tools are needed. If that is not the case, then destructive disassembly might be necessary, and the equations cannot be applied. It is also assumed that the force P is applied at the end of the beam. In the case that this is not true, that means the force is applied a long distance from the end of the beam, then the pull-out force W' can be calculated according to the model. However, at the end the result has to be multiplied by about three.

Step 3: Input geometric parameters. Often not all values are known, therefore, appropriate assumptions have to be made. The design process is usually an iterative one that means it can be necessary to apply the model more than once. The required geometric parameters are the diameter at the joint d, the external diameter d_o of the tube or the internal diameter d_i of the hollow shaft or the thickness t. Further, there are the undercut y, the insertion angle α, and the retention angle α'.

Step 4: Determine if the shaft is rigid and the tube elastic. If yes, then follow step 5, otherwise continue with step 6.

Step 5: Calculate the geometric factor X_N according to Eq. 6.7 and then the transverse force P according to Eq. 6.9. Then continue with Step 7.

Step 6: Calculate geometric factor X_W according to Eq. 6.8 and then the transverse force P according to Eq. 6.10.

Step 7: Determine the critical angle and compare it with the retention angle. If the retention angle is smaller than the critical angle than go to Step 8. In the other case, if the retention angle is bigger than the critical angle, unfastening of the cylindrical snap fit will not be possible, that means the connection will be permanent, or destructive disassembly has to be used.

Step 8: Determine the pull-off force W' according to Equation 6.11 as unfastening or removal force.

6.4.3 Example for Application of U-Force Model for Cylindrical Snap Fit

Suppose a cylindrical snap fit of ABS material is chosen with the dimensions as shown in Figure 6.14. The shaft is assumed rigid. Furthermore, it is assumed that the product is used in a clean environment, that the snap fit is accessible and that it can be unfastened without the use of tools. The design is to be evaluated and it is to determine if the design is unfastening suitable.

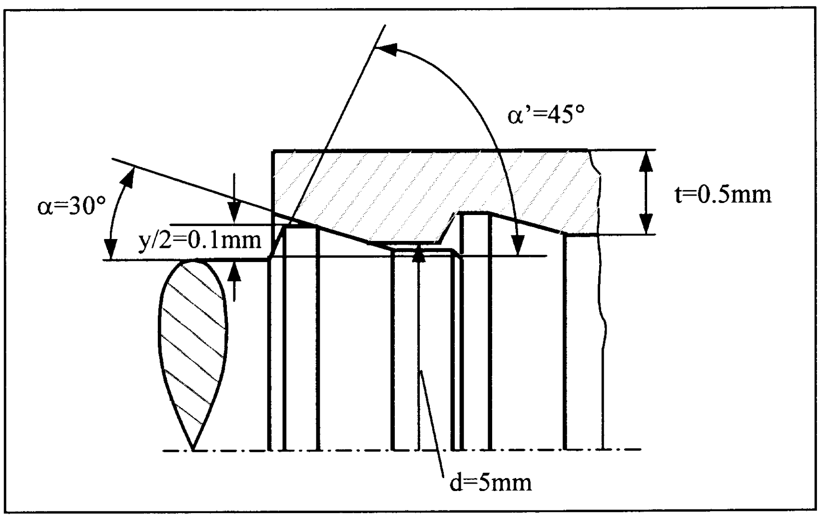


Figure 6.14 Application Example for Cylindrical Snap Fit

1. The material is ABS, that means $E = 2100 N/mm^2$, $\varepsilon_o = 0.06$, $\mu = 0.5$, and $\nu = 0.35$.

2. Assumptions are true.

3. The geometric parameters are $L = 5mm$, $d = 5mm$, $t = 0.5mm$, $y=0.2mm$, $\alpha = 30°$, and $\alpha' = 45°$.

4. Shaft rigid.

5. The geometric factor $X_N = 0.0317$, and $P = 66.6N$

7. The critical angle is $\alpha_{crit} = 63.43°$. The retention angle is smaller than the critical angle.

8. The unfastening or removal force for this cylindrical snap fit is then $W' = 199.8N$ (Eq. 6.11).

Result: Under the given circumstances, the design is suitable for unfastening.

Depending on the requirements for the cylindrical snap fit, different paths can be followed. The pullout forces can become very high for cylindrical snap fits, because they usually provide a high holding force. In order to find suitable design parameters, in next section 6.5 the relevant parameters are analyzed.

6.5 Parameter Study for Snap Fits

In the previous sections design models for the cantilever and the cylindrical snap fits have been presented. To get a better estimate of what kind of dimensions and properties should be chosen for a good design, a parameter study is conducted. Characteristic changes as result of variations in geometrical and material values are considered in detail using a sensitivity study in Pro/Mechanica.

If there is a need to find out the overall effect of varying one or more design parameters, such as dimensions, this could be done by performing a number of similar analyses. The geometry of the model has then to be changed between each analysis.

Pro/Mechanica has an automated routine, which allows specifying the parameter to be varied and its overall range. It then automatically performs all the modifications to the model, and computes results for the intermediate values of the design parameters [Toogood, 2000].

The general procedure is to set up a model – create the geometry, generate the elements, specify loads, constraints and material properties, and choose an analysis. Then a range over which the parameter should vary has to be specified for the design variables. A sensitivity study is set up identifying, which design variable should be active. The procedure automatically increments each specified design variable, and runs a designated analysis on the model for each new configuration. A result window can be set up to show the variation on some measure as a function of a designated design variable.

As a first step a static analysis is performed. A static analysis provides calculations of deflections, stresses, strains, forces, and energies for a mechanical system. Static analyses are performed under the assumption of small deflection, small strains, and linear elastic material behavior. For an estimate, the results obtained here should be sufficient in spite of this assumption. Then for both, the cantilever snap fit and the cylindrical snap fit, a parameter study is performed.

6.5.1 Parameter Study for Cantilever Snap Fits

As stated in section 6.3, cantilever snap fits can be unfastened through a pullout force W' or through a perpendicular force P. The forces are in dependency of the retention angle and the material (coefficient of friction). A model of the simulation setup for the cantilever is shown in Figure 6.15.

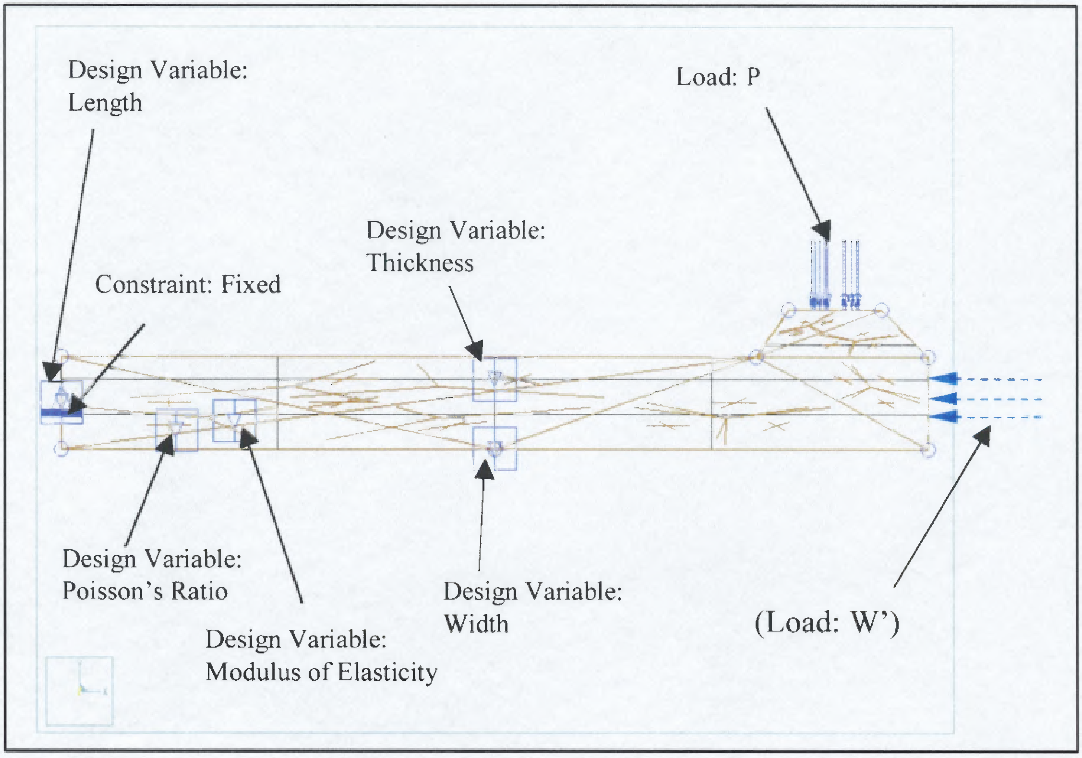


Figure 6.15 Cantilever Model for Pro/M Parameter Study

In the first case it is assumed that W' will be used to unfasten the snap fit, in the second case an unfastening from the direction P is assumed as possible. As design variables, three geometric parameters (beam length, beam width, beam thickness) and two material properties (modulus of elasticity, Poisson's ratio) have been selected. The beam length has a range from 15mm to 30mm, the beam width from 3mm to 15mm, and the thickness from 2mm to 7mm (lower margin dimensions L=15, B=3, t=2 (see application example in section 6.3.3), upper margin dimensions L=30, B=15, t=7). The insertion angle α is for all cases assumed 30° and the retention angle α' 45°. ABS, PC, PEI, and Nylon 6 have been selected as materials. For the material properties, the design variables should then cover a range suitable for all four materials (see material data in

Appendix A, Table A.4). From that follows a range of 2100N/mm^2 to 4000N/mm^2 for the modulus of elasticity and 0.35 to 0.4 for Poisson's ratio. Regarding the constraints for the cantilever, it is assumed that one end is fixed. As a load, there is either P or W'. The load can not be defined as a design variable in Pro/M. To get approximate values for the required forces, W' and P are calculated for the four materials according to the equations of section 6.3, see Table 6.1 and 6.2.

Table 6.1 Calculated Removal Forces for Lower Margin Value of Dimension Range (Q = 1.6 for L/t=7.5 and Curve 4, see Appendix B)

	P [N]	P w/ Q [N]	W' [N]	W' w/Q [N]
ABS	16.8	10.5	50.4	31.5
PC	12.8	8.0	21.3	13.3
PEI	39.2	24.5	58.8	36.8
Nylon 6	42.7	26.7	60.2	37.6

Table 6.2 Calculated Removal Forces for Upper Margin Value of Dimension Range (Q = 2.2 for L/t=4.29 and Curve 4, see Appendix B)

	P [N]	P w/ Q [N]	W' [N]	W' w/Q [N]
ABS	514.5	257.3	1543.5	771.9
PC	392.0	178.2	653.3	297.0
PEI	1200.5	545.7	1800.8	818.6
Nylon 6	1306.7	594.0	1842.0	837.3

Based on these results, a static analysis (material: ABS) has been conducted for W' and P equals 10N, 25N, and 50N for the lower margin dimensions, and W' and P equals 200N, 500N, and 1000N for the upper margin dimensions.

For the results, maximum displacement, maximum principal stress, von Mises stress, and strain energy have been selected, which can be compared for the different W' and P values. Displacement is the movement of a point on the model, measured as the change in position relative to the point's location on the undeformed model. It can be represented either in terms of magnitude or in terms of component direction. For the static analysis, the magnitude of the maximum displacement is just considered. For the sensitivity study, the displacement in y-direction is also looked at. Maximum principal stress is the most positive principal stress in the model. The other principal stress is the minimum principal stress. The planes, on which these stresses act, are called the principal planes. These planes are defined as those on which no shearing stresses exist. Von Mises stress is an equivalent stress that is a combination of all stress components. The von Mises yielding criterion states that a material reaches its elastic limit if the von Mises stress is equal to the material's yield stress in simple tension. Finally, the total strain energy is the sum of strain energy, calculated for different elements. The area under the load-deflection curve and the corresponding strain-strain curve represents the amount of work done on this material. Within the elastic limit, the amount of this work is equal to the elastic energy stored. The results obtained from the static analysis are shown in Table 6.3 (lower margin dimensions) and Table 6.4 (upper margin dimensions) and the graphs to the static analysis in Appendix C.

Table 6.3 Results from Static Analysis for Lower Margin Dimensions

	W'=10N	W'=25N	W'=50N	P=10N	P=25N	P=50N
Max. Displacem. [mm]	0.016	0.039	0.078	4.274	10.686	21.372
Max. Princ. Stress [N/mm^2]	0.379	0.947	1.894	96.875	242.190	484.370
Von Mises Stress [N/mm^2]	2.030	5.074	10.147	74.398	185.990	371.990
Strain Energy [Nmm]	0.001	0.006	0.024	2.012	12.576	50.302
Strain, actual	0.0002	0.0003	0.0010	0.0569	0.1425	0.2850

With small force values already, large displacements can be obtained For the perpendicular force P. Accordingly, the stress and strain energy values are higher for P than for W'.

Table 6.4 Results from Static Analysis for Upper Margin Dimensions

	W'=200N	W'=500N	W'=1000N	P=200N	P=500N	P=1000N
Max. Displacem. [mm]	0.032	0.080	0.160	2.769	6.922	13.843
Max. Princ. Stress [N/mm²]	64.080	160.200	320.400	0.924	2.309	4.618
Von Mises Stress [N/mm²]	2.235	5.588	11.176	41.796	104.490	208.980
Strain Energy [Nmm]	0.001	0.007	0.027	0.609	3.809	15.236
Strain, actual	0.0004	0.0009	0.0019	0.0323	0.0808	0.1615

In order to evaluate the results, the permissible strain limit is considered. For ABS the permissible strain ε_o is 0.06. If, with the maximum displacement as y (from Table 6.3 and 6.4), the actual strain value is calculated by using

$$\varepsilon = 1.5 \cdot \frac{t}{L^2} y \tag{6.20}$$

then the values have to be below the permissible strain value. Here, the strains for removal force W' are significantly below the limit. However, the problem for unfastening with W' is that higher forces are needed to obtain the necessary displacement to overcome the offset of the cantilever snap fit. For P, the higher force values could cause some problems (P=25N, 50N, 1000N). Very short (instantaneous) high strains might be tolerated (P=500N), but there is a high risk of breaking the snap fit. Therefore, if it is possible to unfasten a cantilever with a perpendicular force P, then only small forces should be applied to release the snap in order to avoid the danger of breakage. But P will be preferred to the pullout force W', which requires much higher forces. This concludes the evaluation of the static analysis for the cantilever snap fit.

For the sensitivity study, a load of 25N has been selected. The sensitivity study has also been conducted for 10N and 50N. The curves for all loads are similar, only the amounts vary. However, the single values are not really of interest. The five design variables modulus of elasticity, Poisson's ratio, length, width, and thickness, described above, are applied separately and simultaneously once. The effects that the variations of the design variables have on the maximum displacement, displacement in y-direction, maximum principal, von Mises stress, and strain energy are presented here, see for example Figure 6.16 (more graphs in Appendix D).

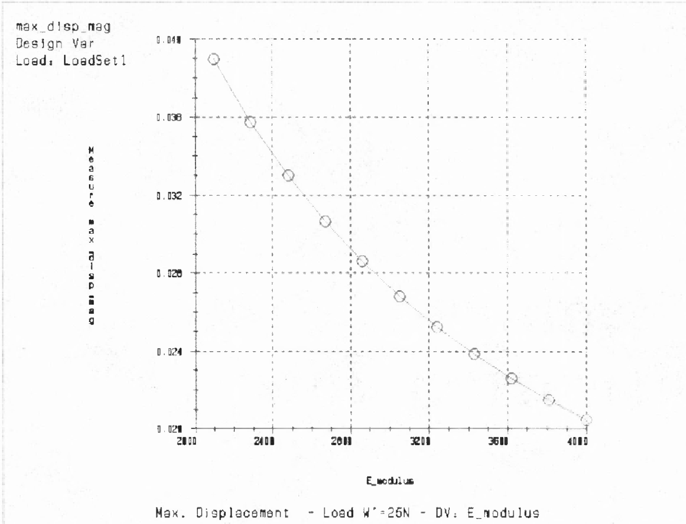


Figure 6.16 Maximum Displacement Depending on Design Variable Modulus of Elasticity for Load W'=25N

Regarding the results for the modulus of elasticity, the curve shape is similar for both load types. With increasing modulus of elasticity the maximum displacement (MaxDis) decreases, and also the displacement in y-direction (DinY), which in this particular case has negative values due to the orientation of the coordinate system, see Figure 6.15. The curve looks similar to the one for strain energy (STE). The maximum principal stress (MPS) is constant for all modulus of elasticity values within the given range. The same is true for the von Mises stress (VMS).

In the next study, Poisson's ratio is the active design variable. With increasing Poisson's ratio the maximum displacement for load W' increases, but decreases for load P. The same happens for DinY. The maximum principal stress increases with higher Poisson's ratio, for both W' and P. VMS shows a decreasing curve for W' and increasing for P. The strain energy goes down for both loads.

Increasing length values mean a rising curve for MaxDis and DinY (for W' even linear proportional). MPS for W' shows a parabola shaped curve with a minimum, for P a linear curve. For P the VMS curve is linearly rising, too, but for W' the curve is an approximately parabola shaped curve with a maximum. STE has an increasing curve. For design variable width, the curves again are in similar shape for both W' and P. With increasing width value all curves are falling. It is the same for the thickness values, but there are some uneven points. If all design variables are in effect, the overall effect for all curves is a decreasing function.

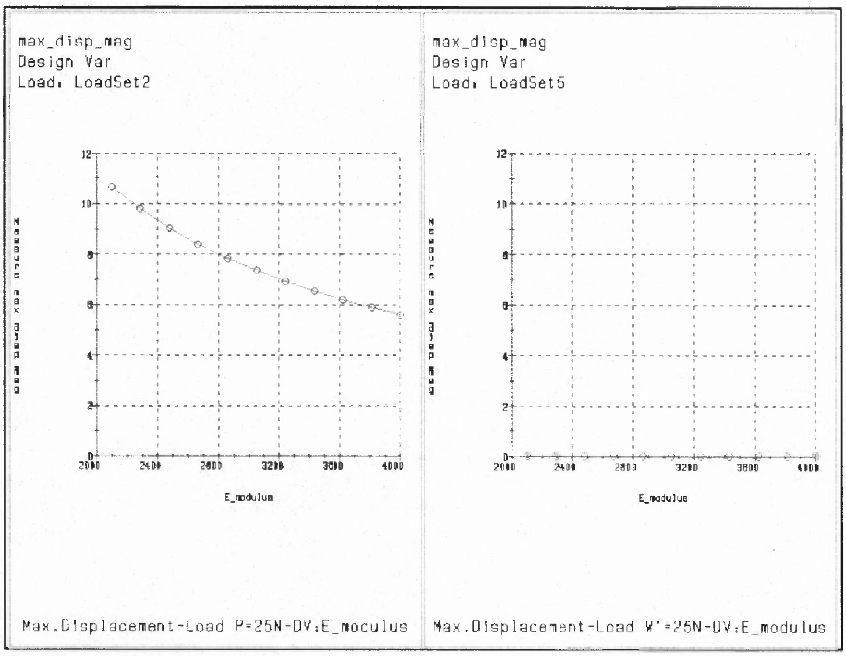


Figure 6.17 Maximum Displacement Depending on Design Variable Modulus of Elasticity for P and W' in Comparison

These results are different in magnitude for P and W’ values. To see the difference, both results are brought to the same scale in Figure 6.17 and 6.18. As can be seen here, the values for P are so much higher, that the W’ appears to be zero.

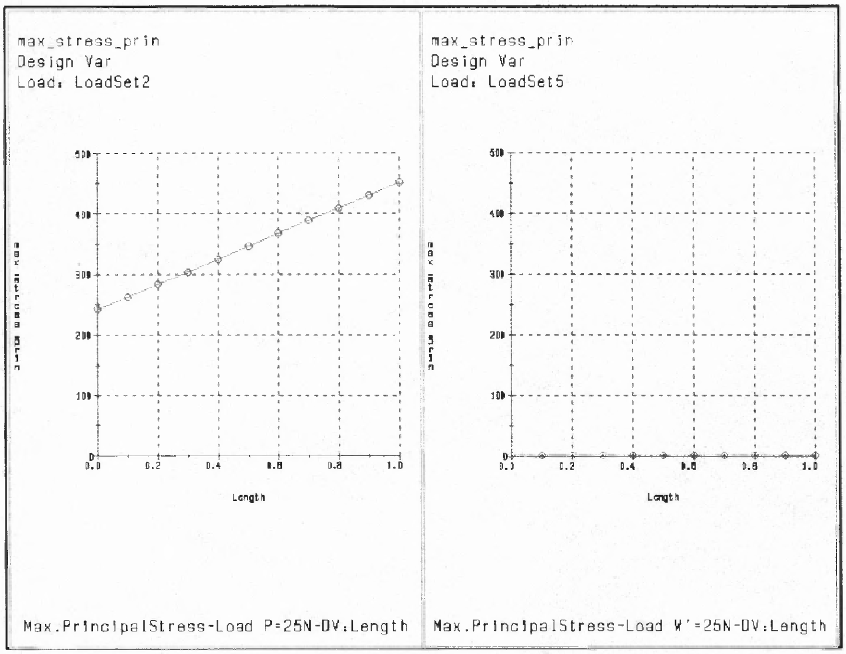


Figure 6.18 Maximum Principal Stress Depending on Design Variable Length and Width for P and W’ in Comparison

As a summary it can be said, that in order to reach a maximum displacement, the modulus of elasticity should be low. The Poisson’s ratio should be high for W’ and low for P. Furthermore, the longer the beam length and the smaller the beam width and the thinner the beam thickness, the higher the displacement will be.

6.5.2 Parameter Study for Cylindrical Snap Fits

In section 6.4, two cases have been distinguished for the calculation of cylindrical snap fits. The force is applied at the end of the beam or the force is applied a long distance from the end of the beam. For the parameter study, the first case will be considered only. Opposed to the cantilever snap fit, it is usually very difficult to apply a transverse force P to unfasten the cylindrical snap fit. Therefore, only the pullout forces W' will be studied here. Among other parameters, the force depends upon the retention angle and the material (coefficient of friction). A model of the simulation setup for the cylindrical snap fit is shown in Figure 6.19 (shaft) and in Figure 6.20 (tube). The cylindrical snap fit is an axisymmetric part, and therefore, can be defined by a planar cross-section, which is revolved around a central axis. This reduces simulation time and simplifies the analysis.

Geometric parameters (shaft radius and tube radius) and material properties (modulus of elasticity, Poisson's ratio) have been selected as design variables. Because of limitations to the design variables in the Pro/M simulation, four cases of different geometric dimensions have been considered for the shaft and the tube, respectively. The shaft radius has a range from 2.5mm to 4mm for case I, one from 4mm to 8mm for case II, one from 8mm to 12mm for case III, and 12mm to 20mm for case IV.

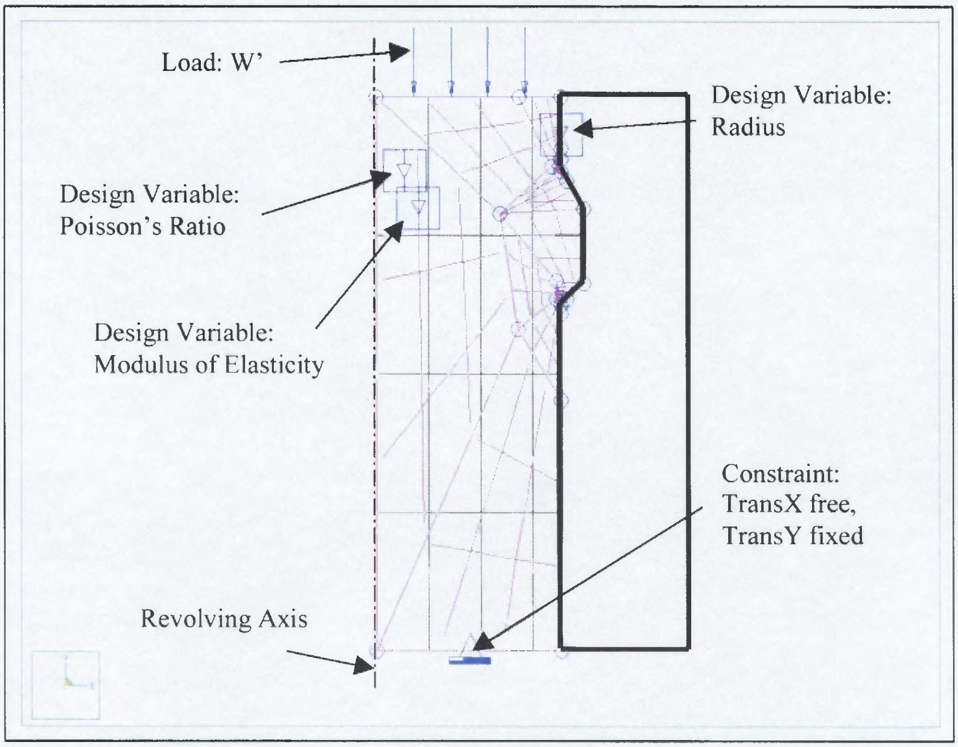


Figure 6.19 Cylindrical Snap Fit Shaft Model for Pro/M Parameter Study

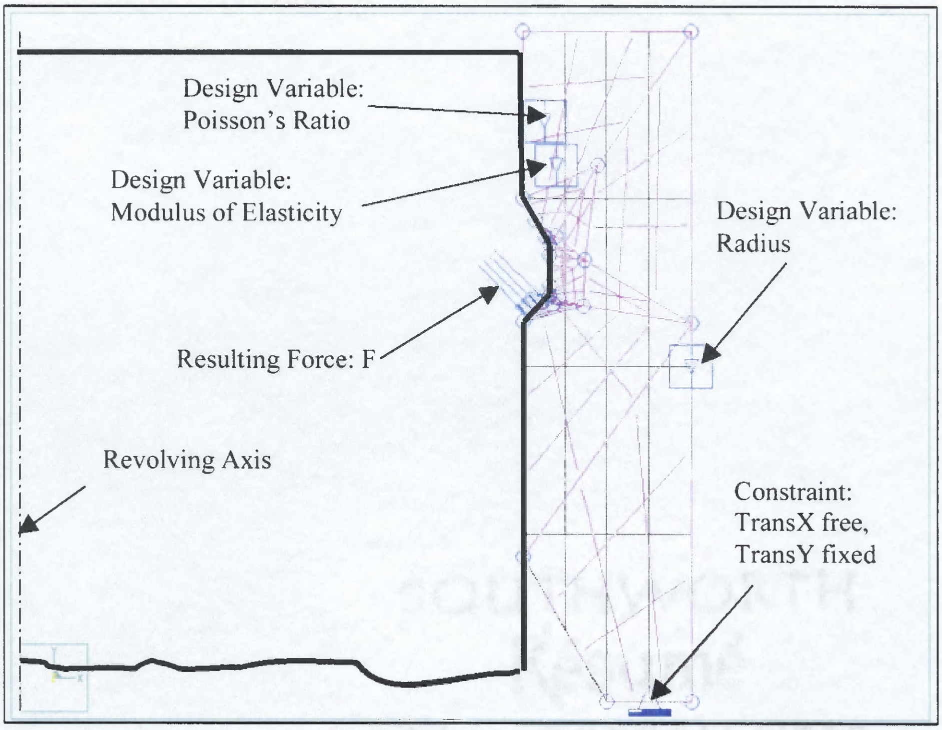


Figure 6.20 Cylindrical Snap Fit Tube Model for Pro/M Parameter Study

The outer radius for the tube has in case I a range of 3mm to 6mm, in case II 6mm to 11mm, in case III 11mm to 16mm, and in case IV 16mm to 20mm. The insertion angle α is for all cases assumed 30° and the retention angle α' 45°. Just as for the cantilever snap fit, materials ABS, PC, PEI, and Nylon 6 have been selected. For the material properties, similarly, the design variables should cover then a range suitable for all four materials (see material data in Appendix A, Table 4). From that follows a range of 2100N/mm^2 to 4000N/mm^2 for the modulus of elasticity and 0.35 to 0.4 for Poisson's ratio. As constraints for the cylindrical snap fit the translation in x is free and the translation in y has to be fixed (axisymmetric case). And the load is the push-out/pullout force W'. As before, the load can not be defined as a design variable in Pro/M. To get approximate values for the required forces, W' and P are calculated for the four materials according to the equations of section 6.4, see Table 6.5.

Table 6.5 Calculated Removal Forces for Cylindrical Snap Fit

	ABS	PC	PEI	Nylon 6
Case I	W'=199.8N (P=66.6N)	W'=84.1N (P=50.5N)	W'=233.1N (P=155.4N)	W'=236.4N (P=167.7N)
Case II	W'=1515.9N (P=505.3N)	W'=635.2N (P=381.1N)	W'=1768.5N (P=1179.0N)	W'=1778.8N (P=1261.9N)
Case III	W'=4420.2N (P=1473.4N)	W'=1855.5N (P=1113.3N)	W'=5156.9N (P=3437.9N)	W'=5202.5N (P=3690.7N)
Case IV	W'=8674.0N (P=2891.3N)	W'=3643.7N (P=2186.2N)	W'=10119.7N (P=6746.5N)	W'=10220.8N (P=7250.7N)

For the static analysis (material: ABS) the loads for the different cases are:

- case I: 100N, 200N, 500N, 1000N, and 2000N,
- case II: 500N, 1500N, 2000N, and 5000N,
- case III: 1500N, 4500N, 6000N, 10000N, and 15000N,
- case IV: 3000N, 9000N, 12000N, 20000N, and 30000N.

The results obtained from the static analysis are shown in Table 6.6 for the shaft and in Table 6.7 for the tube and the graphs to the static analysis in Appendix E.

As mentioned before, the permissible strain limit for ABS is 0.06. With maximum displacement (from Table 6.6 and 6.7), the actual strain value is calculated by using

$$\varepsilon = \frac{y}{d} \tag{6.21}$$

and the values have to be below the limit. As can be seen, the values are below the limit for all forces. But only with the higher force values can a displacement be obtained, which is high enough to overcome the engagement.

Table 6.6 Results from Static Analysis for Shaft

	Max. Displacement [mm]	Max. Principal Stress [N/mm²]	Von Mises Stress [N/mm²]	Strain Energy [Nmm]	Strain, actual
Case I					
W'=100N	0.012	1.762	7.957	0.0175	0.002
W'=200N	0.024	3.523	15.913	0.070	0.005
W'=500N	0.061	8.808	39.783	0.438	0.012
W'=1000N	0.122	17.615	79.565	1.751	0.024
W'=2000N	0.243	35.231	159.130	7.004	0.049
Case II					
W'=500N	0.038	3.049	17.357	0.084	0.005
W'=1500N	0.114	9.148	52.070	0.753	0.014
W'=2000N	0.152	12.197	69.426	1.339	0.019
W'=5000N	0.379	30.493	173.570	8.369	0.047
Case III					
W'=1500N	0.071	2.389	12.001	0.038	0.004
W'=4500N	0.214	7.167	36.002	0.346	0.013
W'=6000N	0.285	9.556	48.003	0.615	0.018
W'=10000N	0.475	15.927	80.006	1.709	0.030
W'=15000N	0.712	23.890	1200.1000	3.845	0.045
Case IV					
W'=3000N	0.095	2.164	9.629	0.024	0.004
W'=9000N	0.285	6.493	28.888	0.215	0.012
W'=12000N	0.380	8.657	38.517	0.383	0.016
W'=20000N	0.633	14.428	64.194	1.063	0.026
W'=30000N	0.949	21.642	96.292	2.391	0.040

For the sensitivity study, for case I a load of 500N, for case II 2000N, case III 4500N, and for case IV 9000N has been selected. The sensitivity study has also been conducted for other loads, but the curves are similar for all loads. The three design variables, modulus of elasticity, Poisson's ratio, and radius are applied separately, and simultaneously once. The effects that the variation of the design variables have on the maximum displacement, maximum principal stress, von Mises stress, and strain energy are presented here, see for example Figure 6.21 (more graphs in Appendix F).

Table 6.7 Results from Static Analysis for Tube

	Max. Displacement [mm]	Max. Principal Stress [N/mm²]	Von Mises Stress [N/mm²]	Strain Energy [Nmm]	Strain, actual
Case I					
F=100N	0.030	23.040	38.678	0.714	0.005
F=200N	0.060	46.080	77.357	2.858	0.010
F=500N	0.151	115.200	193.390	17.860	0.025
F=1000N	0.302	230.400	386.780	71.440	0.050
F=2000N	0.603	460.800	773.570	285.760	0.101
Case II					
F=500N	0.036	18.958	32.285	0.400	0.003
F=1500N	0.107	56.873	96.854	3.603	0.009
F=2000N	0.143	75.831	129.140	6.405	0.012
F=5000N	0.357	189.580	322.850	40.031	0.030
Case III					
F=1500N	0.083	13.777	26.713	0.302	0.004
F=4500N	0.249	41.332	80.138	2.715	0.011
F=6000N	0.332	55.110	106.850	4.826	0.015
F=10000N	0.553	91.849	179.090	13.405	0.025
F=15000N	0.830	137.770	267.130	30.161	0.069
Case IV					
F=3000N	0.127	14.807	25.098	0.290	0.004
F=9000N	0.381	44.421	75.294	2.614	0.012
F=12000N	0.508	59.228	100.390	4.647	0.016
F=20000N	0.846	98.713	167.320	12.909	0.026
F=30000N	1.269	148.070	250.980	29.046	0.040

The results for the shaft are looked at first. Considering the graphs for the design variable modulus of elasticity, the curves are similar for all loads. That means, the maximum displacement decreases with increasing modulus of elasticity, the maximum principal stress and the von Mises stress stay constant, and the strain energy also decreases with increasing modulus of elasticity.

For the design variable Poisson's ratio, the maximum displacement and the maximum principal stress increase linear proportional with increasing ν for the case I and II. But for case III the displacement decreases linearly proportional, and for case IV it has a curve with a minimum.

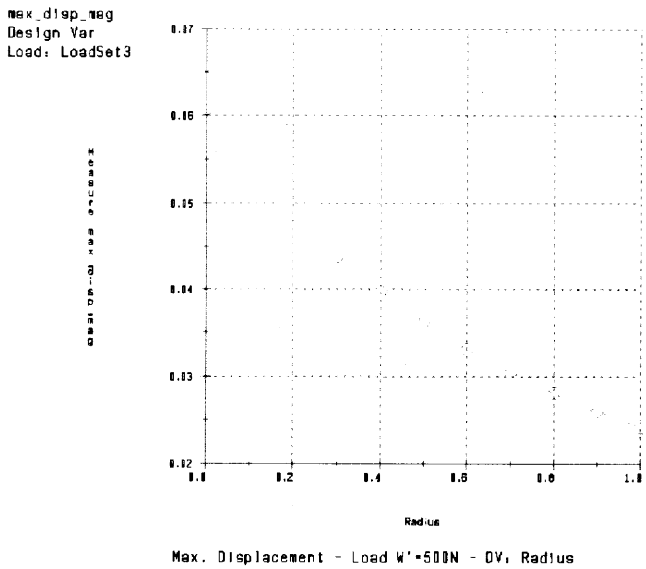


Figure 6.21 Maximum Displacement Depending on Design Variable Modulus of Elasticity for the Shaft and a Load of W'=500N

The von Mises stress and the strain energy decrease in all cases linear proportional with increasing ν. For the design variable radius, in all cases the maximum displacement, the von Mises stress, and the strain energy have a decreasing curve. In case I and II, the maximum principal stress has a decreasing curve, too. For case III and IV, the curve is s-shaped. In the case, where all design variables are active, the results are mainly determined by the radius. That means the shapes of the curves are similar to the once for the radius.

For the tube, the results are similar. If the design variable modulus of elasticity is active, the results are similar to the ones above. That means, the maximum displacement decreases with increasing modulus of elasticity, the maximum principal stress and the von Mises stress stay constant, and the strain energy also decreases with increasing modulus of elasticity. For the design variable Poisson's ratio, the maximum displacement and the strain energy increase linear proportional with increasing ν. The maximum principal stress is also linear increasing, only in case II first the curve decreases and then increases proportionally. The von Mises stress behaves similar, in case I and III the curves is linear decreasing, but in case II and IV the curve rises again after a certain point. When the radius is the active design variable, the maximum displacement, and also the strain energy decrease with increasing radius. The maximum principal stress has a curve that is slightly different in every case, but basically the curve is rising extremely, has a peak (maximum), and then decreases again, only in case IV the curve behaves in an opposite manner (minimum). The von Mises stress curve decreases with increasing radius in all cases except case I, where the curve drops to a peak and then rises again. If all design variables are active, the results look similar to the one of the radius. That means the radius is the driving design variable.

For cylindrical snap fits, usually high forces are needed to obtain enough displacements to overcome the engagement through the undercut. Therefore, it is important to look at the maximum displacement. To reach maximum displacement, the modulus of elasticity should be low for both the shaft and the tube. Overall, the Poisson's ratio should be high for maximum displacement, but might be different for specific geometric combinations (see case III and IV of the shaft, Appendix E). The smaller the radius of the shaft or tube is, the bigger is the maximum displacement.

The parameter study shows how important it is, to look at the effects geometric and material parameters have. For the cantilever snap fits, for example, the possibility of applying a force from direction P enables much lower unfastening forces than from axial direction (W'). However, not in every design access from direction P can be implemented. According to this study, if designing a cantilever snap fit, it is preferred to have a long (>L), narrow (<B), and thin (<t) beam. This would yield a higher displacement. For the cylindrical snap fit, smaller radii are preferred. They need less unfastening effort. Considering the material properties, with a lower modulus of elasticity less unfastening effort is required. The Poisson's ratio should be high for easier unfastening from direction W' (axial direction in the case of cylindrical snap fits) and low for unfastening from direction P (for cantilever snap fits only). For cylindrical snap fits, removal forces can become quite large. Therefore, it is suggested to consider smaller retention angles compared to cantilever snap fits besides the other geometric recommendations.

CHAPTER 7

CONCLUSIONS AND RECOMMENDATIONS FOR FUTURE RESEARCH

7.1 Contributions

This thesis made the following contributions to the understanding of the unfastening component of disassembly:

1. In this research, unfastening and related parameters have been defined grounded in the theory of assembly and disassembly.

A standard nomenclature for defining unfastening related parameters and variables have been introduced for the first time.

2. A model to obtain unfastening effort values has been developed.

The U-Effort model is a detailed study about the unfastening effort and the design attributes of commonly used fasteners. The unfastening effort encompasses all effects that different influencing factors can have on an unfastening process. These influencing attributes for the unfastening effort regarding the geometry and shape of the fasteners have been considered in the model in the first part of the study. Therefore, basic guidelines for the ease of unfastening have been provided for designers to use.

3. Disassembly motions have been analyzed.

The U-Effort model has been extended to include the effects of unfastening motions and hence estimate disassembly complexity. The unfastening or disassembly motions have been set into relationship with influencing factors like material, the condition at the end-of-life of the product, tools, and accessibility. Symbols for the disassembly motions have been introduced, which can be used to simplify the unfastening process by marking the specific fastener type on the product with visuals. It would be desirable to include these or other standardized visuals in the design process of future products.

4. A model to calculate removal forces for cantilever and cylindrical snap fits has been developed.

As an addition to the U-Effort model, the U-Force model covers in detail the effects of unfastening forces. In the U-Force model, unfastening considerations have been included in the design phase through the calculation of unfastening forces. A flowchart has been developed for the design process of cantilever and cylindrical snap fits, which includes unfastening considerations. This is a useful application in addition to the U-Effort model and can easily be used by designers during the design process. The parameter study provides an answer to the question in which way certain geometrical shapes and material properties influence the unfastening forces.

7.2 Recommendations for Future Research

The author would like to recommend further research regarding the following aspects to be done:

1. Implementation of the U-Effort and U-Force model into special computer software for disassembly.
2. Comparing study using the U-Effort model for different product brands.
3. Designing of new fasteners, which make the unfastening process easier and more efficient.
4. Taking the given models for manual disassembly to the next step and develop possible extended models for automated disassembly.

APPENDIX A

MATERIAL DATA

Material properties are important in the unfastening process. Here, different material data are shown.

Table A.1 General Material Properties [Rensselaer Polytechnic Institute, 1995, Mott, 1999, et al.]

Material	Frict. Coeff.	Yield Stress s_y		Young's Modulus E		Flexural Stress		Flexural Modulus	
		psi	MPa	psi	MPa	psi	MPa	psi	MPa
ABS	0.12					9000	62.055	270000	1861.65
ABS Dow Magnum 3661	0.75	5000	34.475	270000	1861.65				
ABS GE Cycolac	0.12	6000	41.370	325000	2240.88				
ABS GE Cycolac AR		7000	48.265	330000	2275.35				
ABS GE Cycolac BDT 6500		7000	48.265	320000	2206.40				
ABS GE Cycolac DFAR		6200	42.749	340000	2344.30				
ABS GE Cycolac GPM5500		6900	47.576	360000	2482.20				
ABS + PC	0.12					13200	91.014	360000	2482.20
Acetal	0.12	8700	59.987	400000	2758.00				
High-Impact Polystyrene	0.20	3481	24.001	255257	1760.00				
Lexan	0.12					11000	75.845	300000	2068.50
Modified PPO GE Noryl		8800	60.676	374500	2582.18				
Nylon 6,6	0.16	11300	77.914	409000	2820.06				
PBT	0.12					12000	82.740	340000	2344.30
PBT + PC	0.12					10000	68.950	250000	1723.75
PC GE Lexan HPIR		9000	62.055	345000	2378.78				
PC GE Lexan SP1010R		8500	58.608	300000	2068.50				
PC + ABS Dow Pulse 1725	0.65	8400	57.918	360000	2482.20				
PC+ABS GE Cycoloy C1200		8800	60.676	330000	2275.35				
PC+ABS GE Cycoloy MC1300		7300	50.334	310000	2137.45				
PC+ABS GE Cycoloy MC8002		8200	56.539	320000	2206.40				
PC Dow Calibre 800-10	0.55	8800	60.676	300000	2068.50				
PEI	0.12					22000	151.690	480000	3309.60
PEI GE ULTEM 1010F		15200	104.804	430000	2964.85				
PEI GE ULTEM 2110		16600	114.457	630000	4343.85				
PEI GE ULTEM 2400		27000	186.165	1700000	11721.50				
Polycarbonate	0.12	8850	61.021	300000	2068.50				
Polycarbonate	0.14					34000	234.430	14200	97.91
Polycarbonate DOW Calibre 800-10	0.55	8800	60.676	300000	2068.50				
Polypropylene	0.16	4900	33.786	200000	1379.00				
PPE + PS	0.12					13500	93.083	360000	2482.20
PPE+PS GE Noryl N190HX		9050	62.400	357000	2461.52				
PPE+PS GE Noryl PX1269		8000	55.160	420000	2895.90				
PPE+PS GE Noryl PXW20		5900	40.681	280000	1930.60				

Table A.2 Allowable Strain Values, ε_o [Honeywell/Allied Signal, 1998, University of Erlangen-Nuremberg, 1998]

Material	epsilon, ε_o
unfilled semi-crystalline thermoplastics	
PE (Polyethylene)	8.0%
PP (Polypropylene)	6.0%
PA (polyamide) - under moisture	6.0%
PA (polyamide) - dry	4.0%
Nylon 6	8.0%
POM (Polyoxymethylene)	6.0%
PBT (Polybutylene terephthalate)	5.0% - 8.8%
PEI (Polyetherimide)	9.8%
unfilled amorphous thermoplastics	
PC (Polycarbonate)	4.0% - 9.2%
PC + ABS (Polycarbonate + Acrylonitrile...)	3.0%
PC/PET (Polycarbonate/ Polyethylene ....)	5.8%
ABS (Acrylonitrile-butadiene-styrene)	2.5% - 7.0%
CAB (Cellulose acetate butyrate)	2.5%
PVC (Polyvinyl chloride)	2.0%
PS (Polystyrene)	1.8%
Acetal	1.5%(1)
glass filled thermoplastics	
30% glass filled PA (polyamide)	2.0%
30% glass filled PA (polyamide) dry	1.5%
30% glass filled Nylon 6	2.1% (1)
30% glass filled PC (Polycarbonate)	1.8%
30% glass filled PET (Polyethylene terephthalate)	1.5%
30% glass filled PBTP	1.5%
30% glass filled ABS (Acrylonitrile-butad....)	1.2%
45% glass filled PPS (Polyphenylenesulfide)	1.0%

(1) 70% of tensile yield strain value

Table A.3 Coefficient of Friction [Honeywell/Allied Signal, 1998][2]

MATERIAL	μ
PEI	0.20-0.25
PC	0.25-0.30
Acetal	0.20-0.35
Nylon 6	0.17-0.26
PBT	0.35-0.40
PC/PET	0.40-0.50
ABS	0.50-0.60
PET	0.18-0.25

Table A.4 Material Data [Parametric Technology, 2000, Mark, 1999, G. Carter, D. Paul, 1991]

	E [N/mm2]	ε_o	ν	μ
ABS	2100	0.060	0.35	0.50
PC	2400	0.040	0.38	0.25
PEI	3000	0.098	0.35	0.20
Nylon 6	4000	0.080	0.40	0.17

[2] Material tested against itself

APPENDIX B

DEFLECTION MAGNIFICATION FACTOR

The deflection magnification factor, Q, depends on the location of the snap fit and includes the influence through the aspect ratio of beam length-to-beam thickness, and different beam configurations, shown in Figure B.1 [Honeywell/Allied Signal, 1996]. That means, usually the base of the cantilever is assumed as rigid. However, that is not every time the case, sometimes a cantilever hook protrudes out of a plate. The position can make a difference for the deflection value.

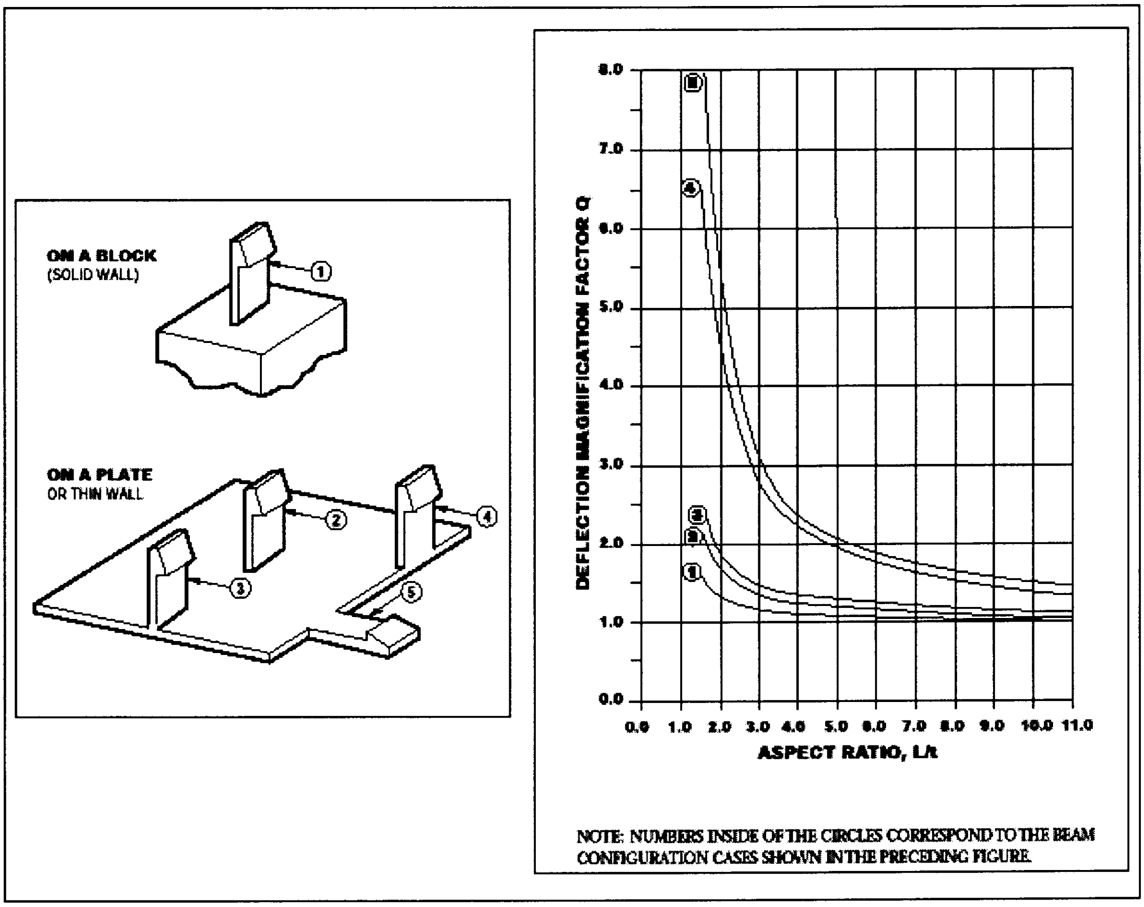


Figure B.1 Beam Configurations and Deflection Magnification Factor Q

Five beam configurations were considered: a snap fit on a solid wall (1), in the middle of the part (2), with its width parallel to the part edge (3), on the edge of the molded part continuing in the same plane (4), and with its thickness parallel to the part edge (5).

APPENDIX C

RESULTS OF THE STATIC ANALYSIS FOR THE CANTILEVER SNAP FIT WITH APPLIED PULLOUT FORCE W' AND PERPENDICULAR FORCE P AS REMOVAL FORCES

Lower Dimension Values:

1. Load: W' = 10N

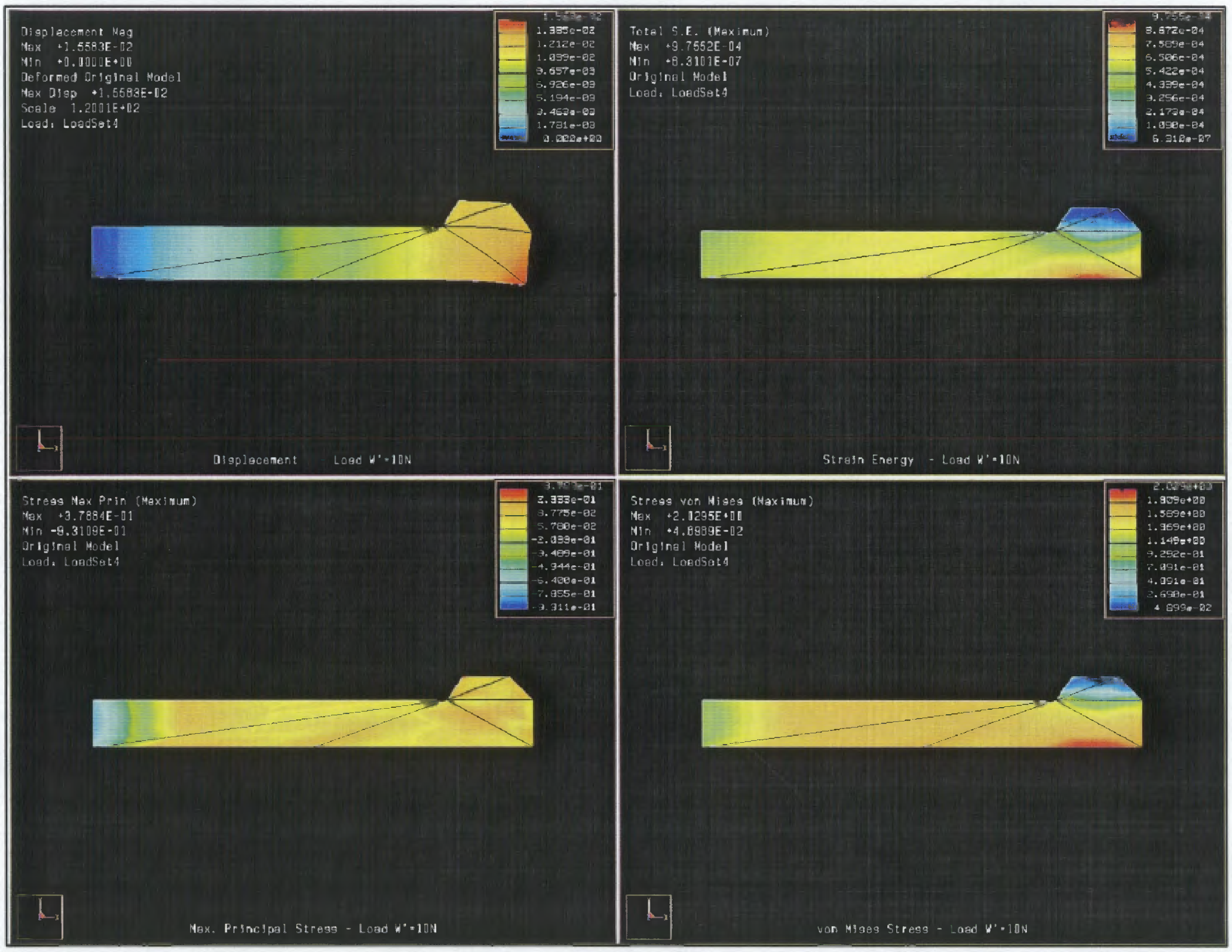


Figure C.1 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load W' of 10N

2. Load: W' = 25N

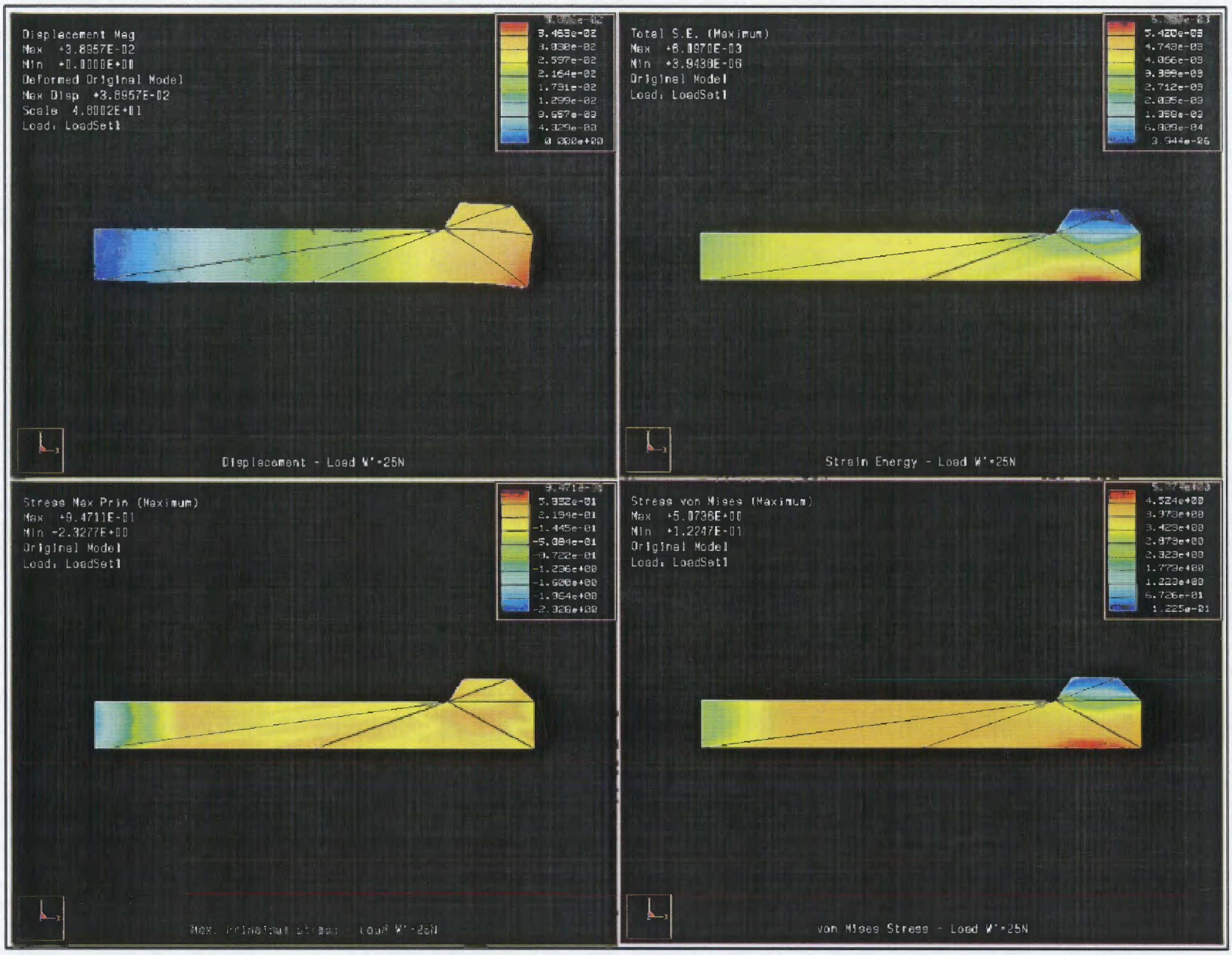


Figure C.2 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load W' of 25N

3. Load: W' = 50N

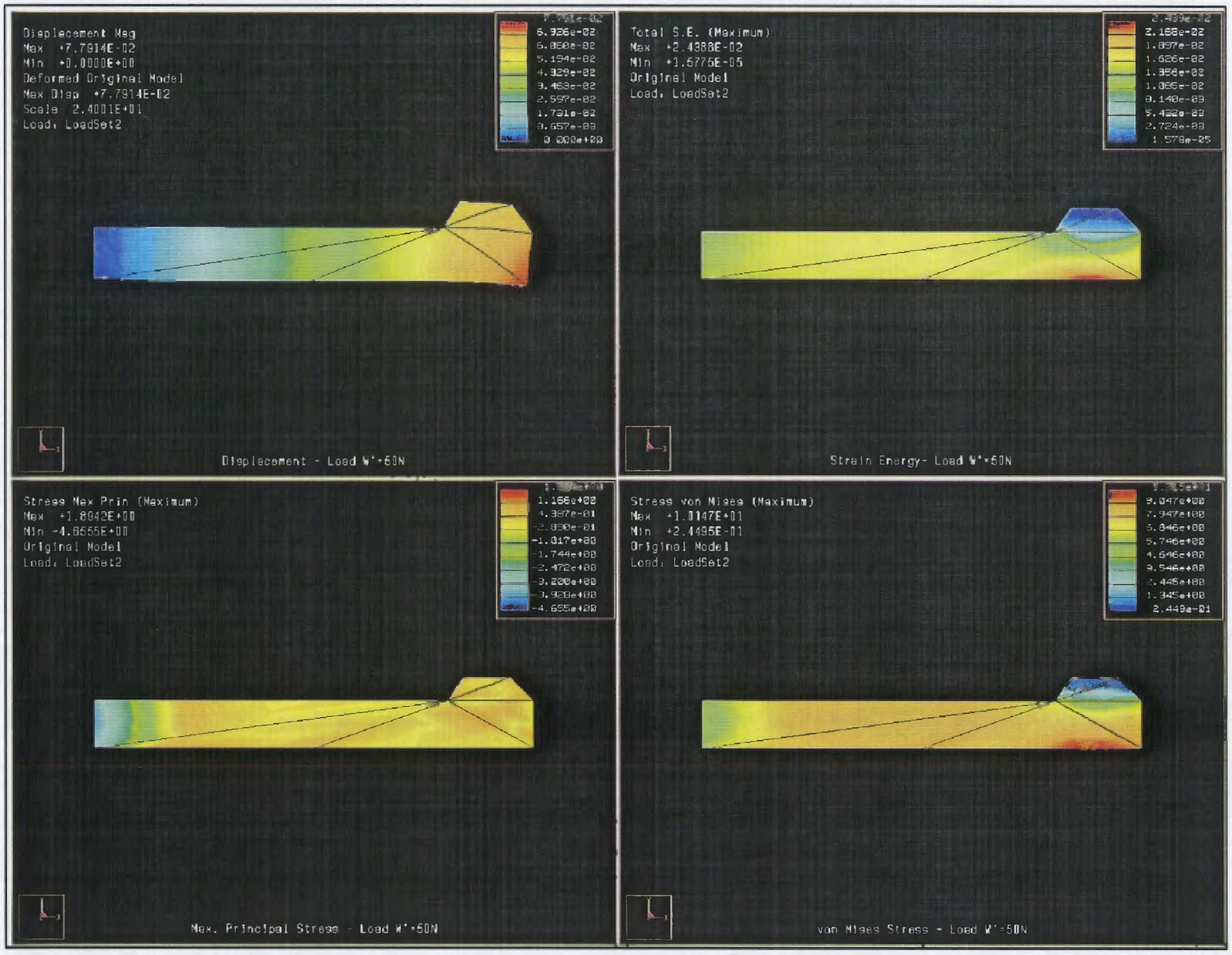


Figure C.3 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load W' of 50N

4. Load: P = 10N

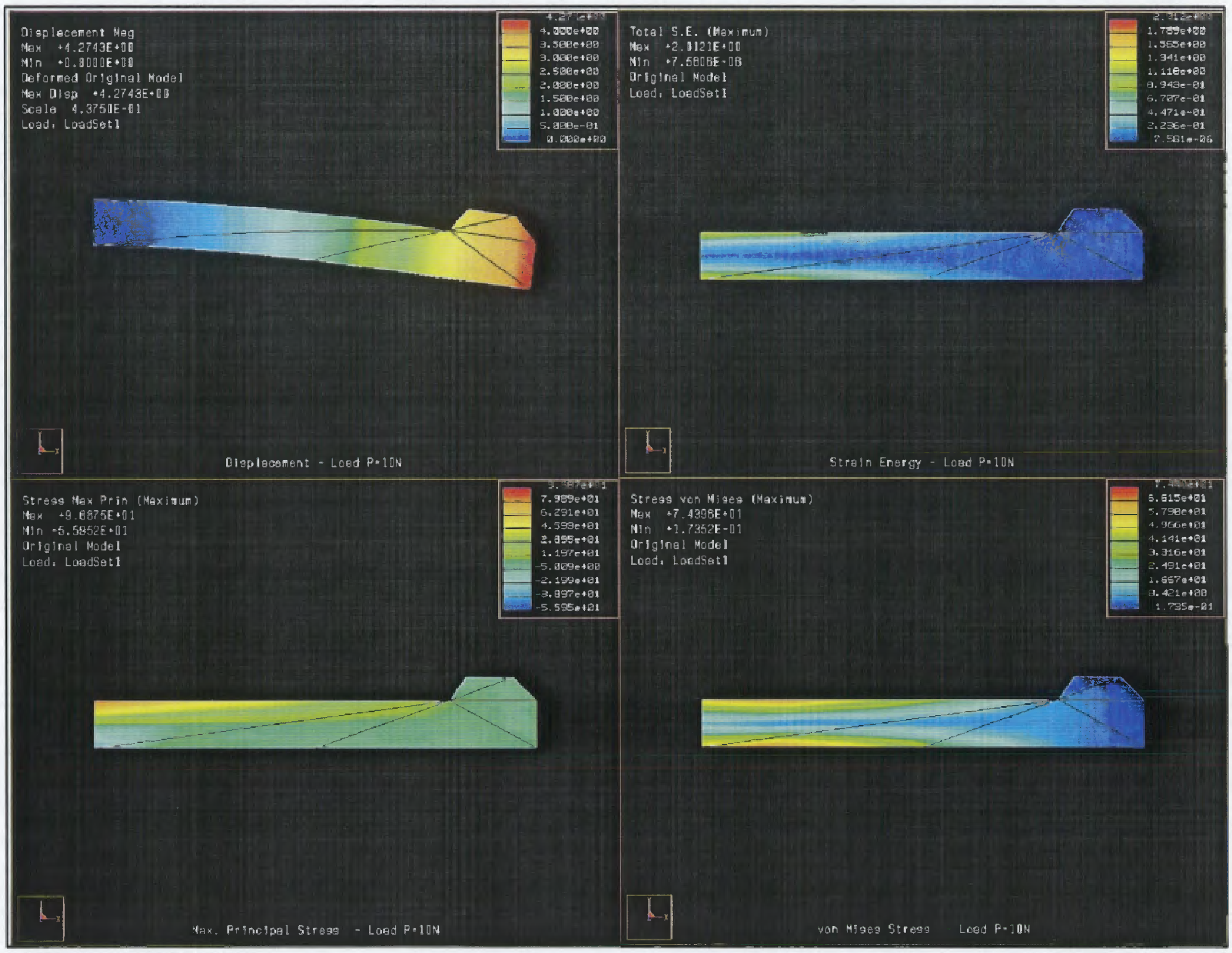


Figure C.4 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load P of 10N

SOUTHWORTH
Résumé
100% COTTON FIBER

5. Load: P = 25N

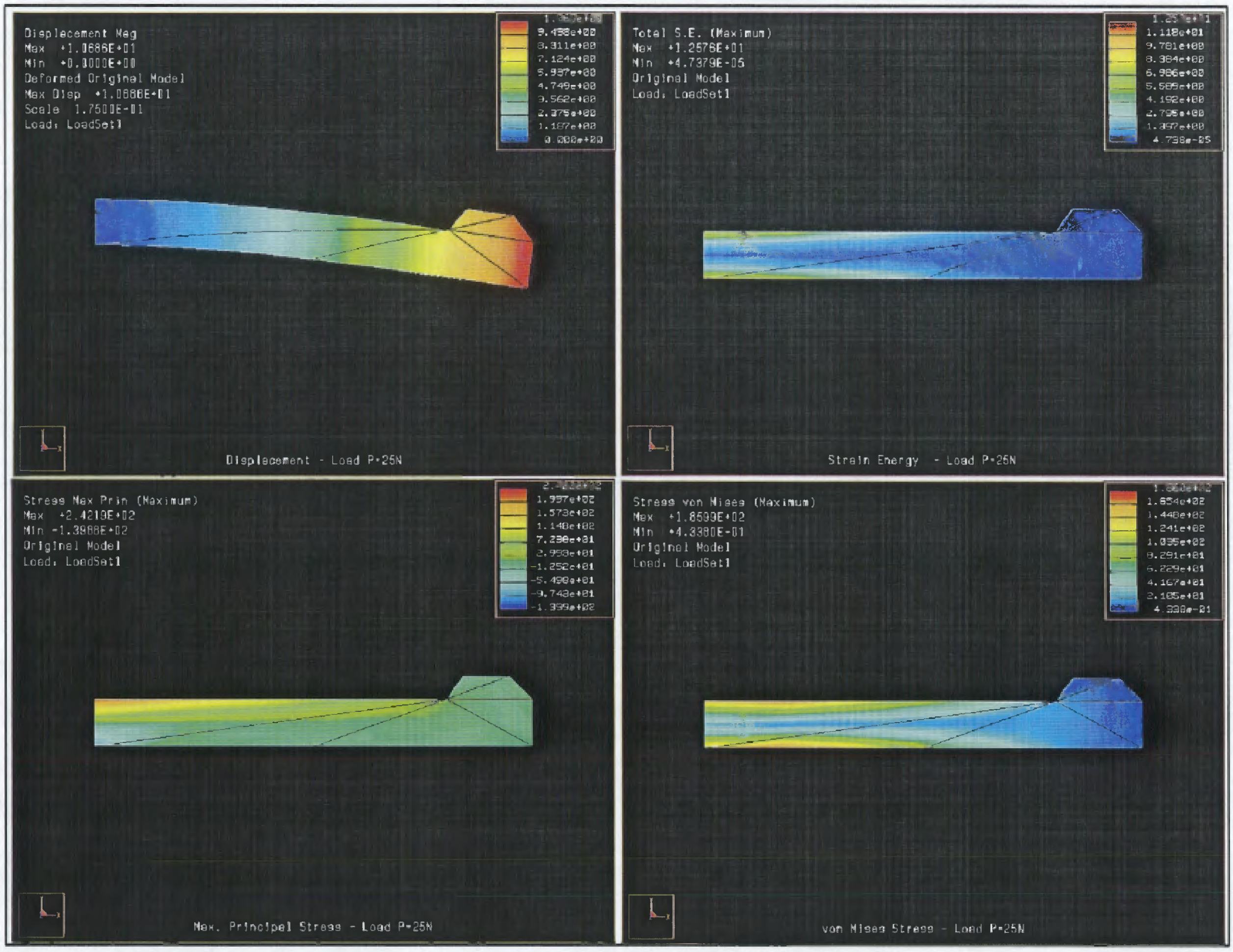


Figure C.5 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load P of 25N

6. Load: P = 50N

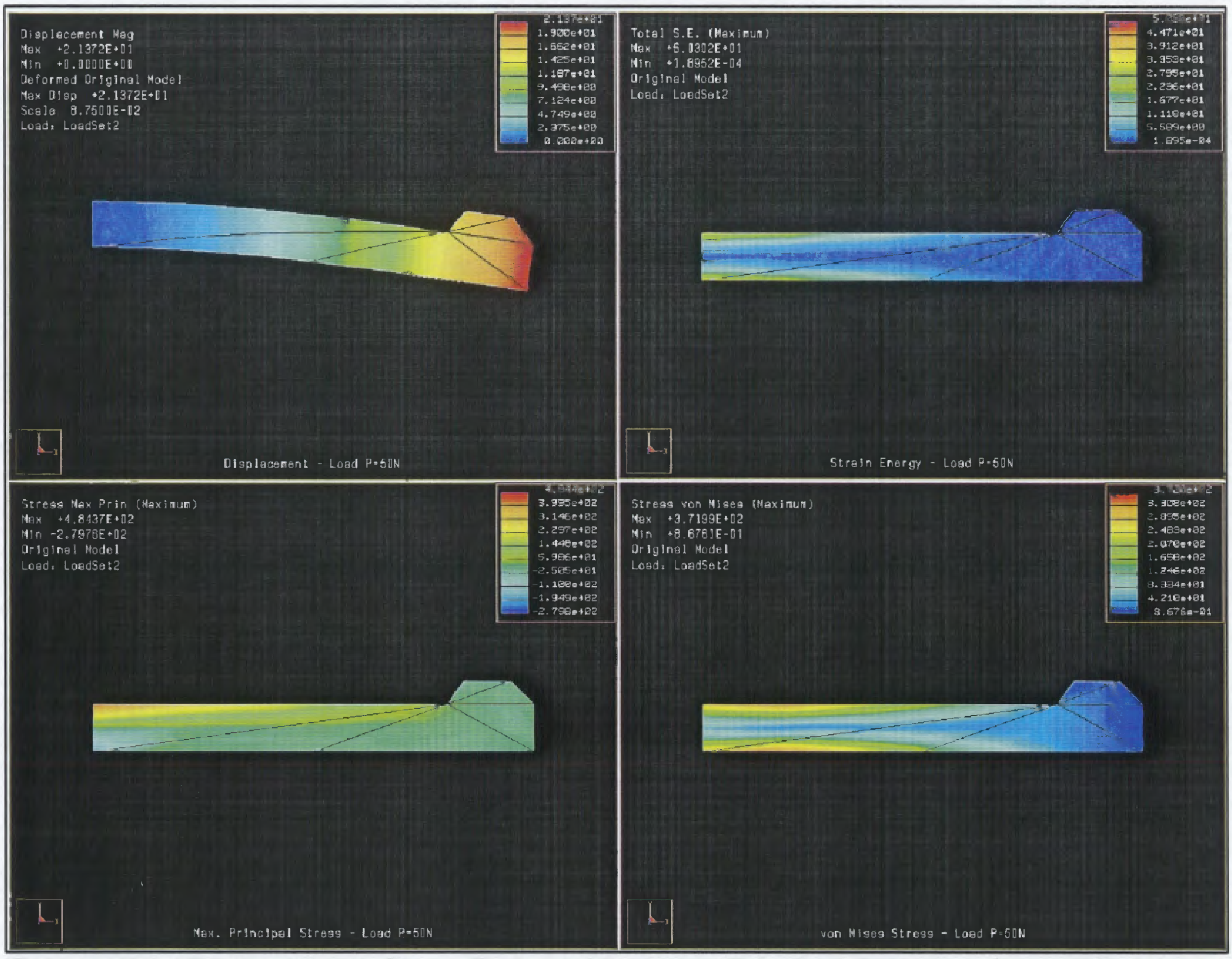


Figure C.6 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load P of 50N

SOUTHWORTH
Résumé
100% COTTON FIBER

Upper Dimension Values:

1. Load: W' = 200N

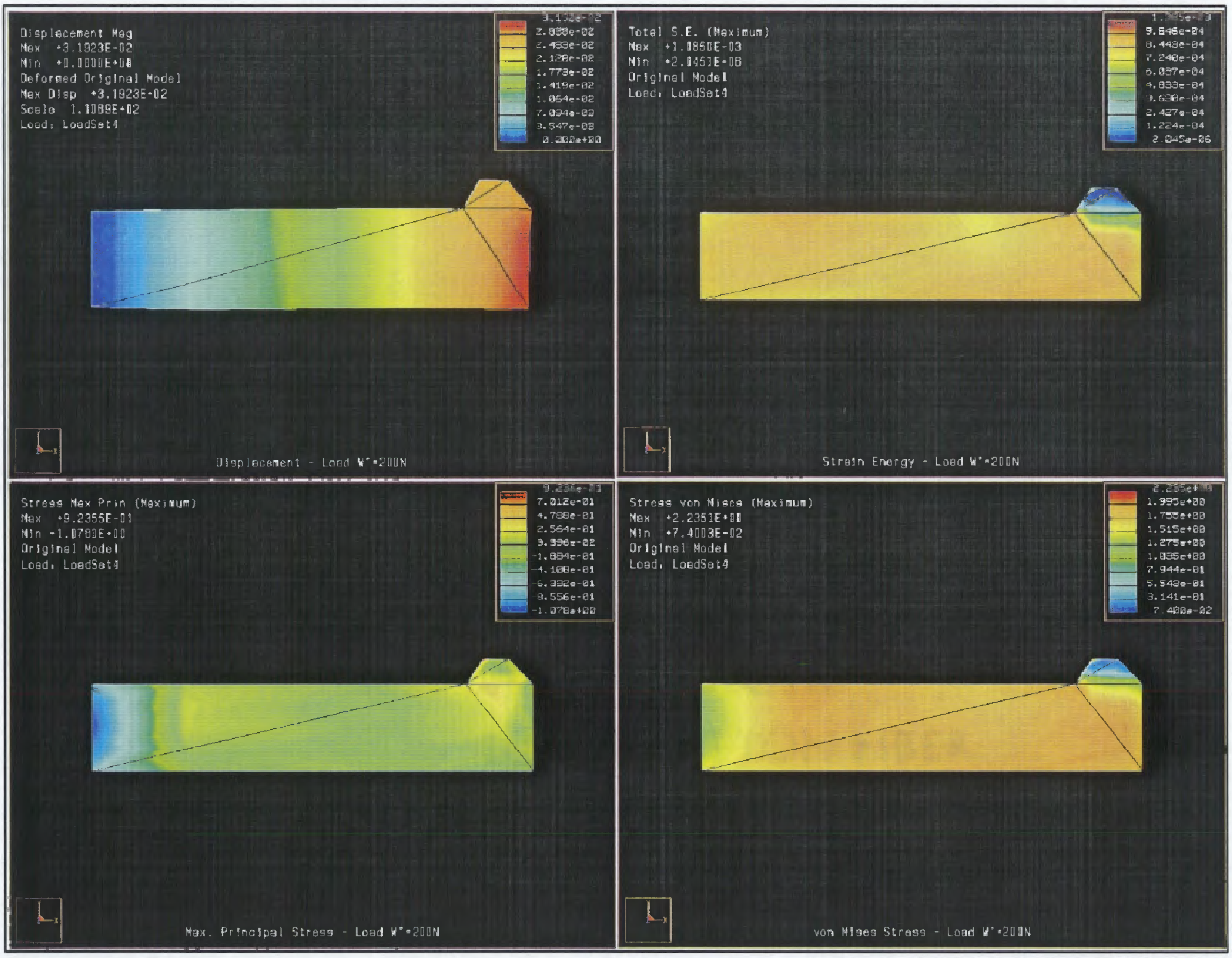


Figure C.7 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load W' of 200N

2. Load: W'=500N

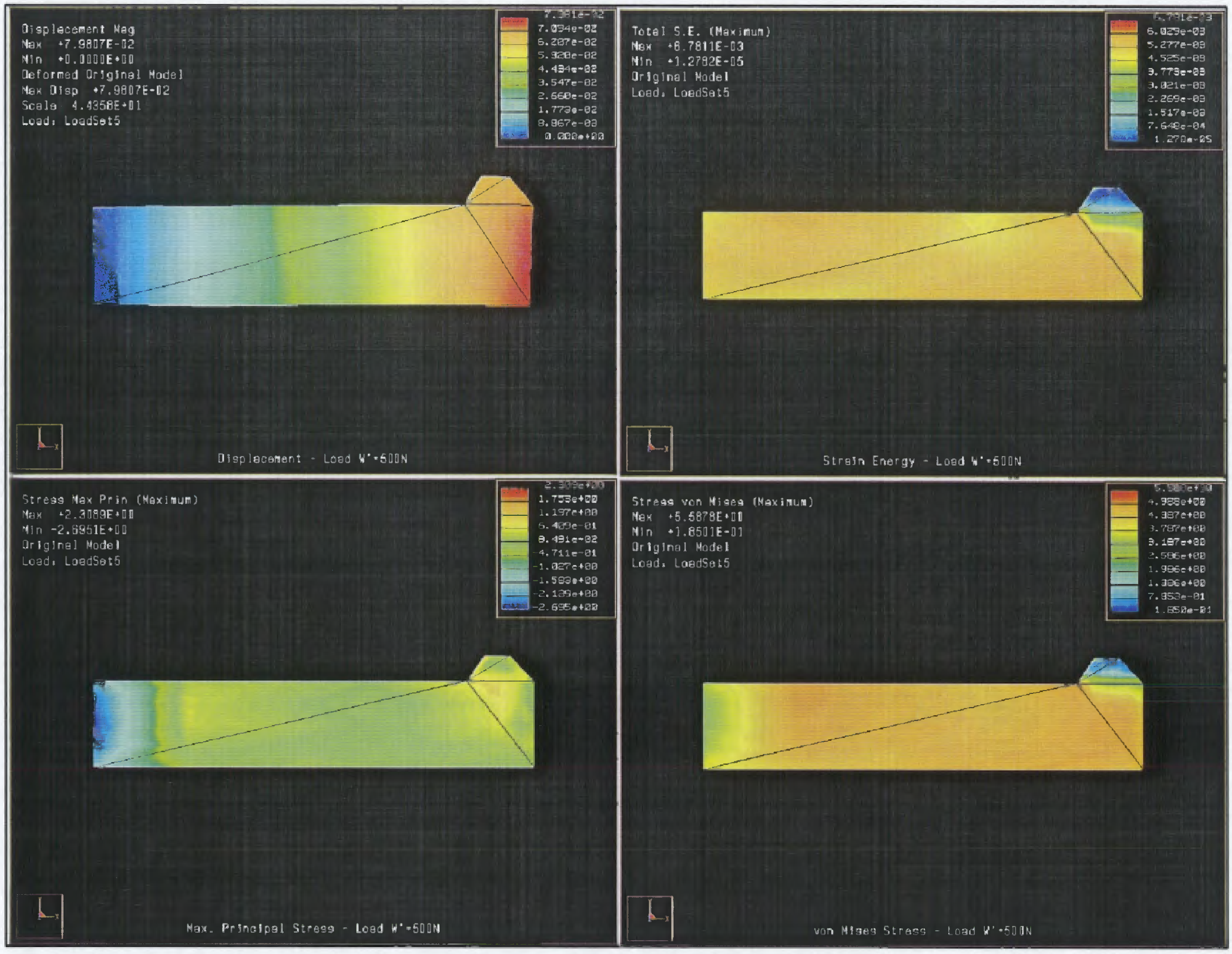


Figure C.8 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load W' of 500N

SOUTHWORTH
Résumé
100% COTTON FIBER

3. Load: W' = 1000N

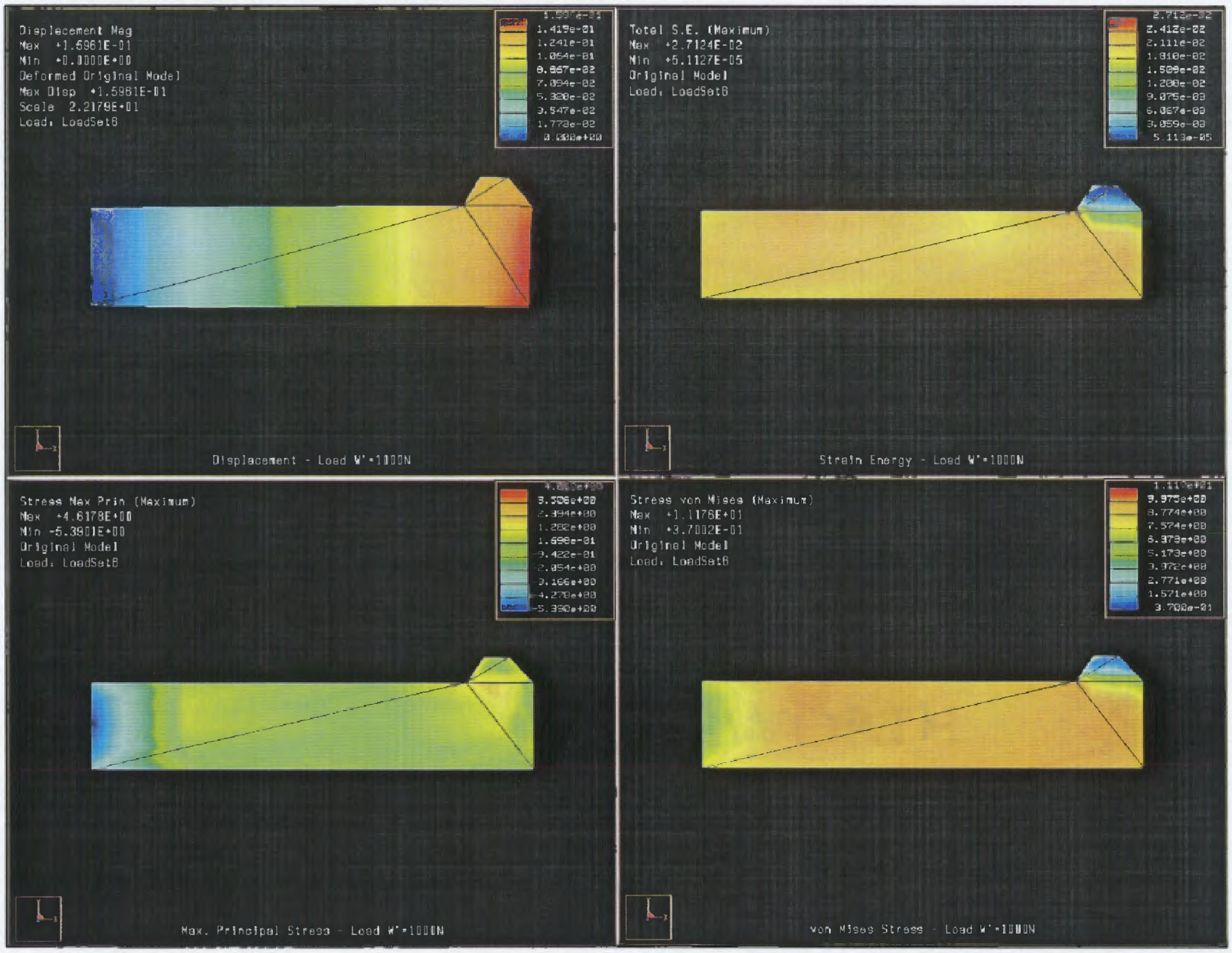


Figure C.9 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load W' of 1000N

4. Load: P = 200N

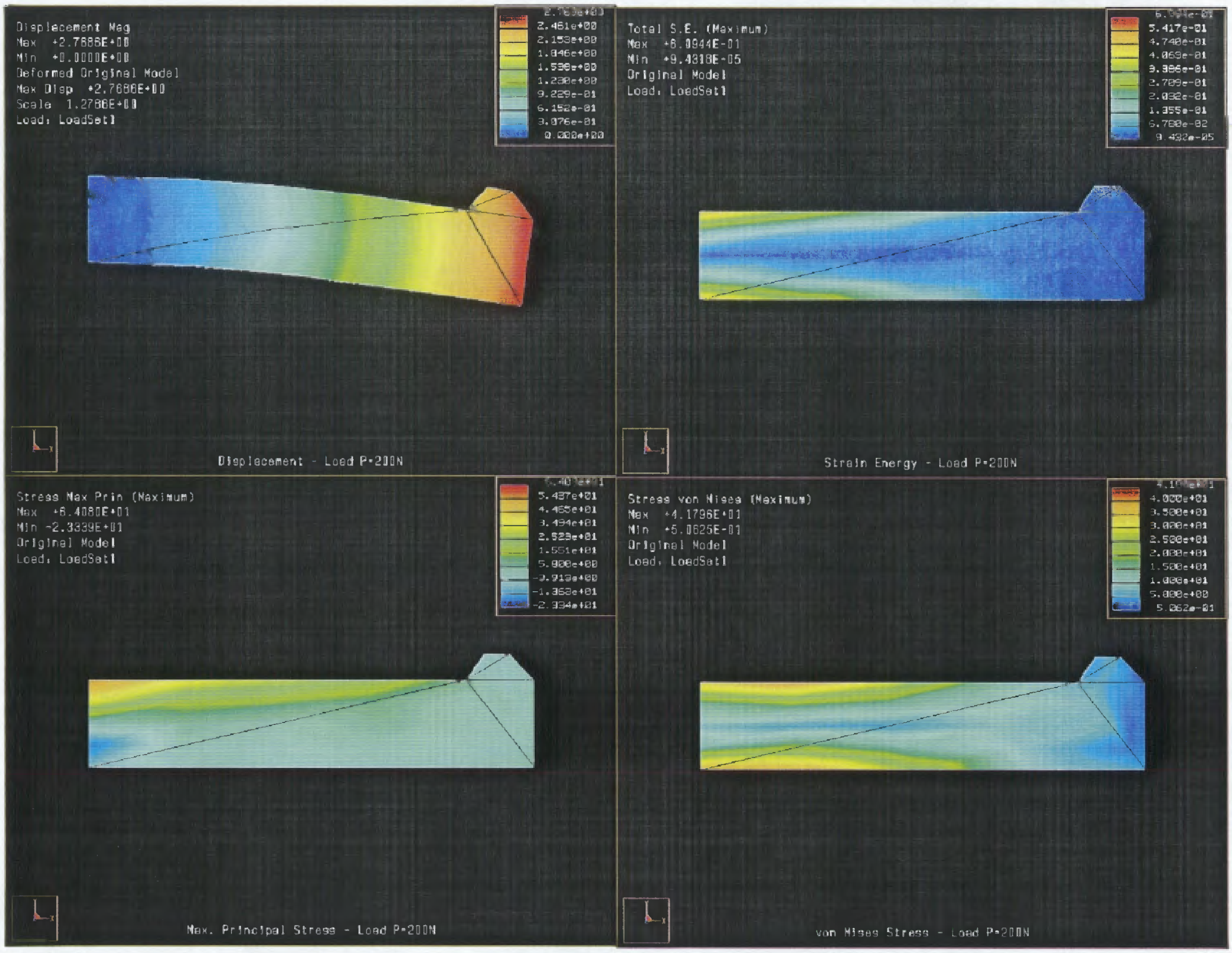


Figure C.10 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load P of 200N

5. Load: P = 500N

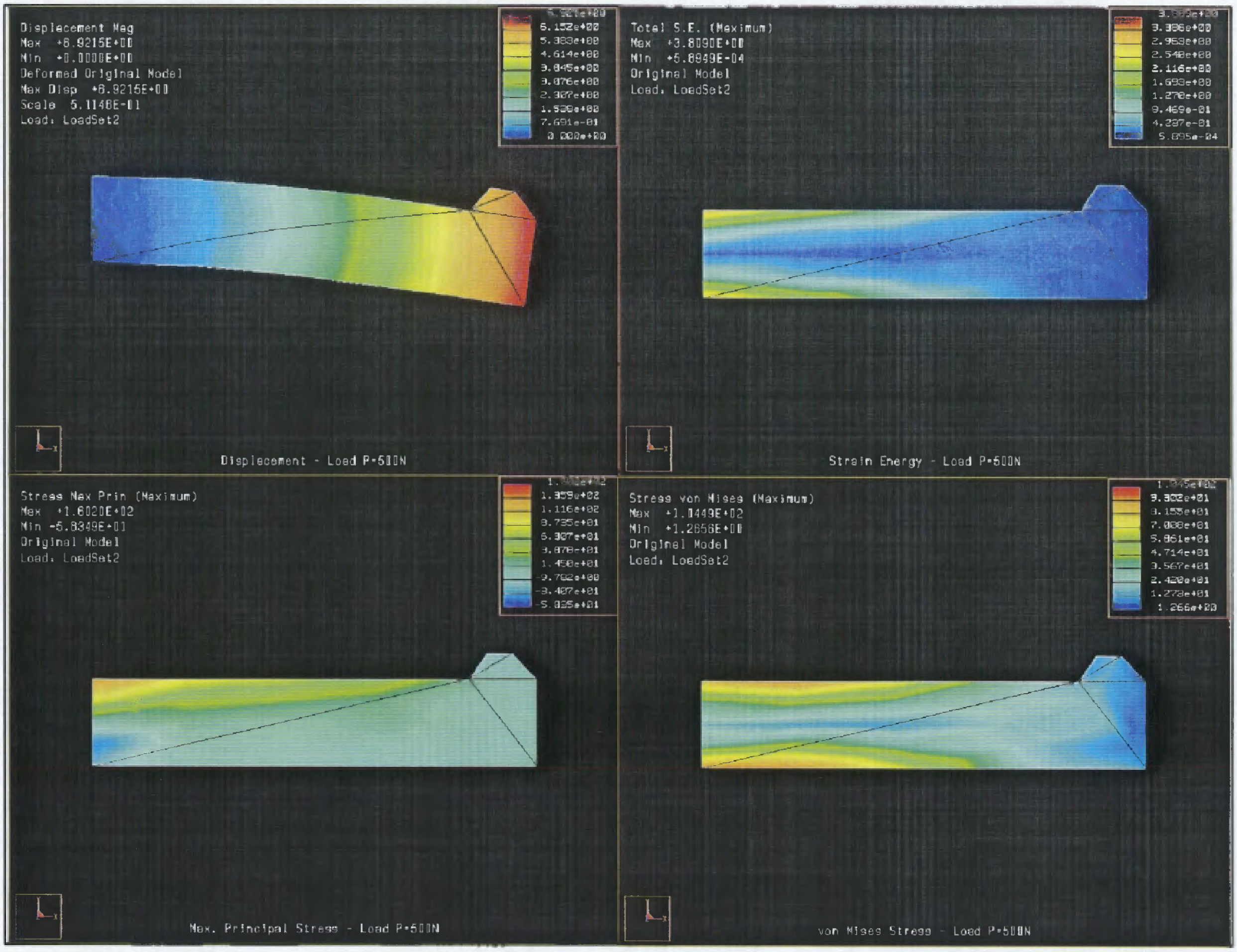


Figure C.11 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load of 500N

6. Load: P = 1000N

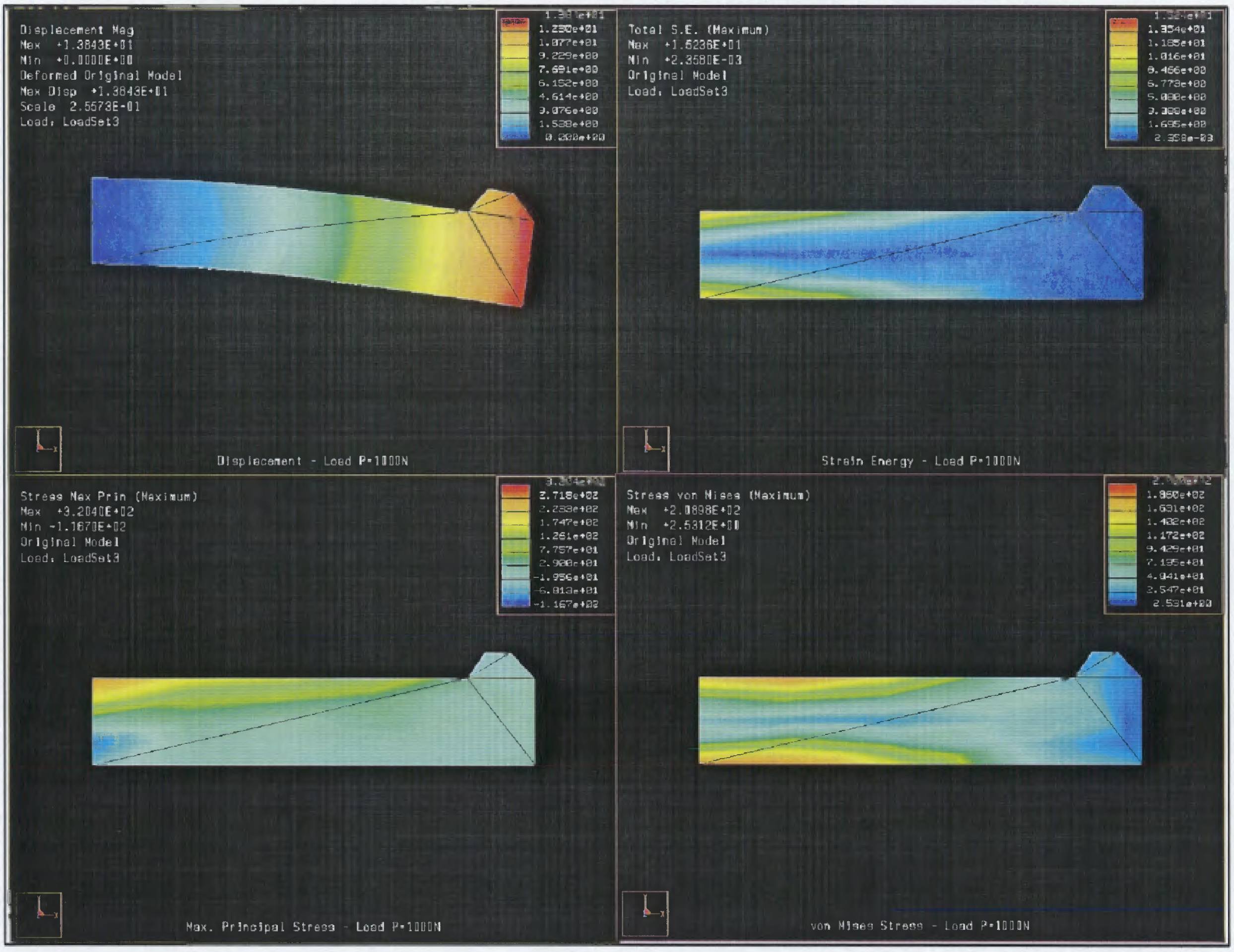


Figure C.12 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load of 1000N

APPENDIX D

RESULTS OF THE PARAMETER STUDY FOR THE CANTILEVER SNAP FIT WITH APPLIED PULLOUT FORCE W' AND PERPENDICULAR FORCE P AS REMOVAL FORCE - FOR A FORCE VALUE OF 25N

- Parameter Study – W'=25N

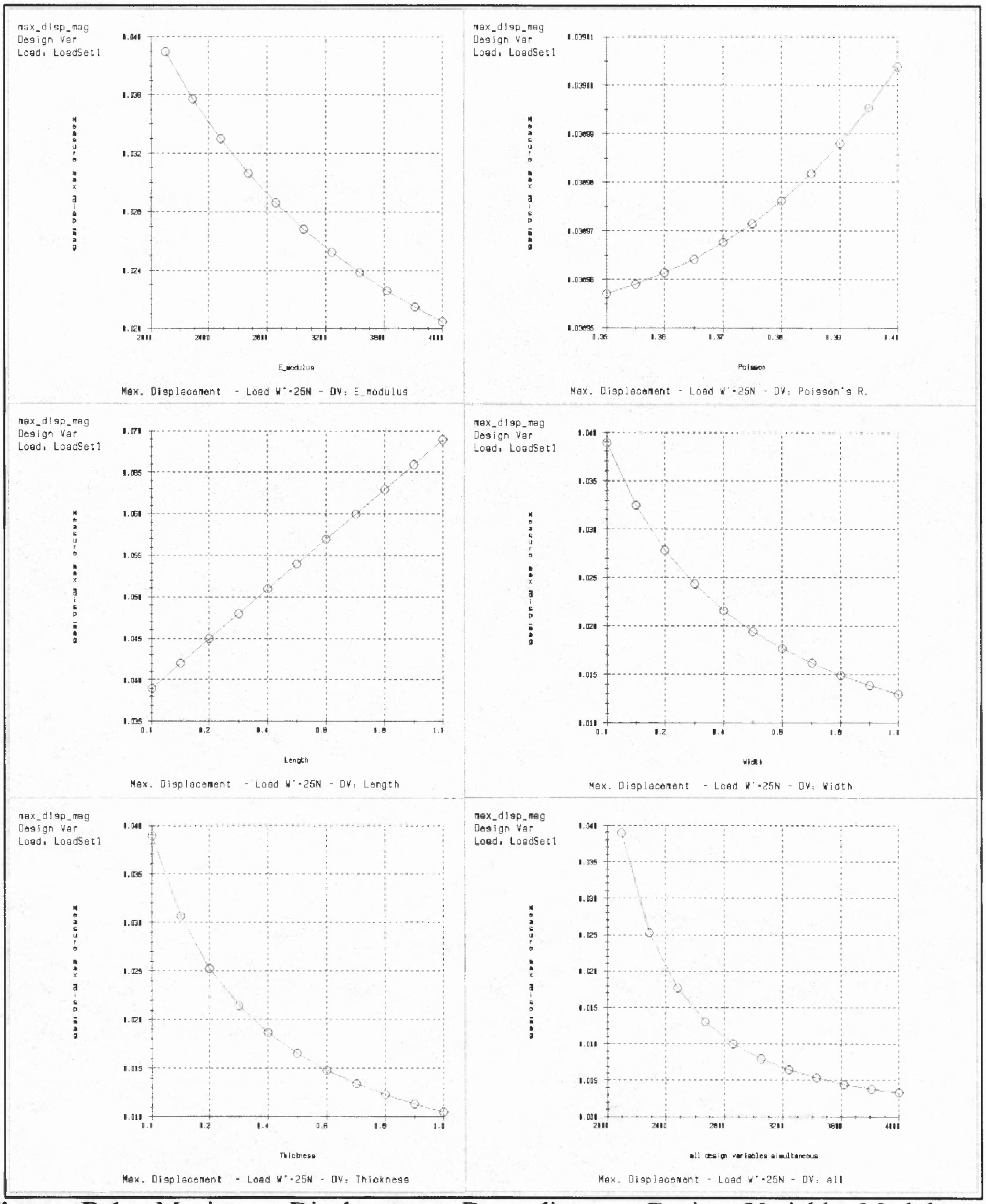


Figure D.1 Maximum Displacement Depending on Design Variable Modulus of Elasticity, Poisson's Ratio, Length, Width, Thickness, and All Design Variables Simultaneous

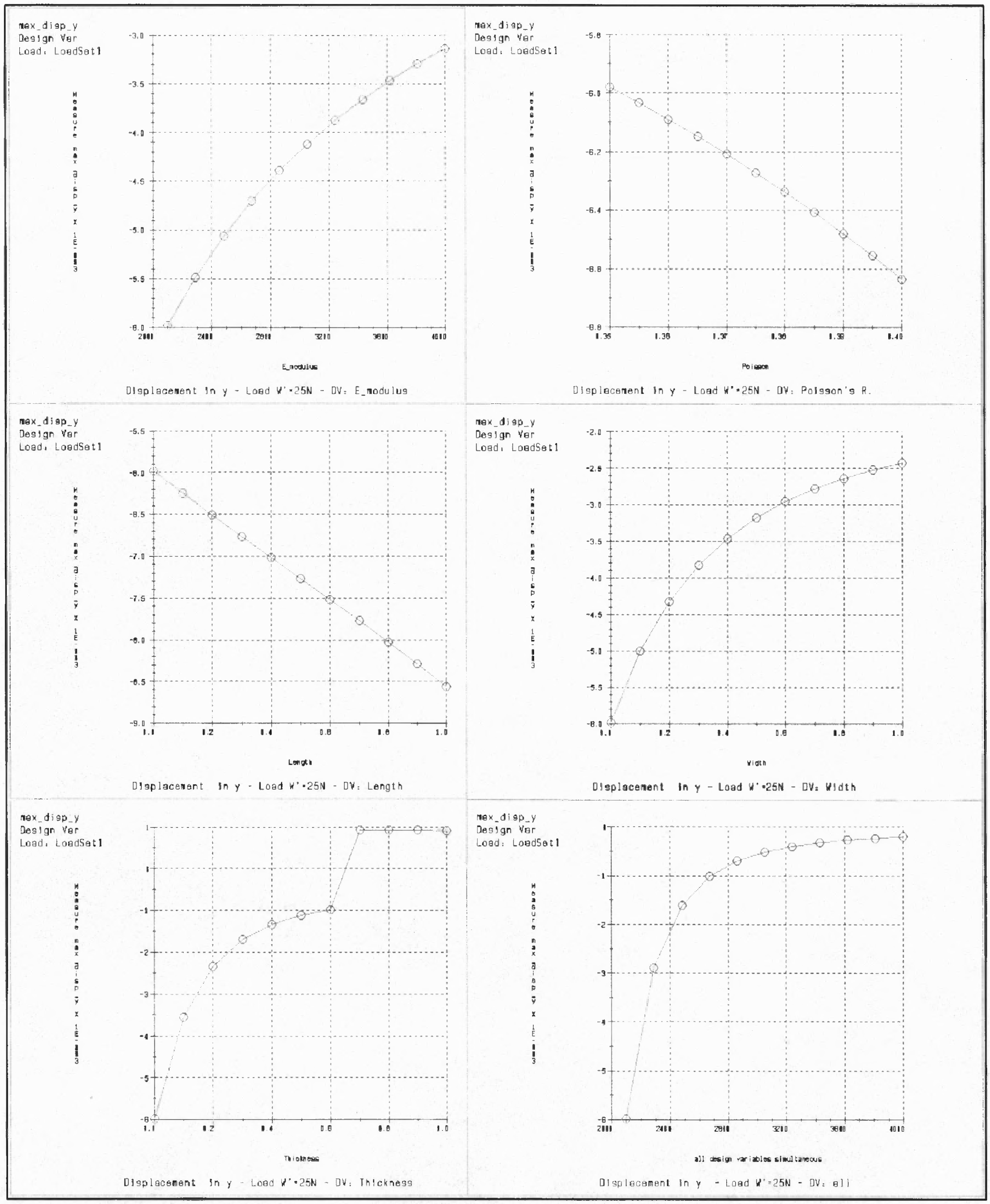


Figure D.2 Displacement in y Depending on Design Variable Modulus of Elasticity, Poisson's Ratio, Length, Width, Thickness, and All Design Variables Simultaneous

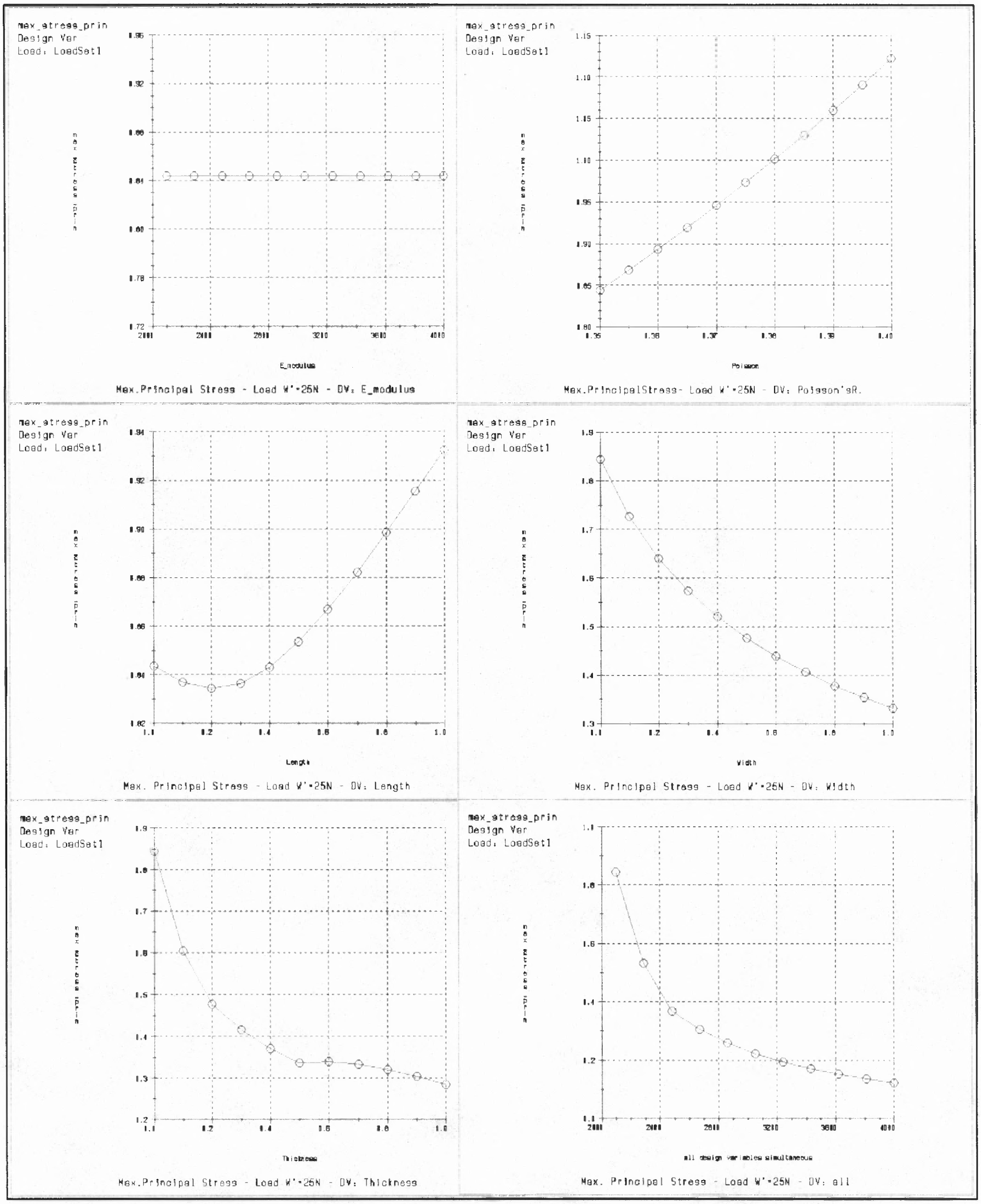


Figure D.3 Maximum Principal Stress Depending on Design Variable Modulus of Elasticity, Poisson's Ratio, Length, Width, Thickness, and All Design Variables Simultaneous

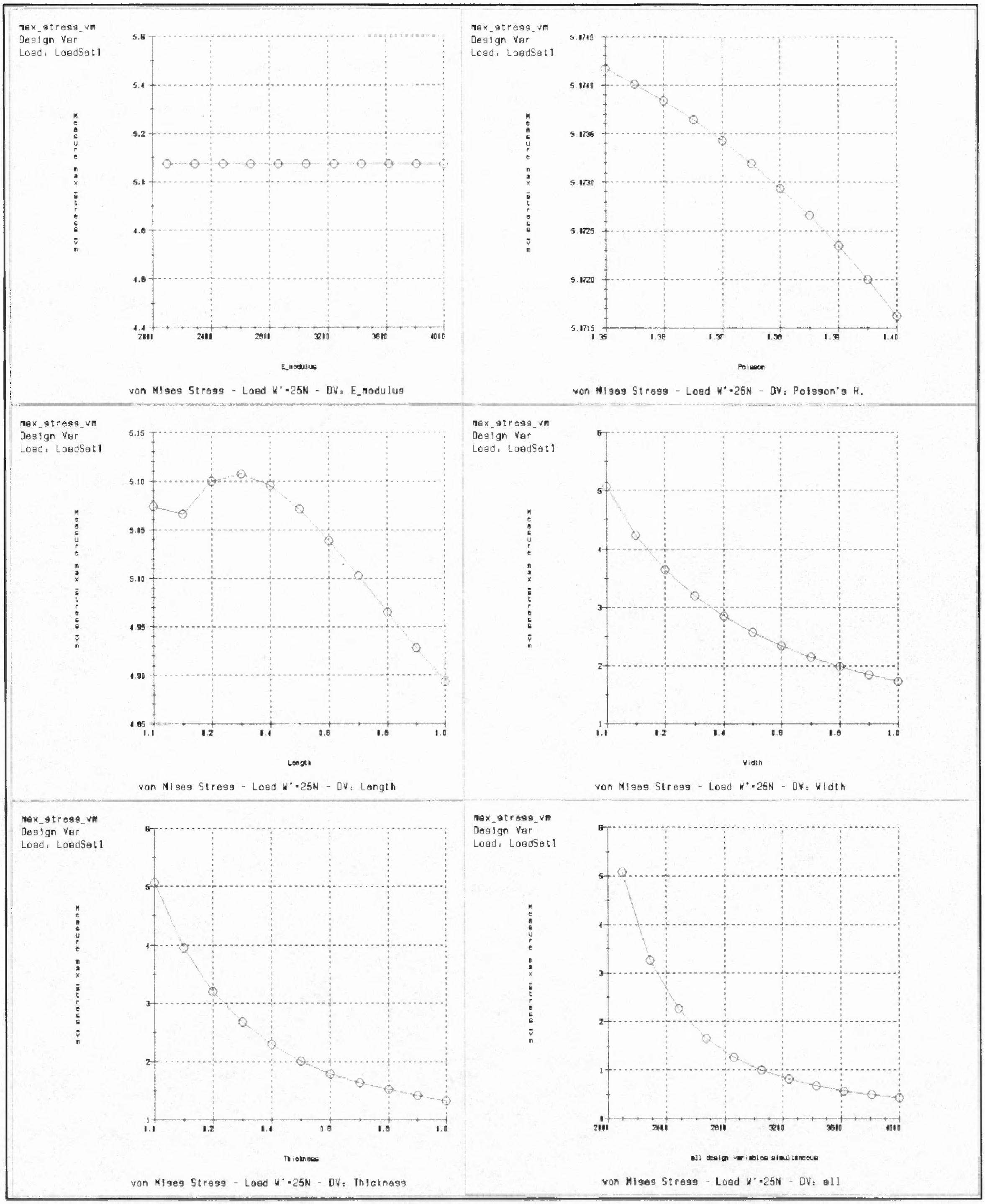


Figure D.4 Von Mises Stress Depending on Design Variable Modulus of Elasticity, Poisson's Ratio, Length, Width, Thickness, and All Design Variables Simultaneous

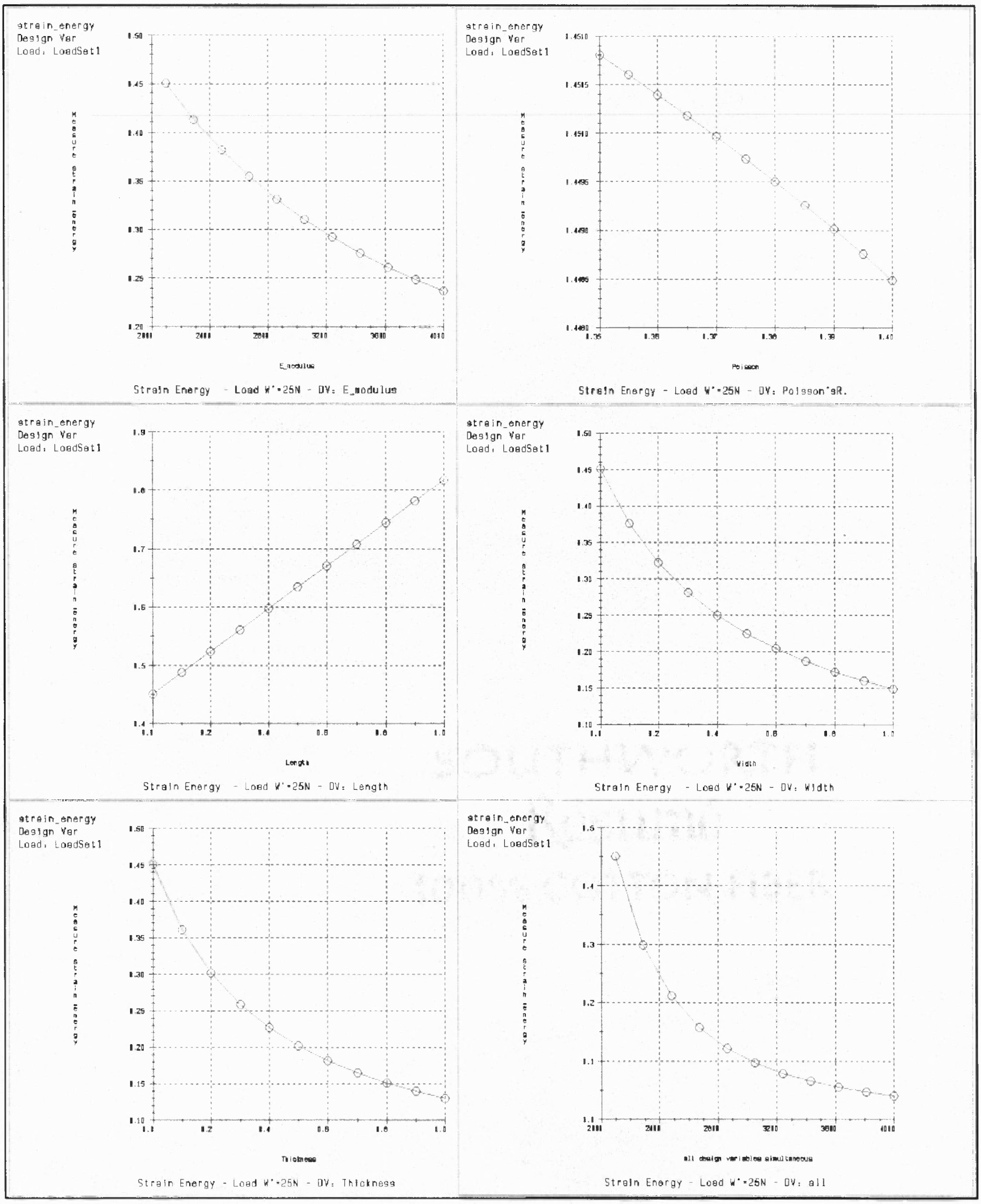


Figure D.5 Strain Energy Depending on Design Variable Modulus of Elasticity, Poisson's Ratio, Length, Width, Thickness, and All Design Variables Simultaneous

♦ Parameter Study – P=25N

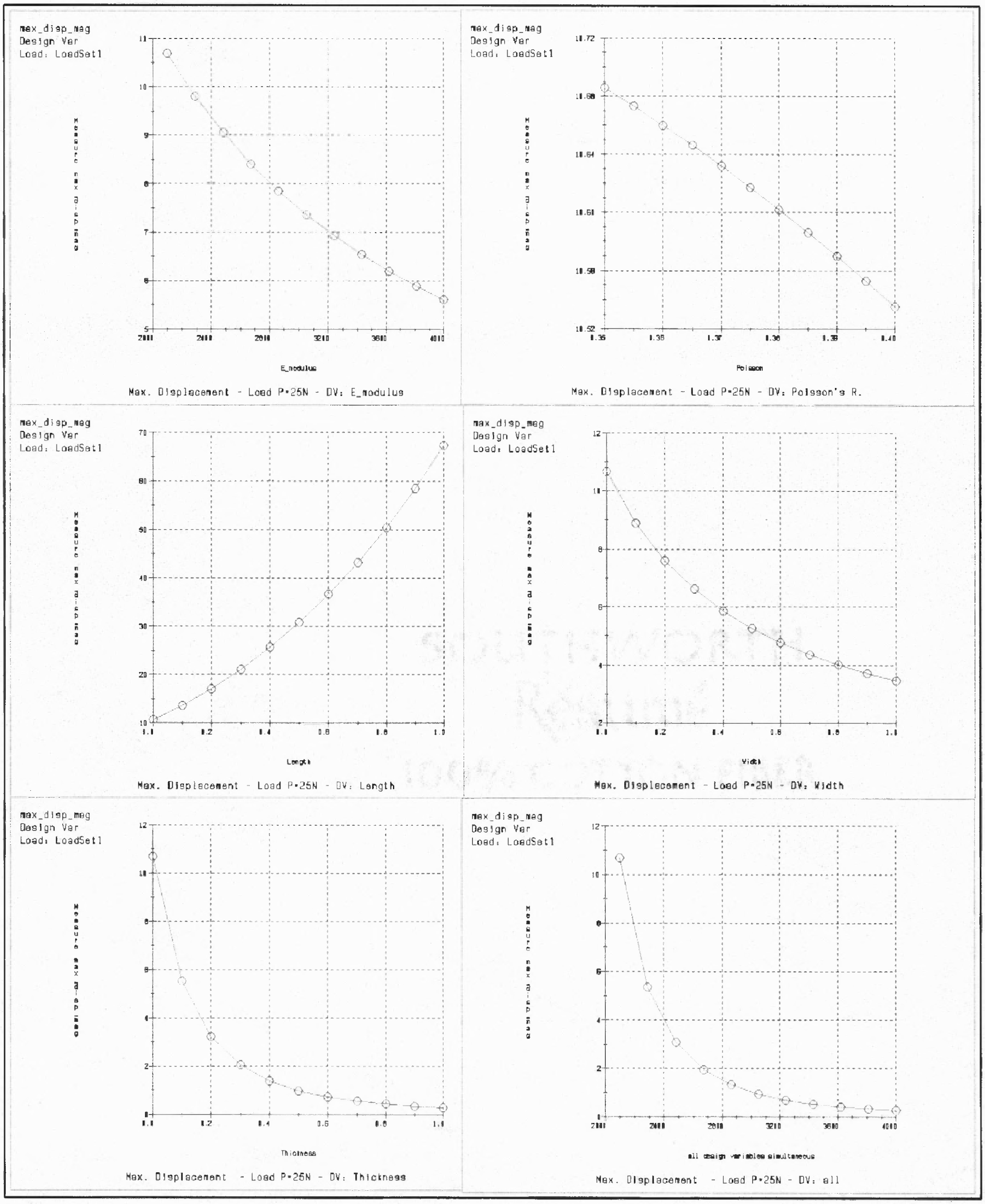


Figure D.6 Maximum Displacement Depending on Design Variable Modulus of Elasticity, Poisson's Ratio, Length, Width, Thickness, and All Design Variables Simultaneous

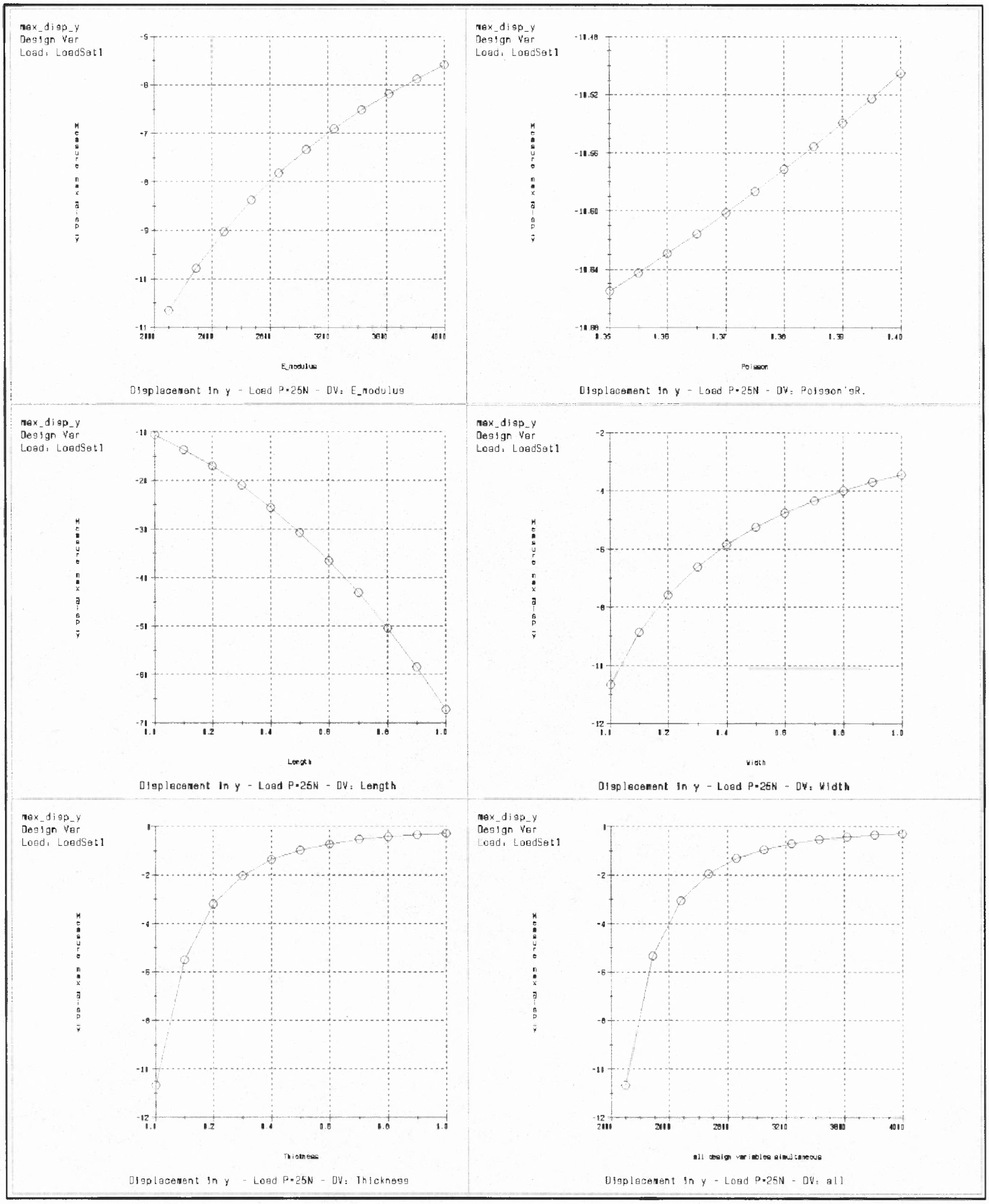


Figure D.7 Displacement in y Depending on Design Variable Modulus of Elasticity, Poisson's Ratio, Length, Width, Thickness, and All Design Variables Simultaneous

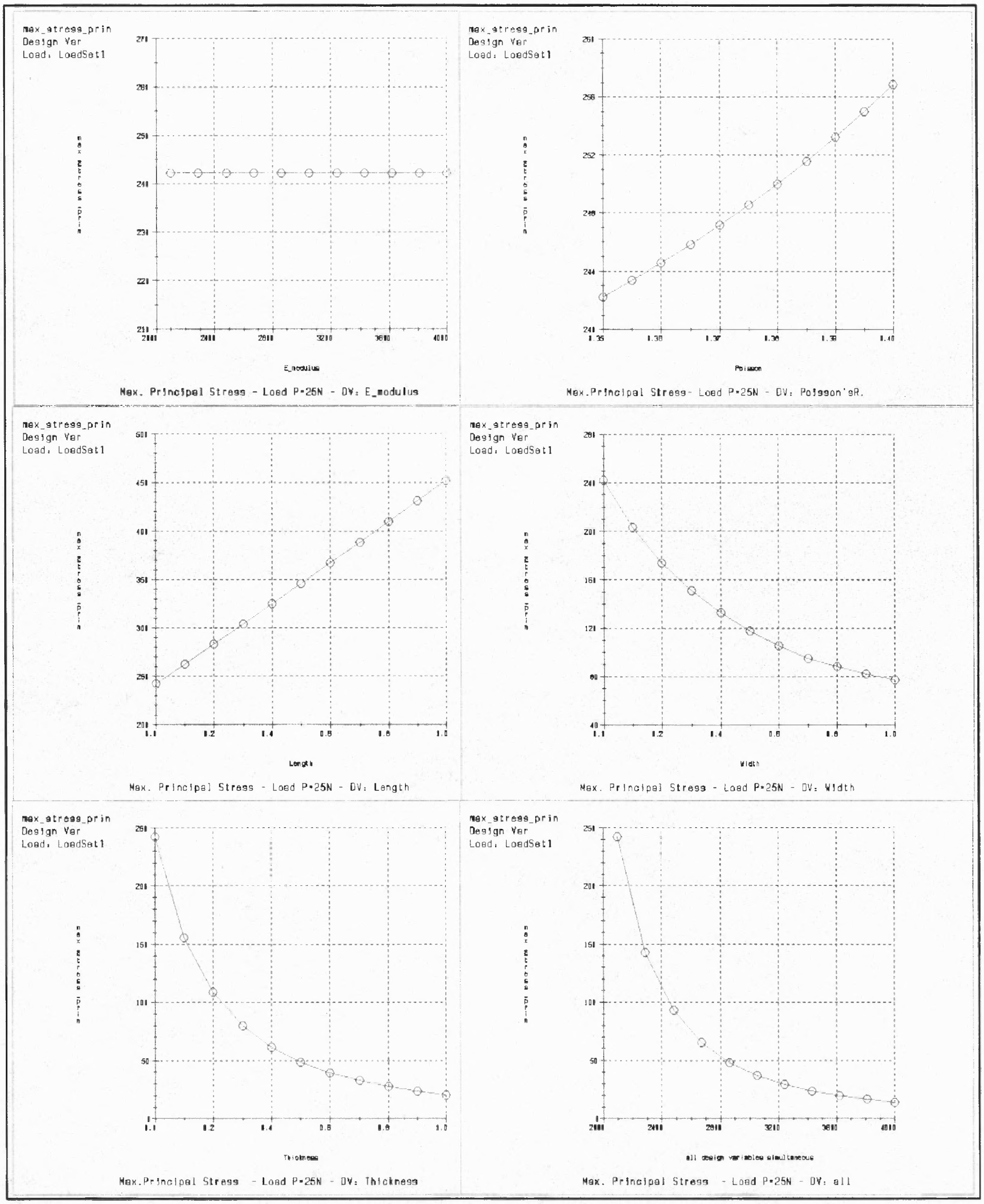


Figure D.8 Maximum Principal Stress Depending on Design Variable Modulus of Elasticity, Poisson's Ratio, Length, Width, Thickness, and All Design Variables Simultaneous

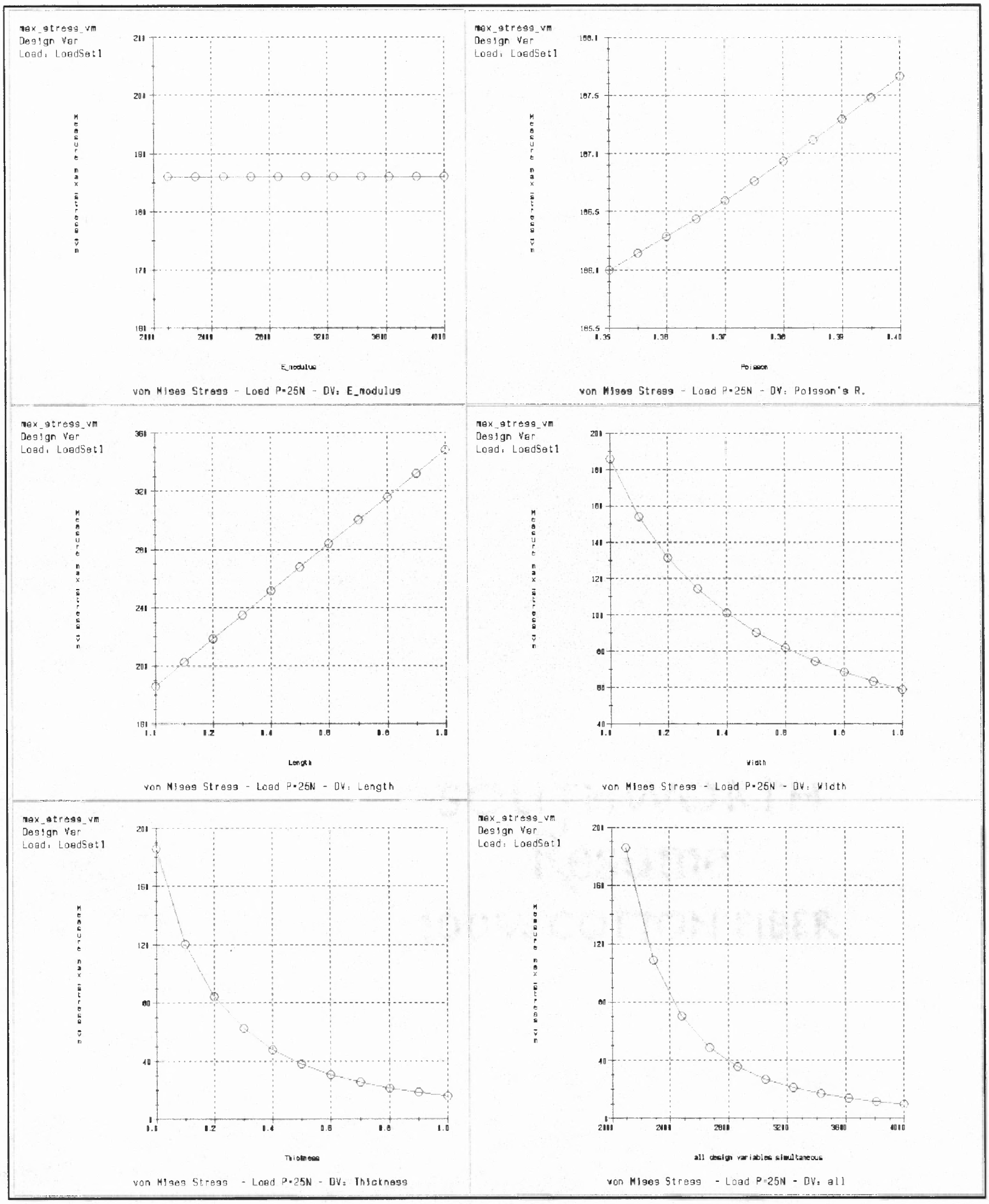


Figure D.9 Von Mises Stress Depending on Design Variable Modulus of Elasticity, Poisson's Ratio, Length, Width, Thickness, and All Design Variables Simultaneous

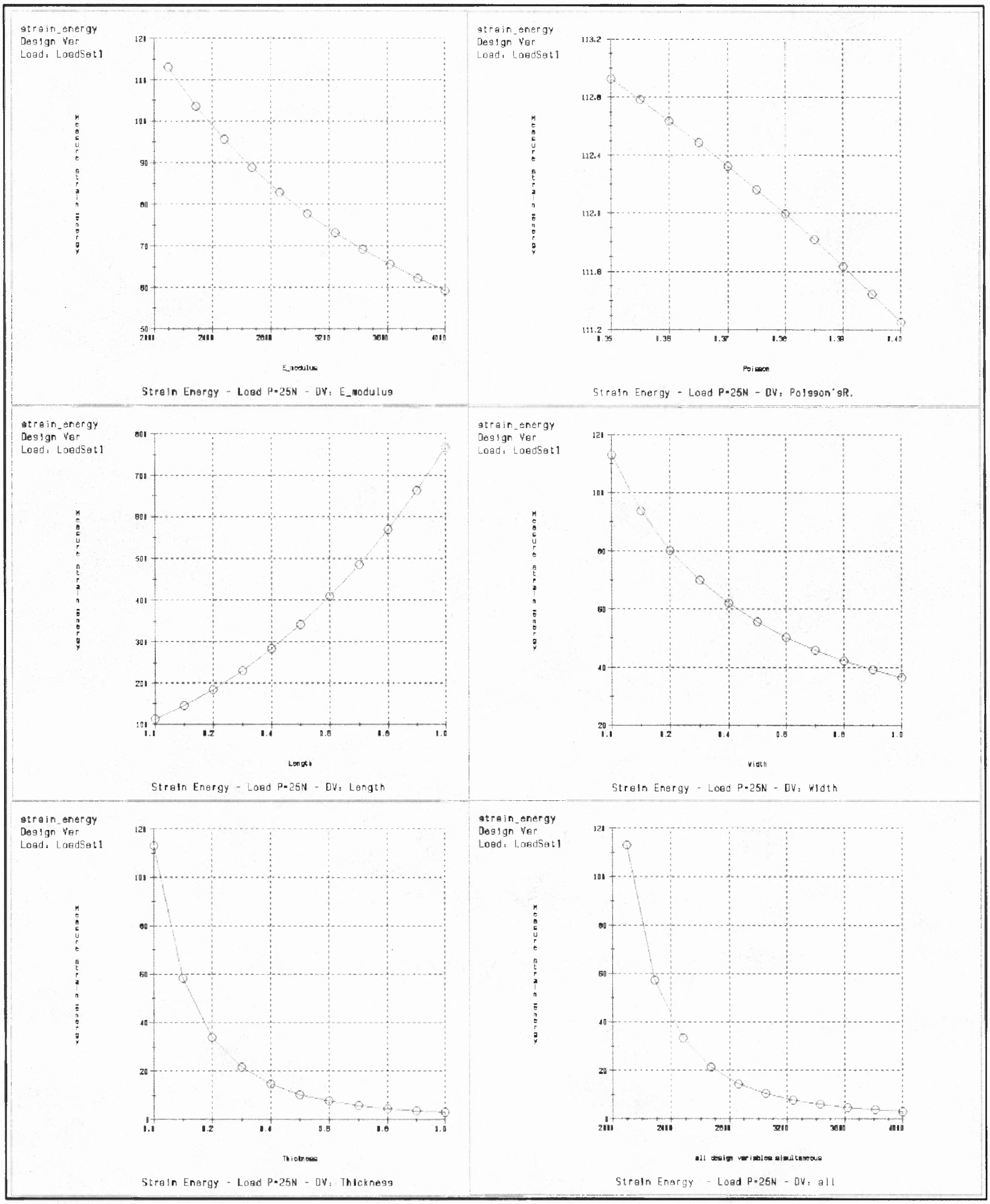


Figure D.10 Strain Energy Depending on Design Variable Modulus of Elasticity, Poisson's Ratio, Length, Width, Thickness, and All Design Variables Simultaneous

APPENDIX E

RESULTS OF THE STATIC ANALYSIS FOR THE CYLINDRICAL SNAP FIT WITH APPLIED PULLOUT FORCE W' AS REMOVAL FORCE FOR THE SHAFT AND RESULTING FORCE F FOR THE TUBE

Shaft - Case I

1. Load: W' = 100N

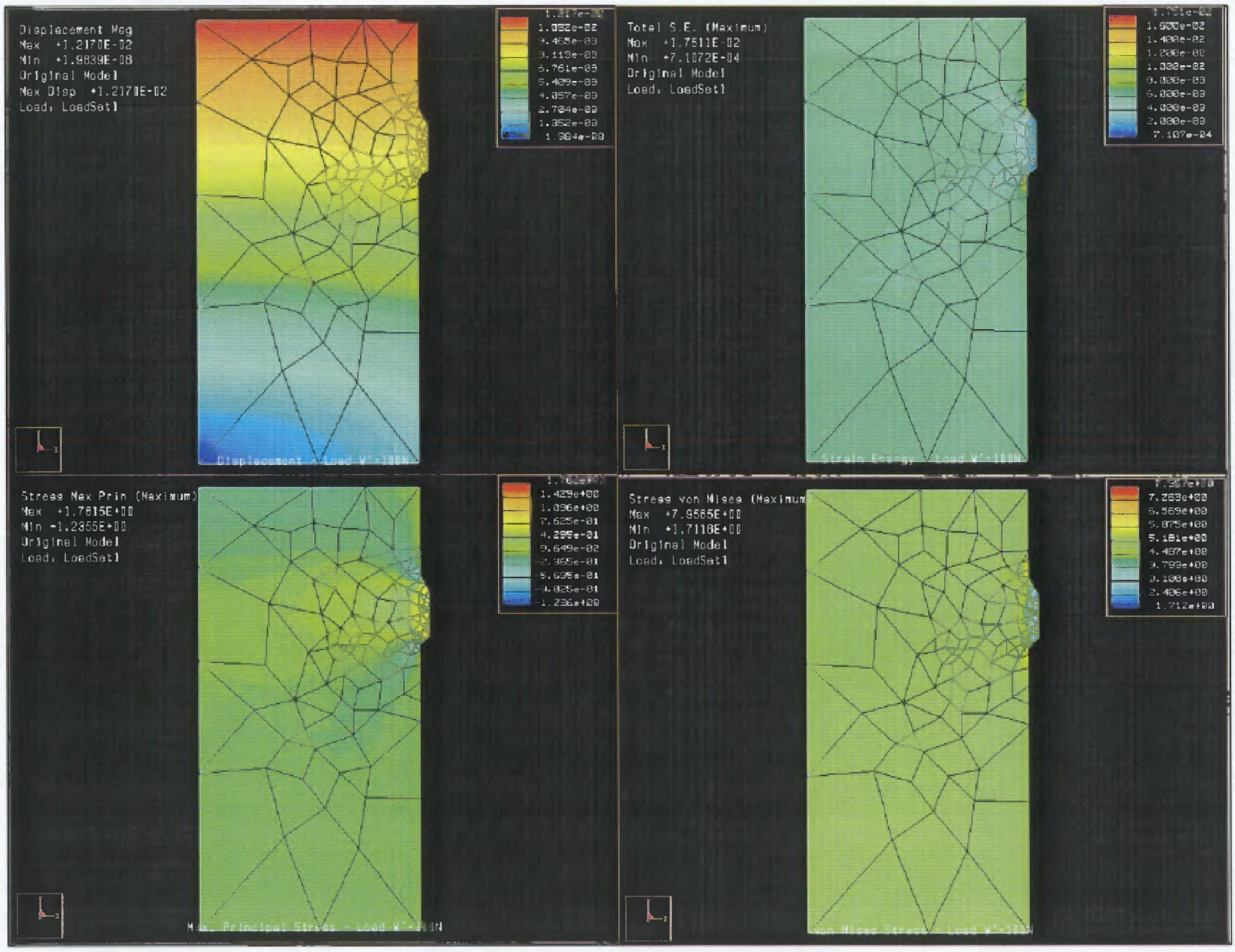


Figure E.1 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load W' of 100N

Shaft - Case I

2. Load: W' = 200N

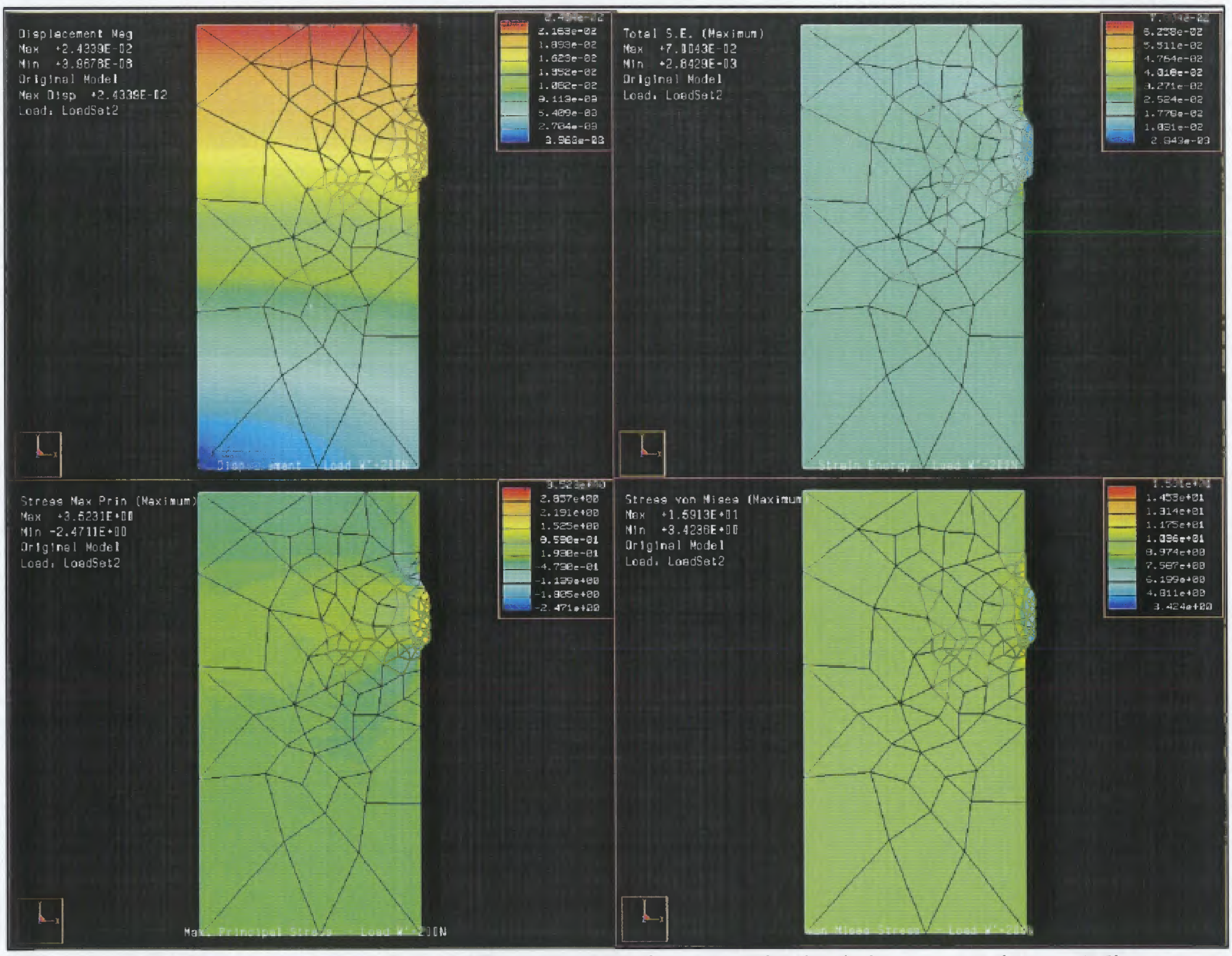


Figure E.2 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load W' of 200N

Shaft - Case I

3. Load: W' = 500N

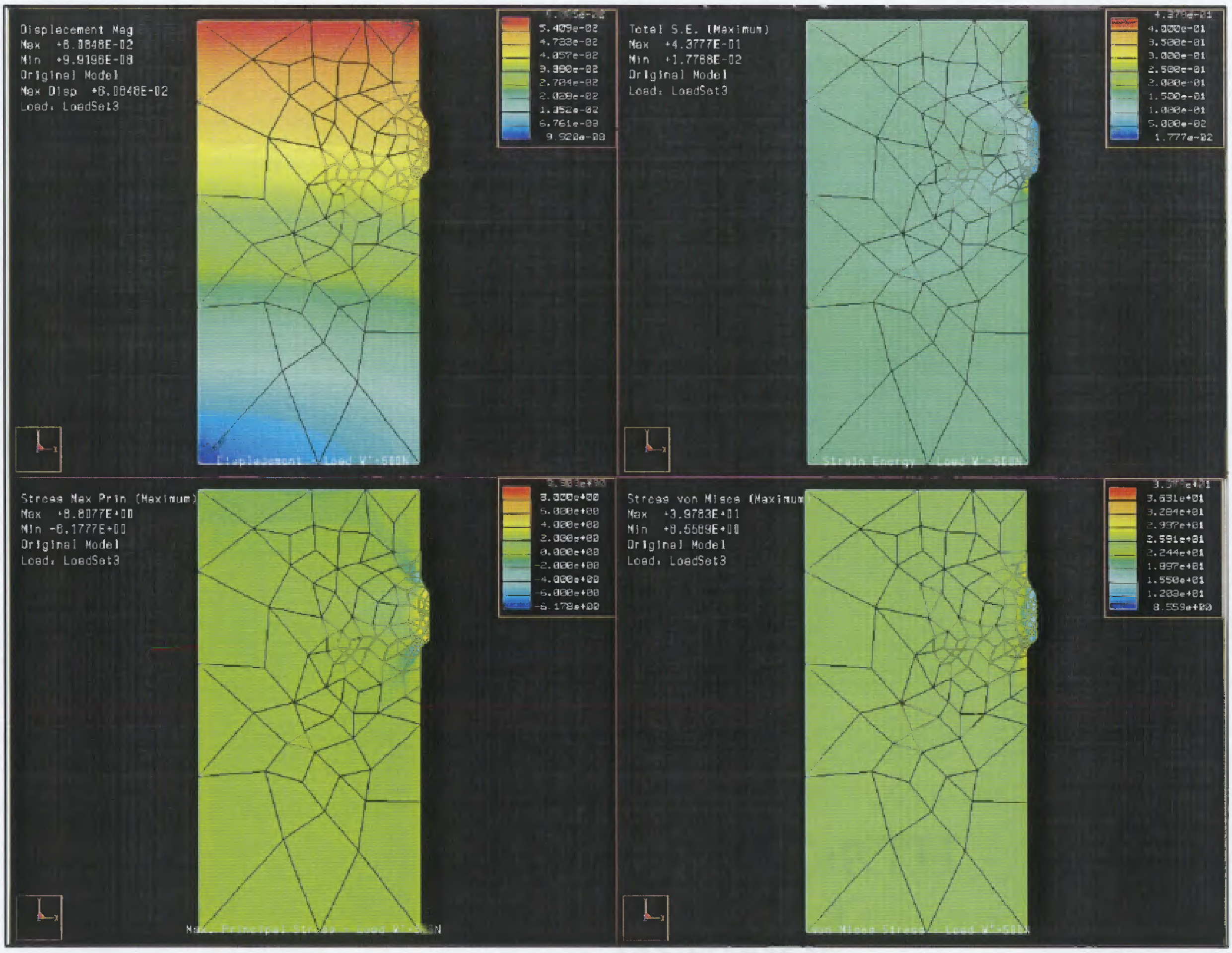


Figure E.3 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load W' of 500N

Shaft - Case I

4. Load: W' = 1000N

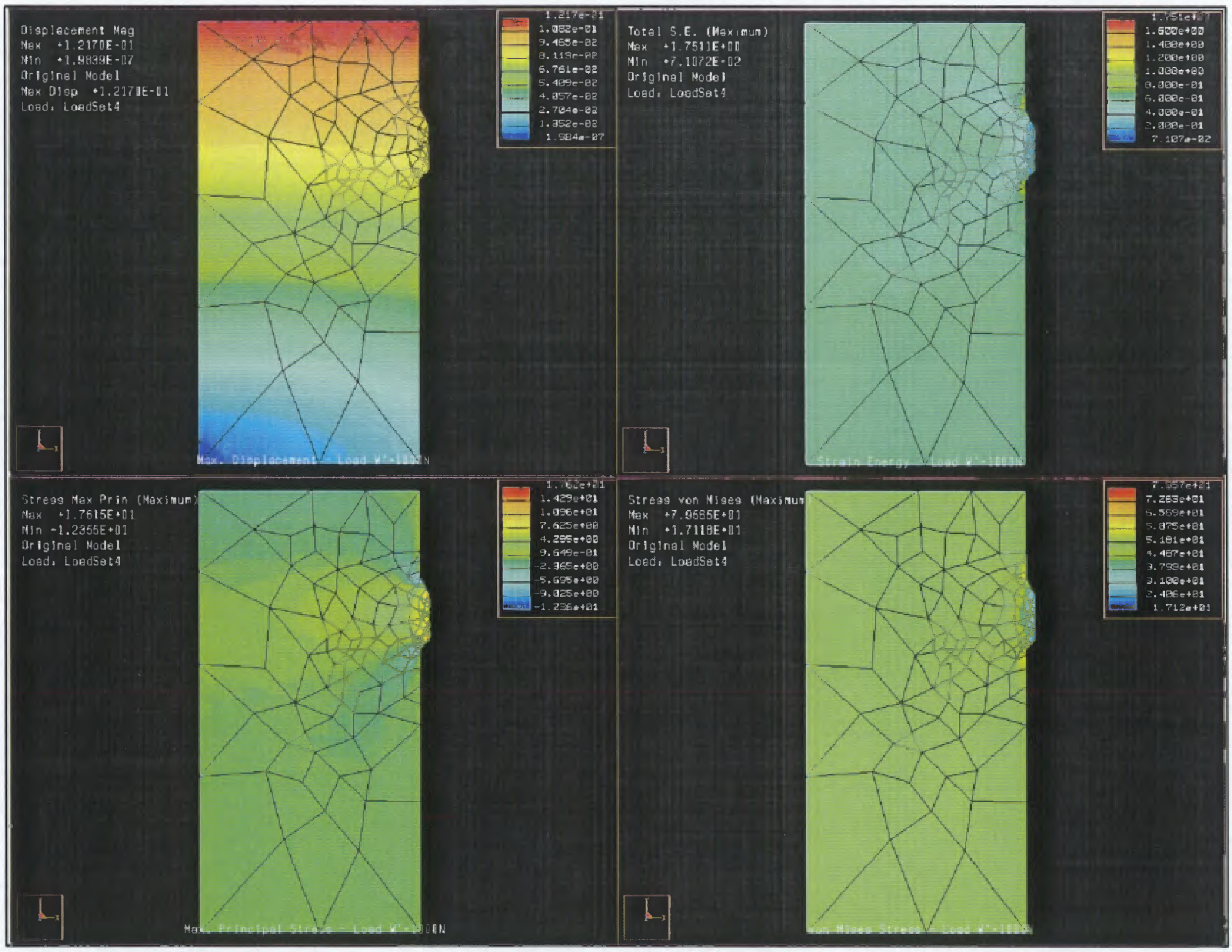


Figure E.4 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load W' of 1000N

Shaft - Case I

5. Load: W’ = 2000N

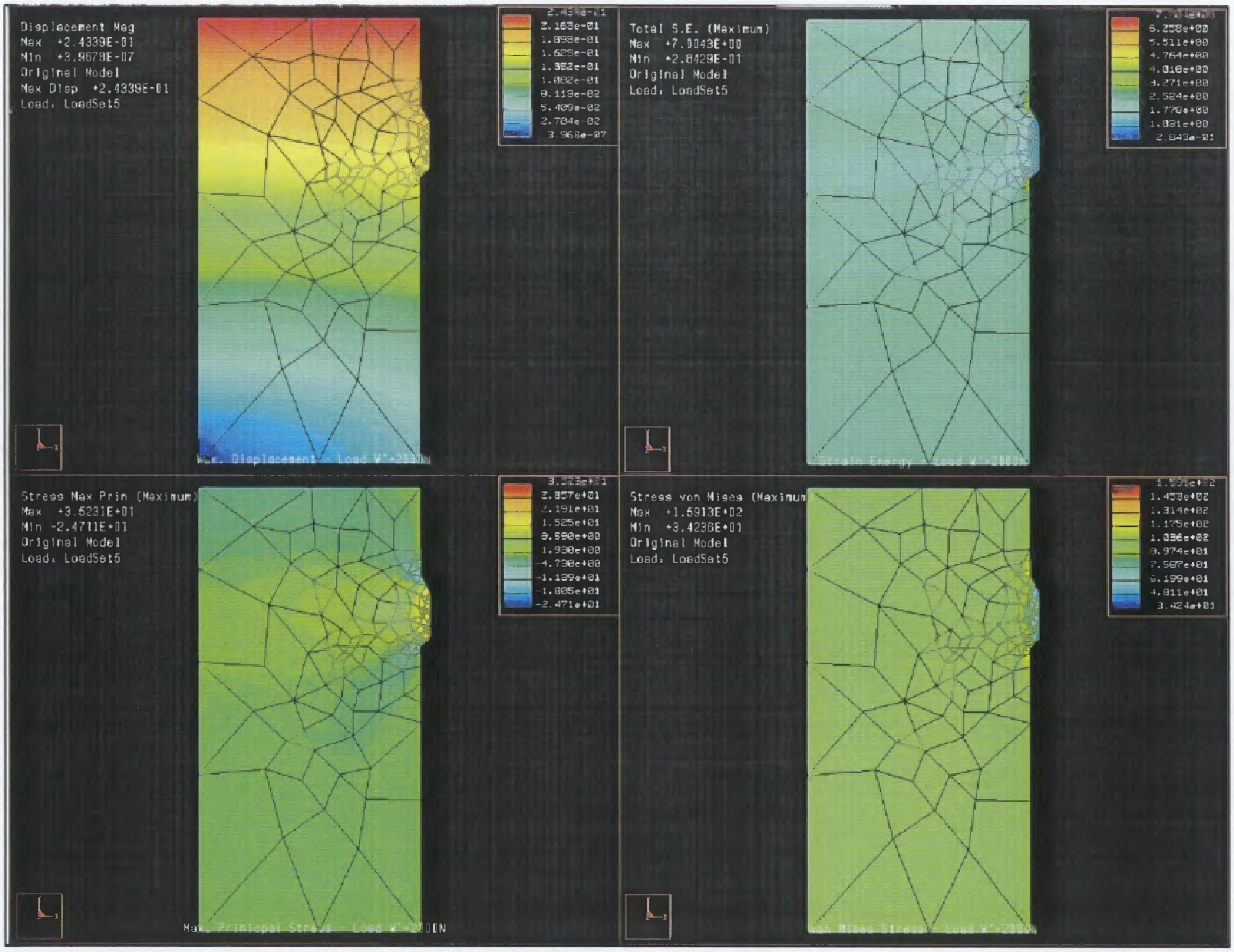


Figure E.5 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load W’ of 2000N

Tube - Case I

1. Load: F = 100N

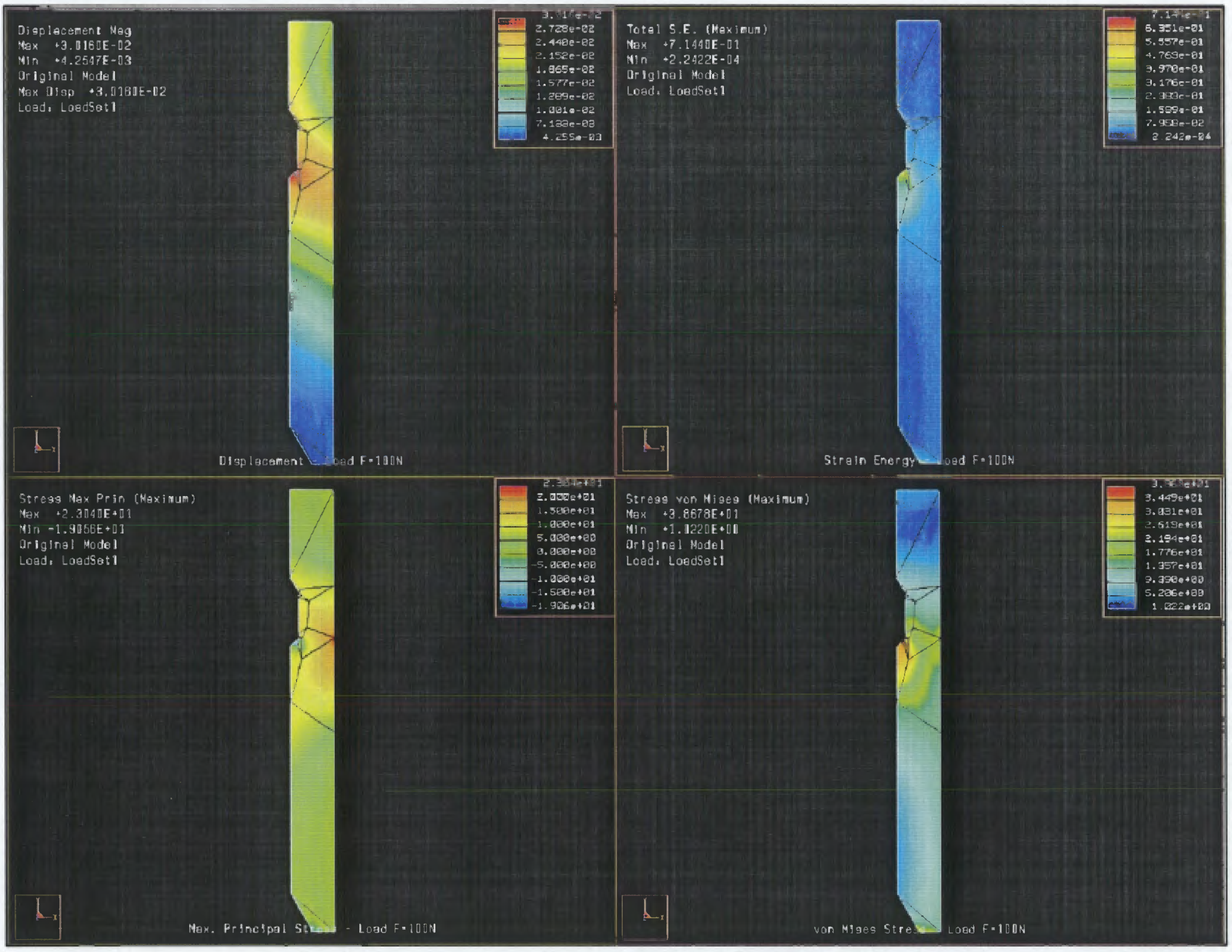


Figure E.6 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load F of 100N

Tube - Case I

2. Load: F = 200N

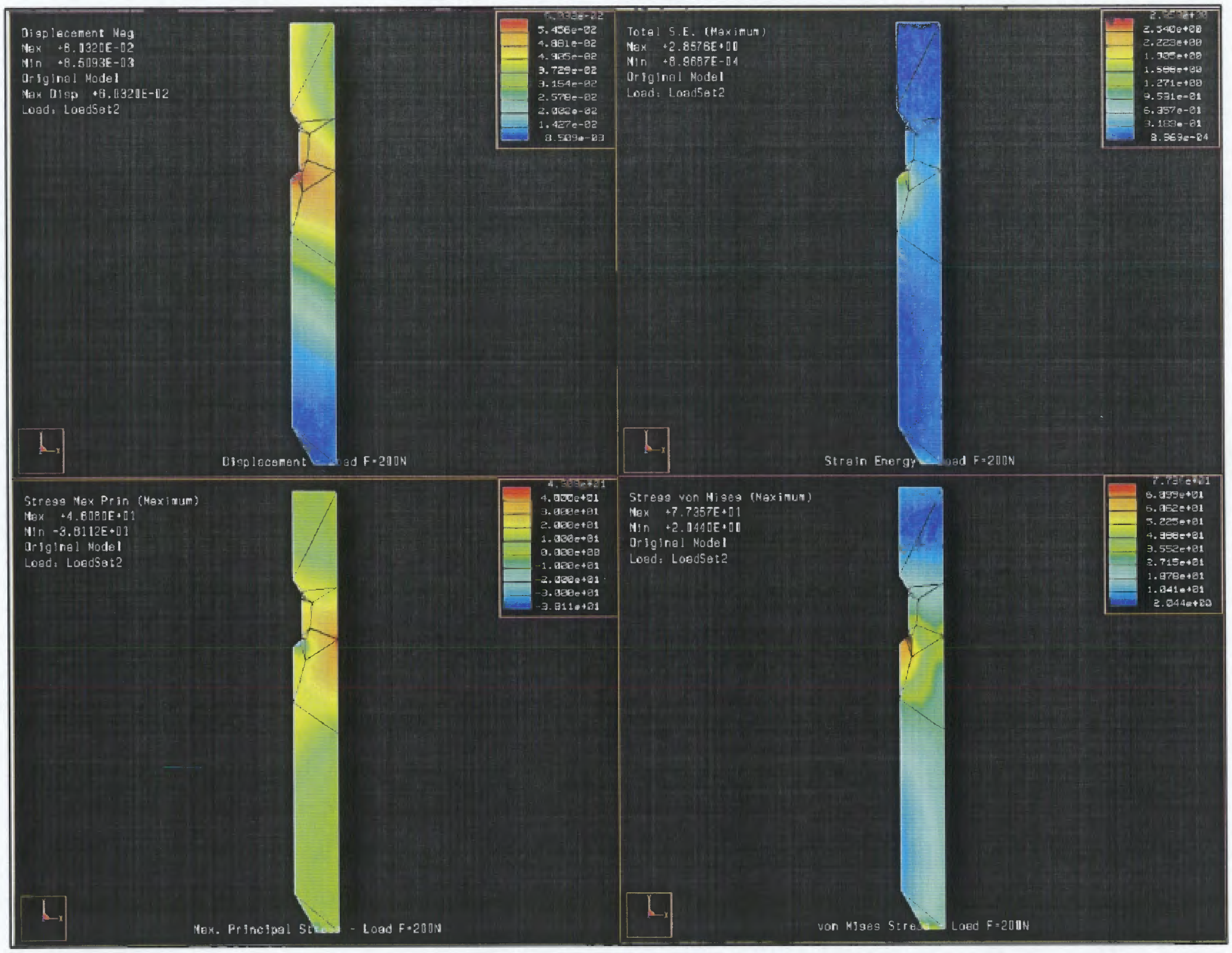


Figure E.7 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load F of 200N

Tube - Case I

3. Load: F = 500N

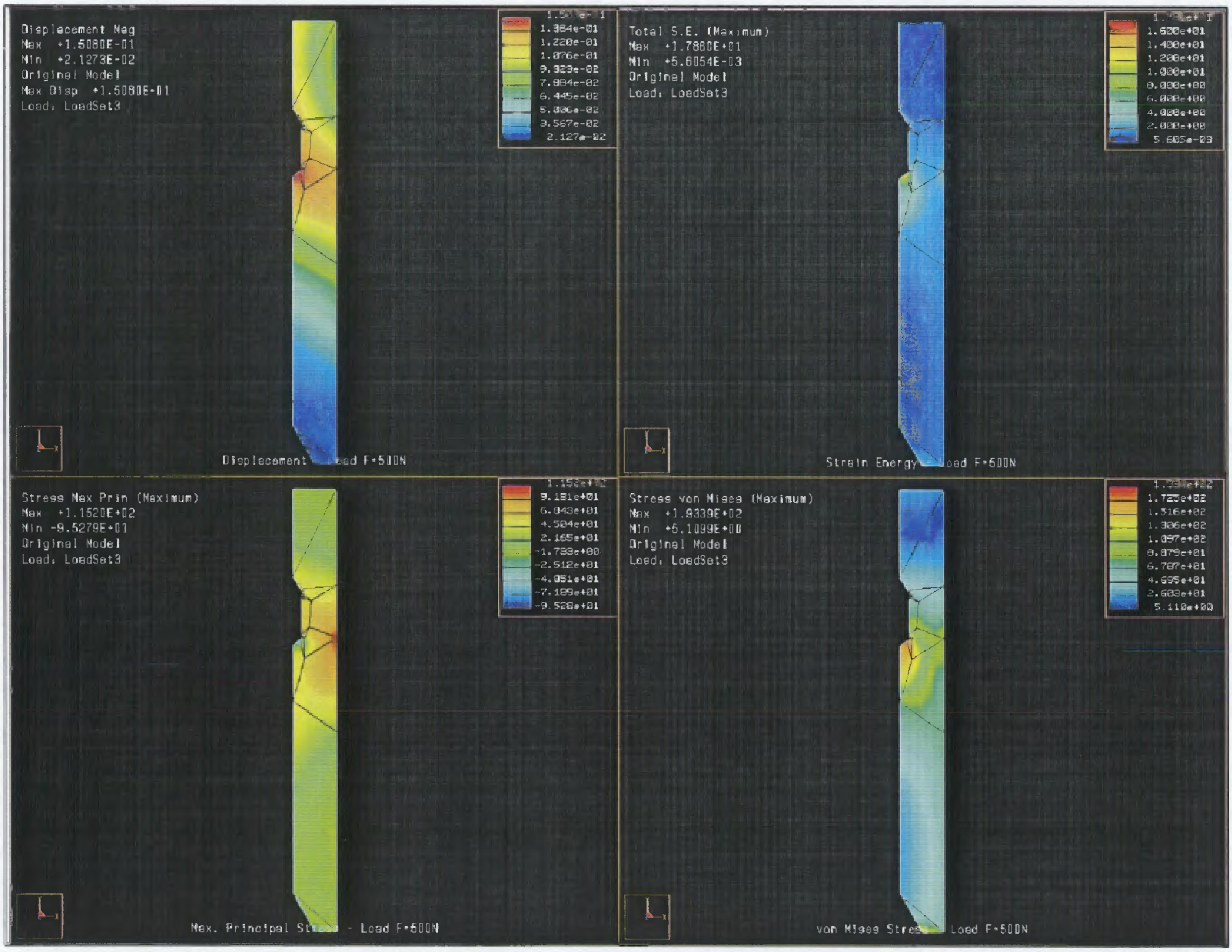


Figure E.8 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load F of 500N

Tube - Case I

4. Load: F = 1000N

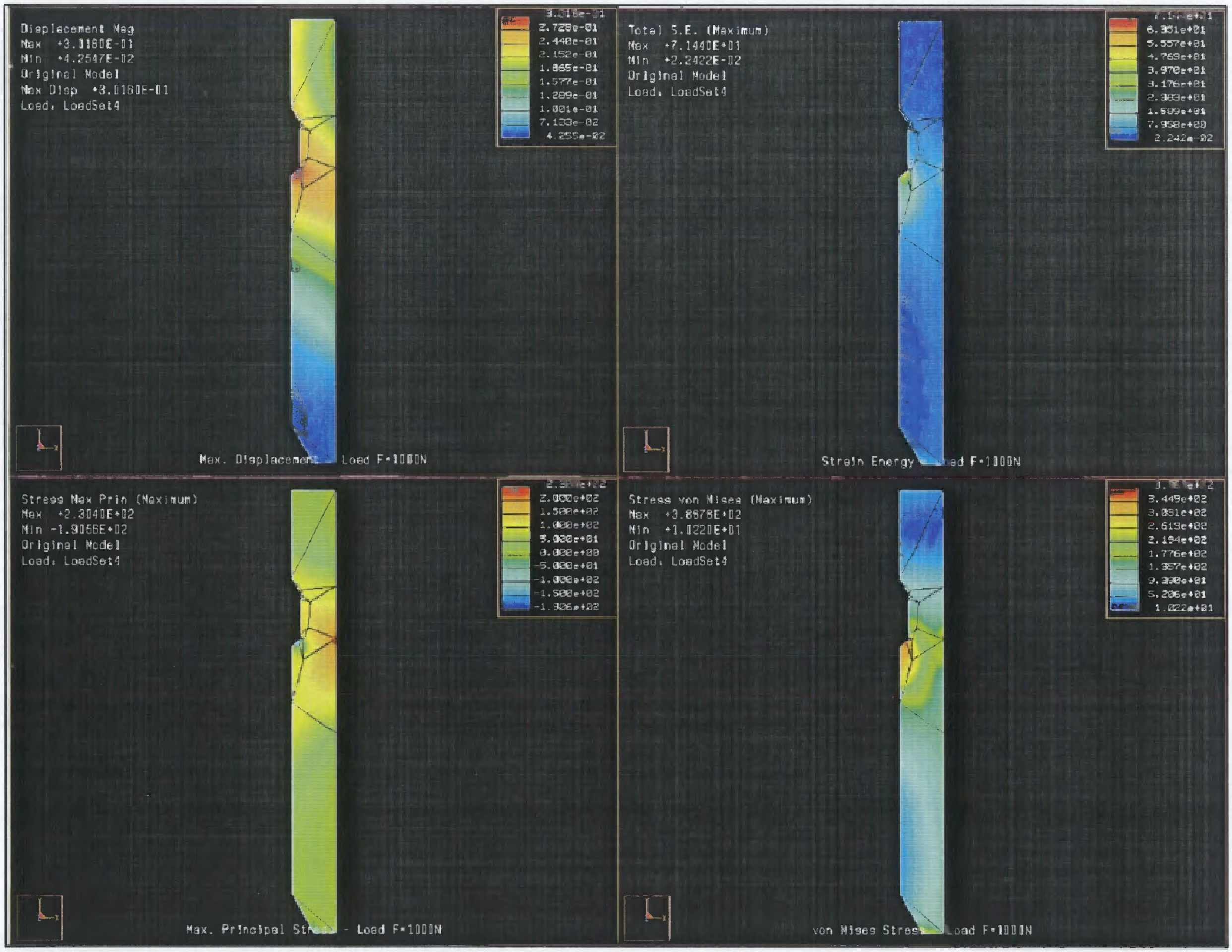


Figure E.9 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load F of 1000N

Tube - Case I

5. Load: F = 2000N

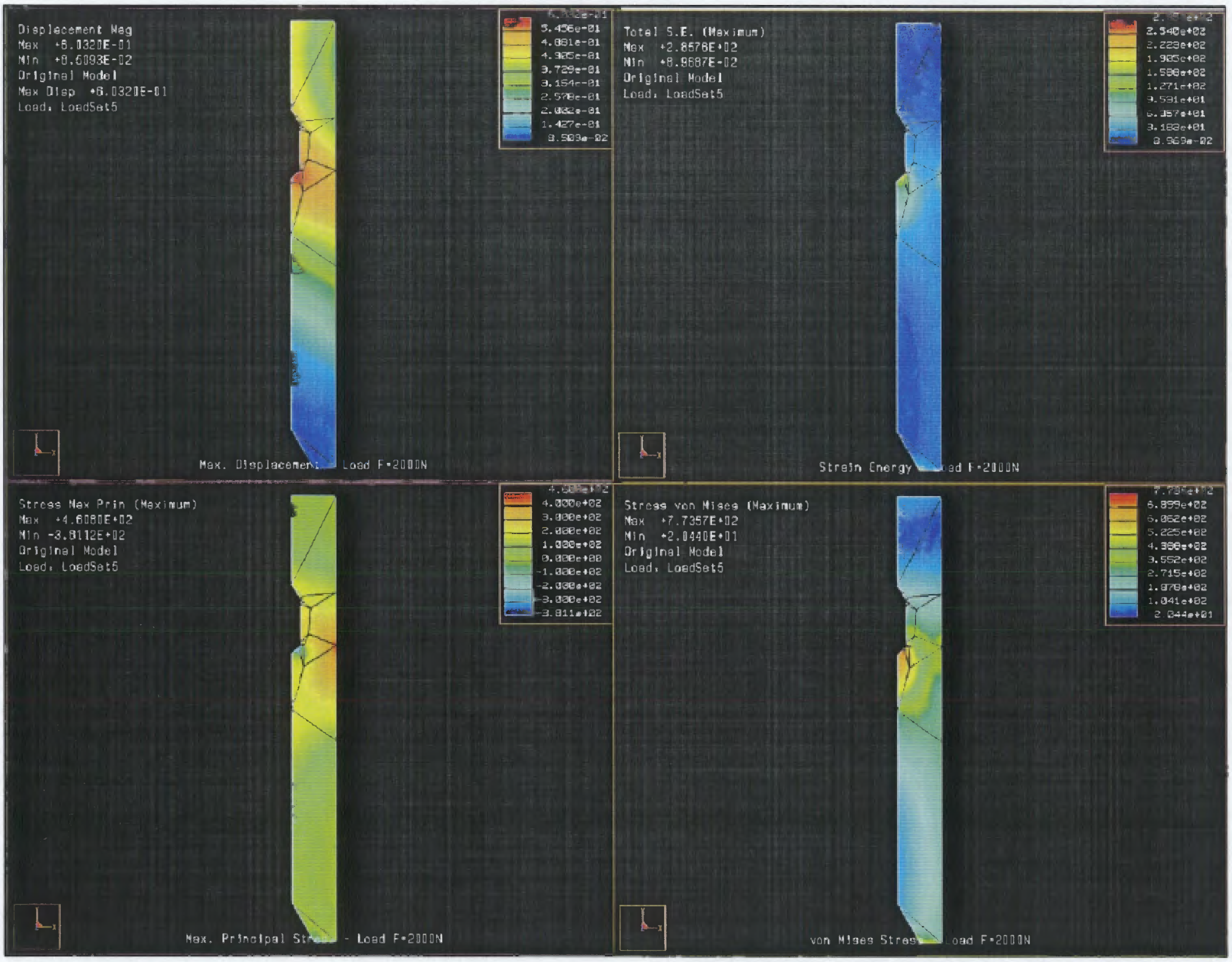


Figure E.10 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load F of 2000N

Shaft - Case II

1. Load: W' = 500N

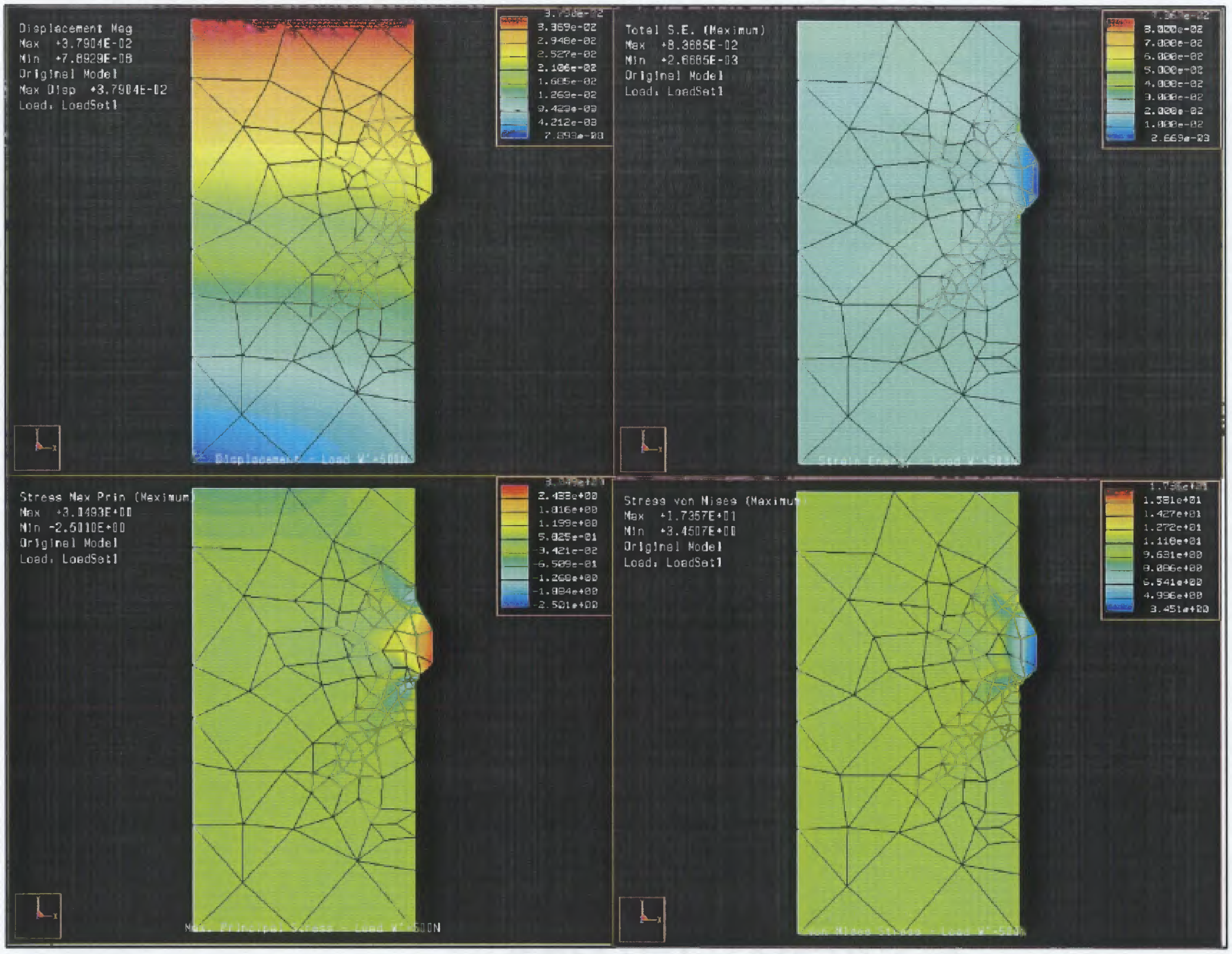


Figure E.11 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load W' of 500N

Shaft - Case II

2. Load: W' = 1500N

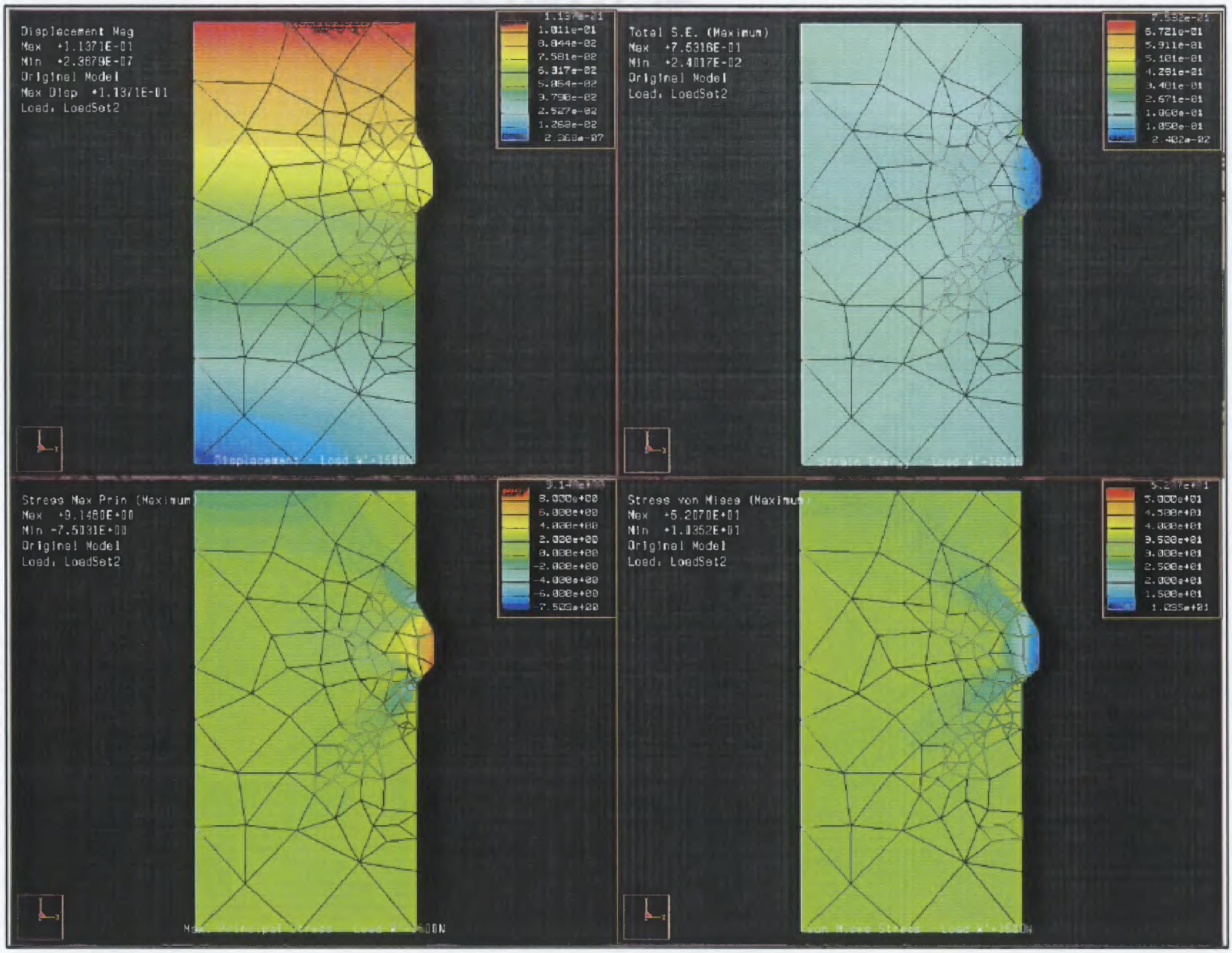


Figure E.12 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load W' of 1500N

Shaft - Case II

3. Load: W' = 2000N

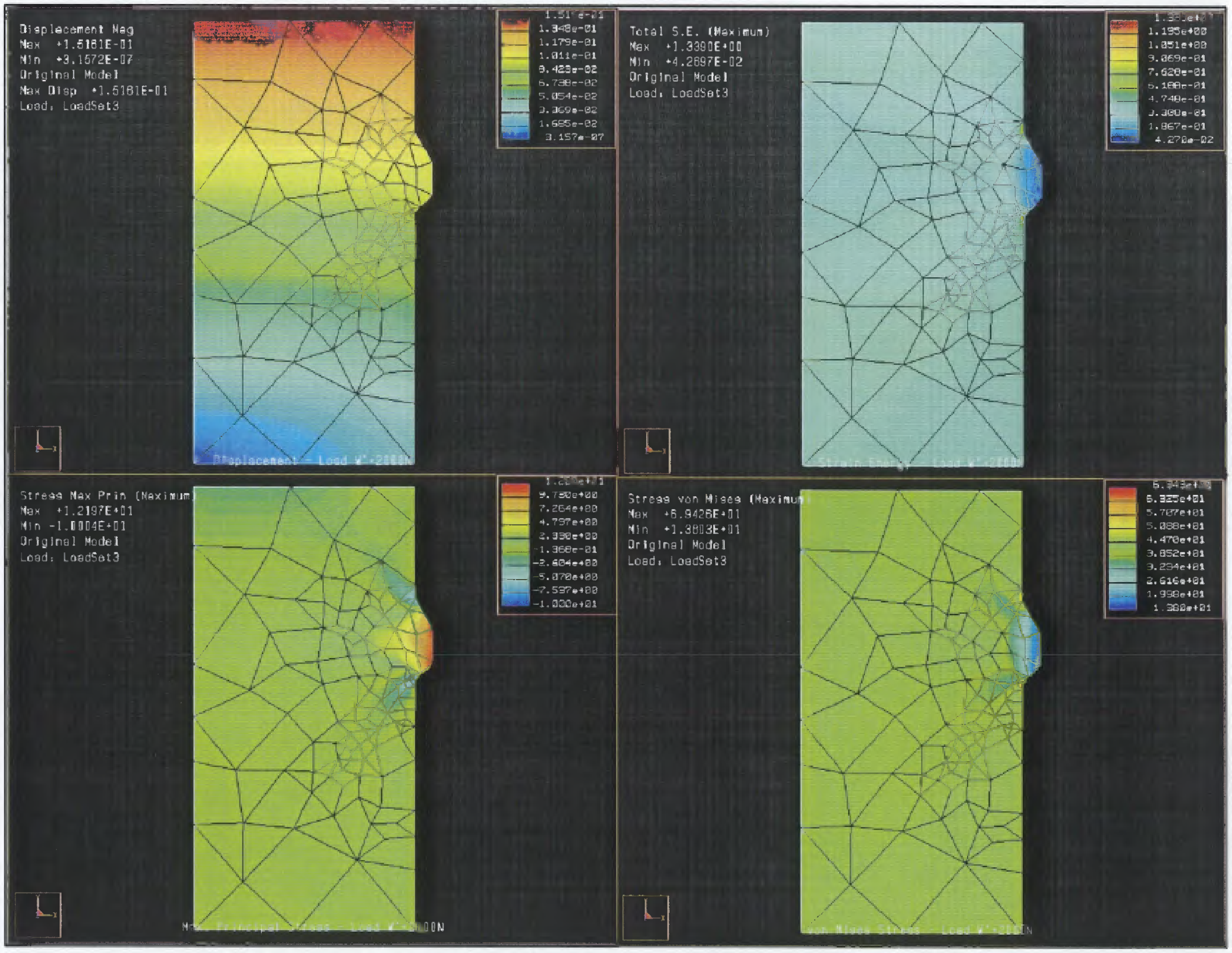


Figure E.13 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load W' of 2000N

Shaft - Case II

4. Load: W' = 5000N

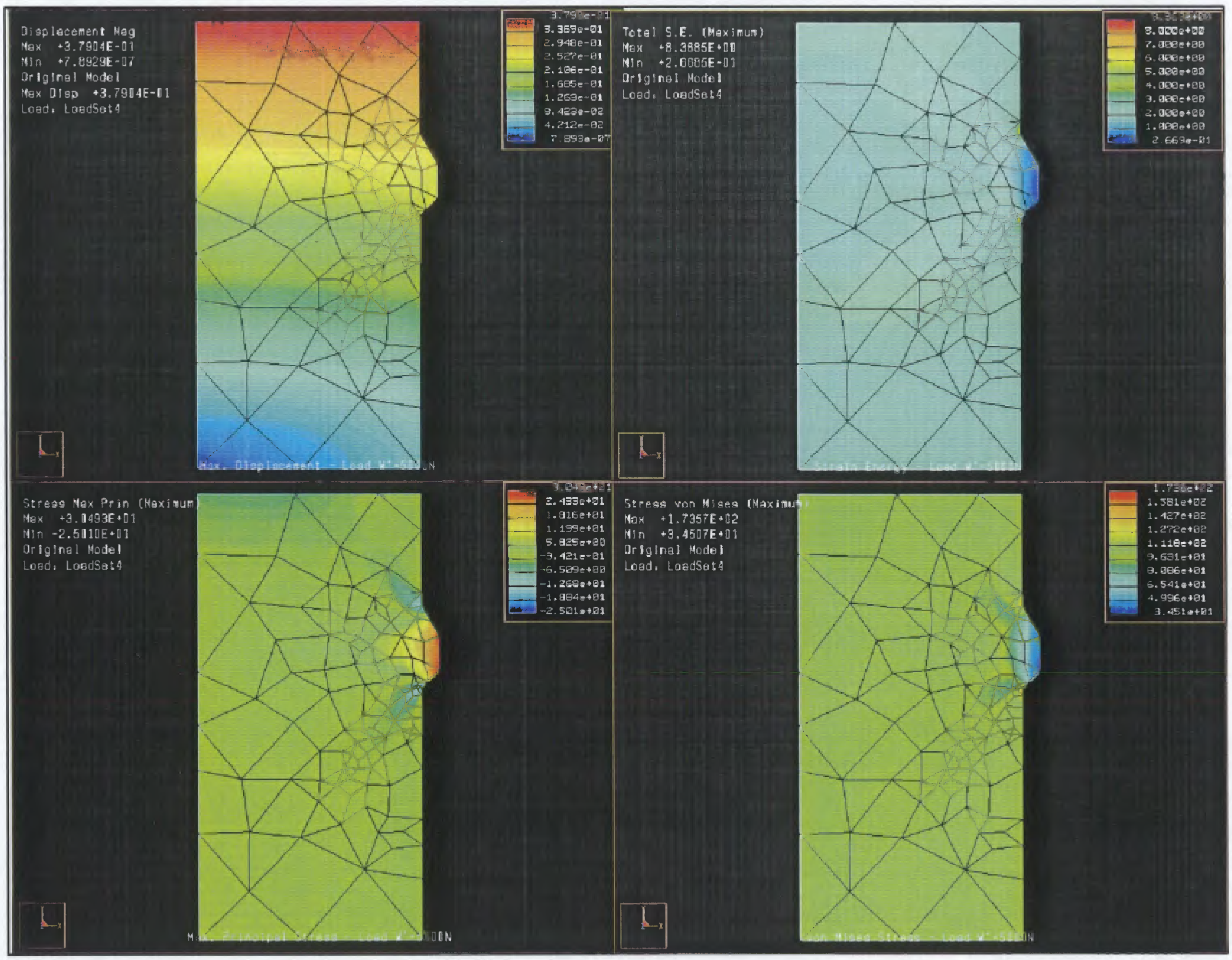


Figure E.14 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load W' of 5000N

Tube - Case II

1. Load: F = 500N

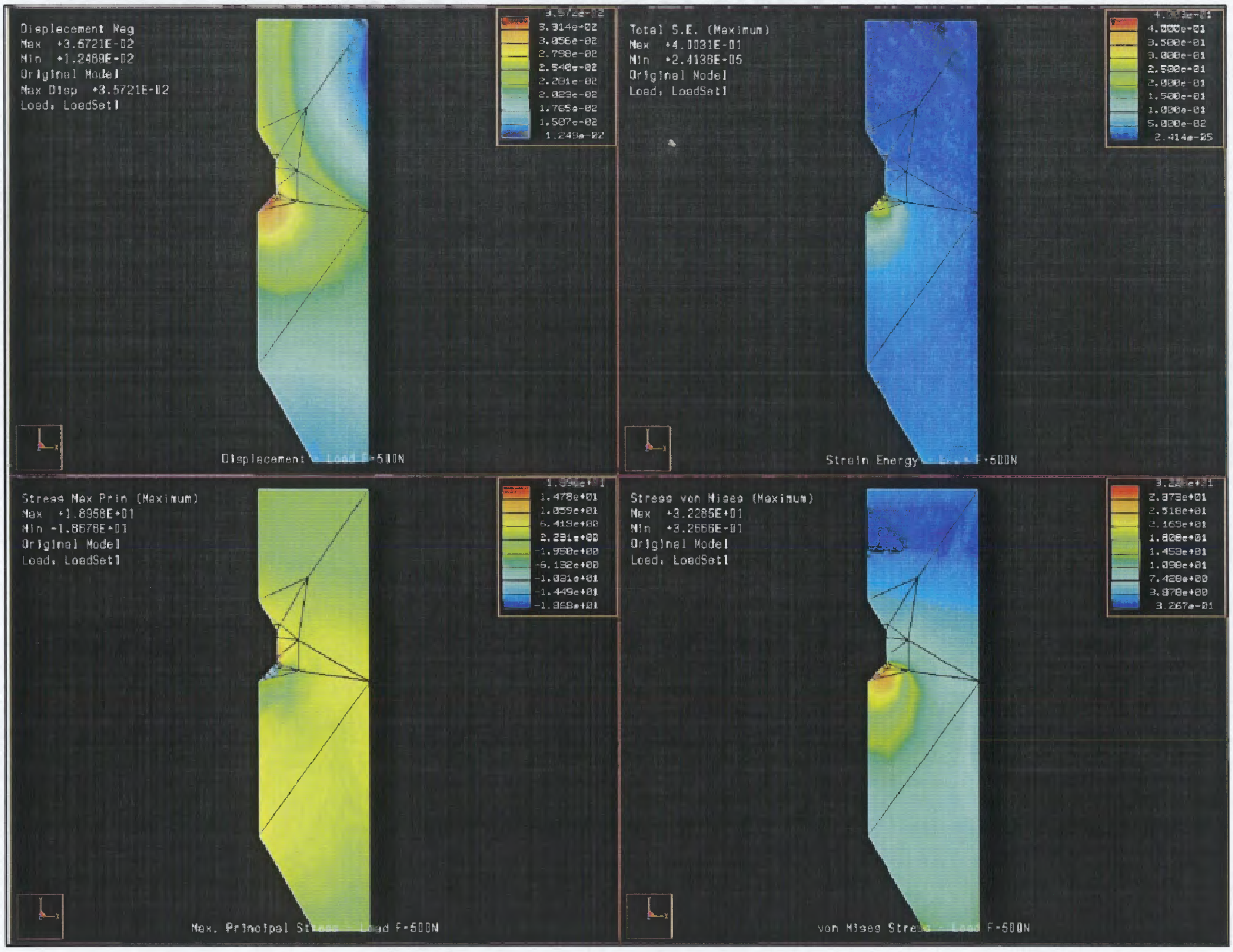


Figure E.15 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load F of 500N

Tube - Case II

2. Load: F = 1500N

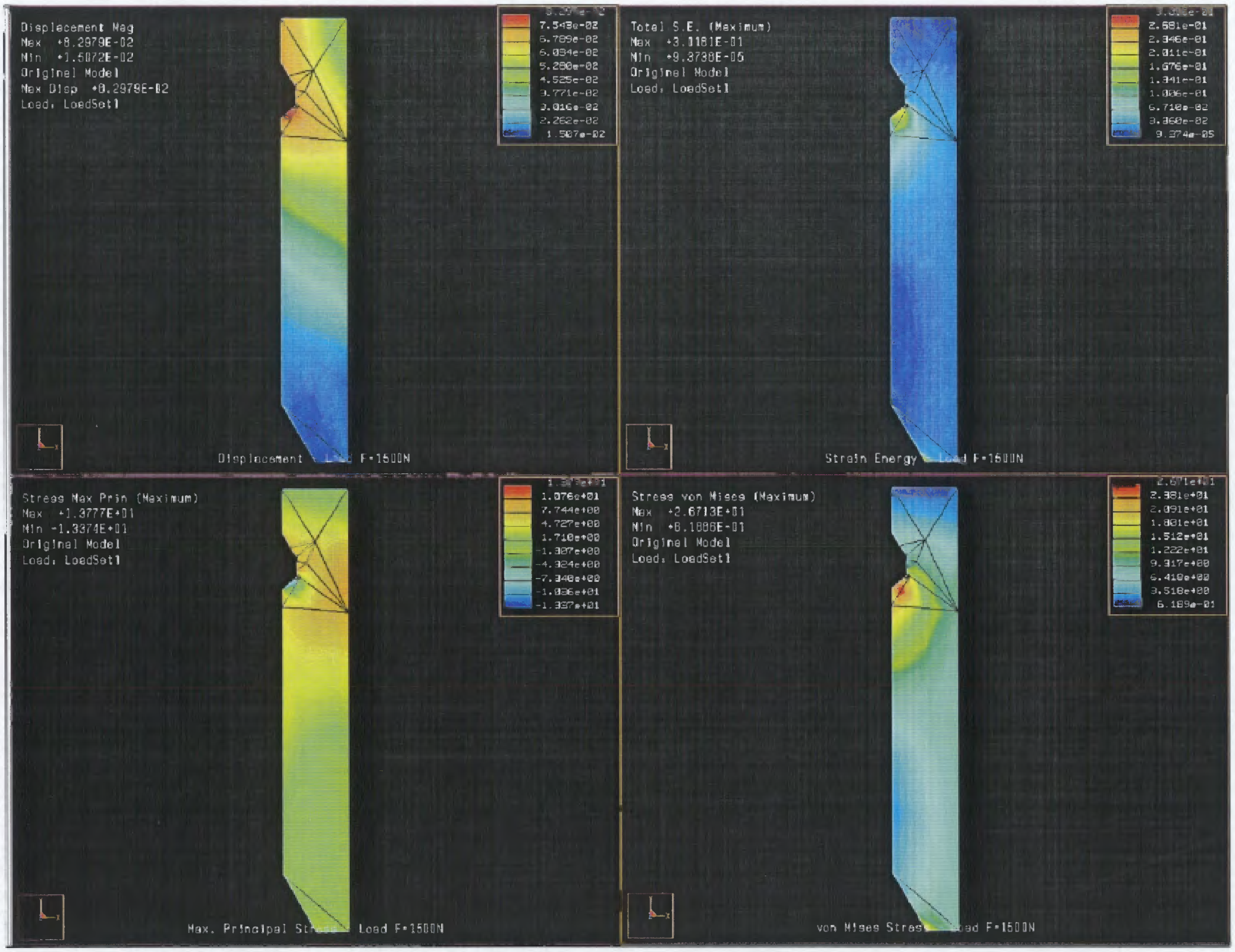


Figure E.16 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load F of 1500N

Tube - Case II

3. Load: F = 2000N

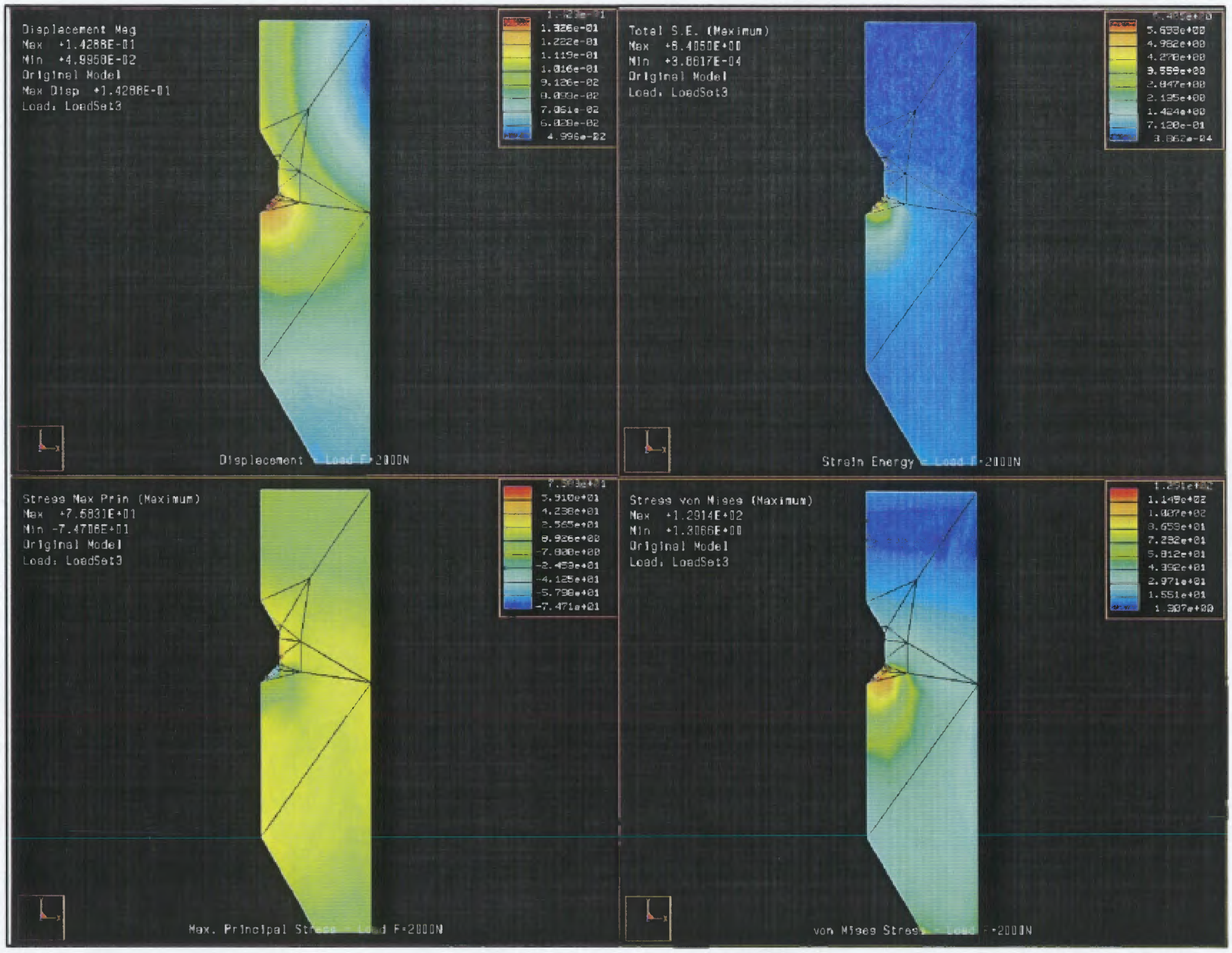


Figure E.17 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load F of 2000N

Tube - Case II

4. Load: F = 5000N

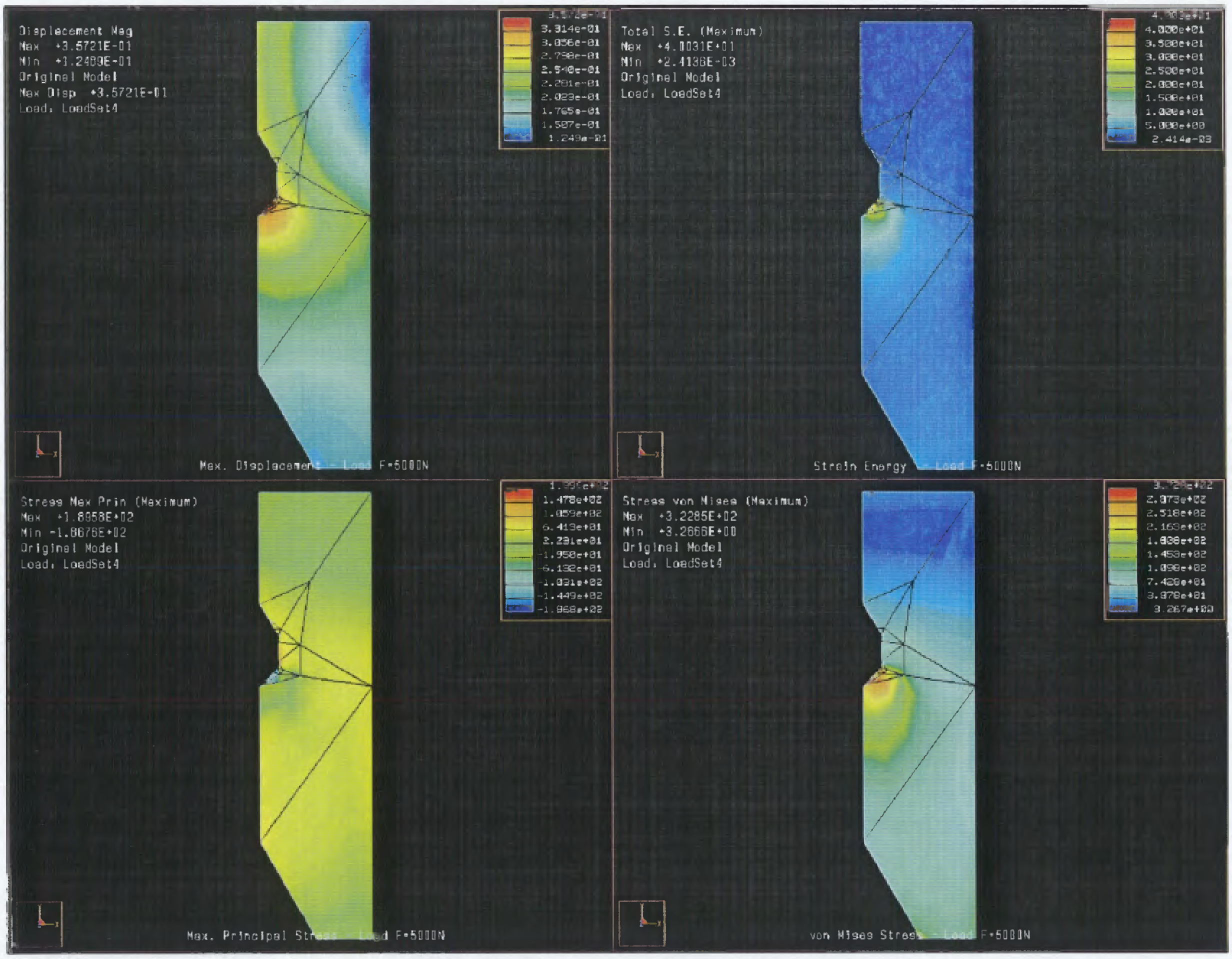


Figure E.18 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load F of 5000N

Shaft - Case III

1. Load: W' = 1500N

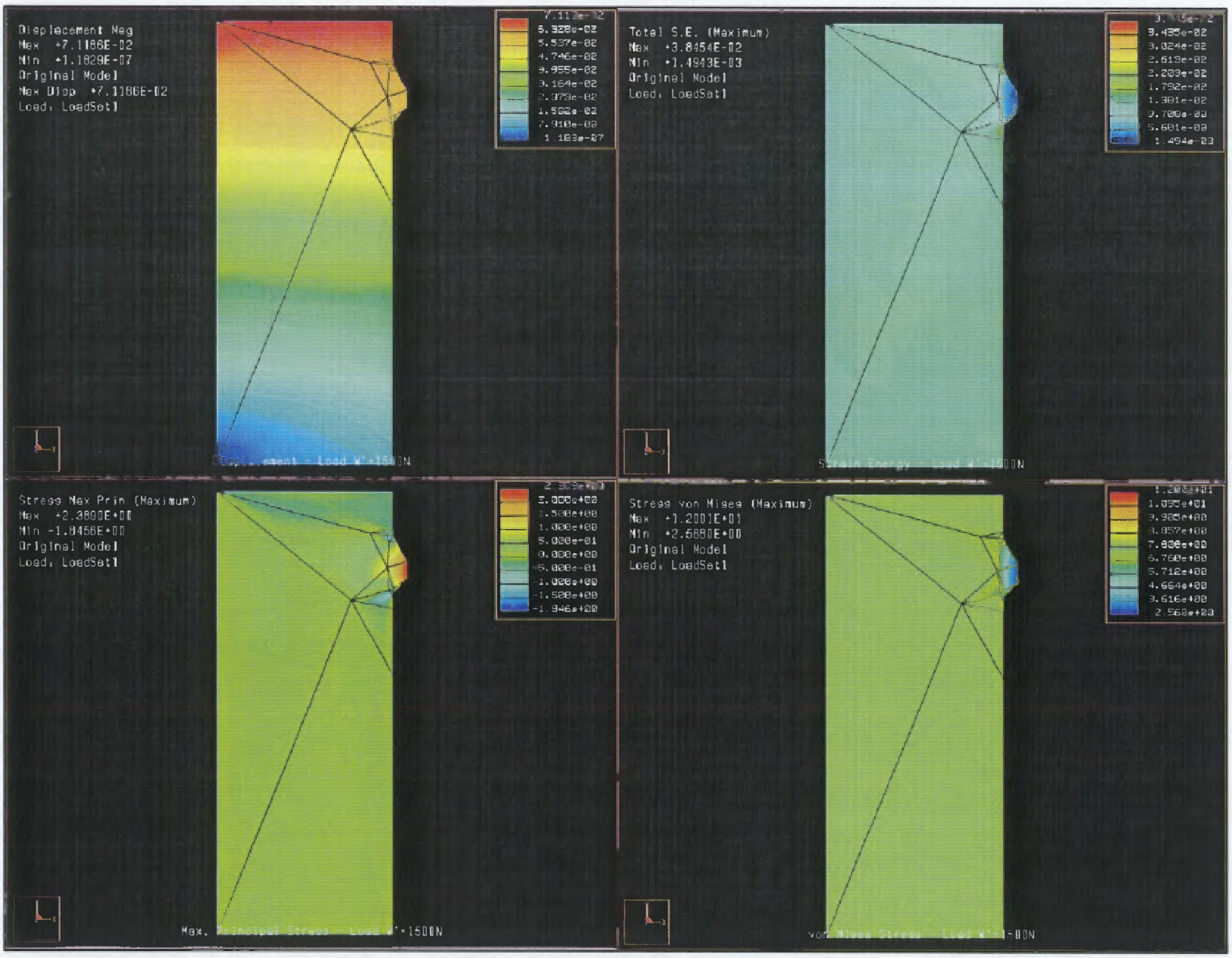


Figure E.19 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load W' of 1500N

Shaft - Case III

2. Load: W' = 4500N

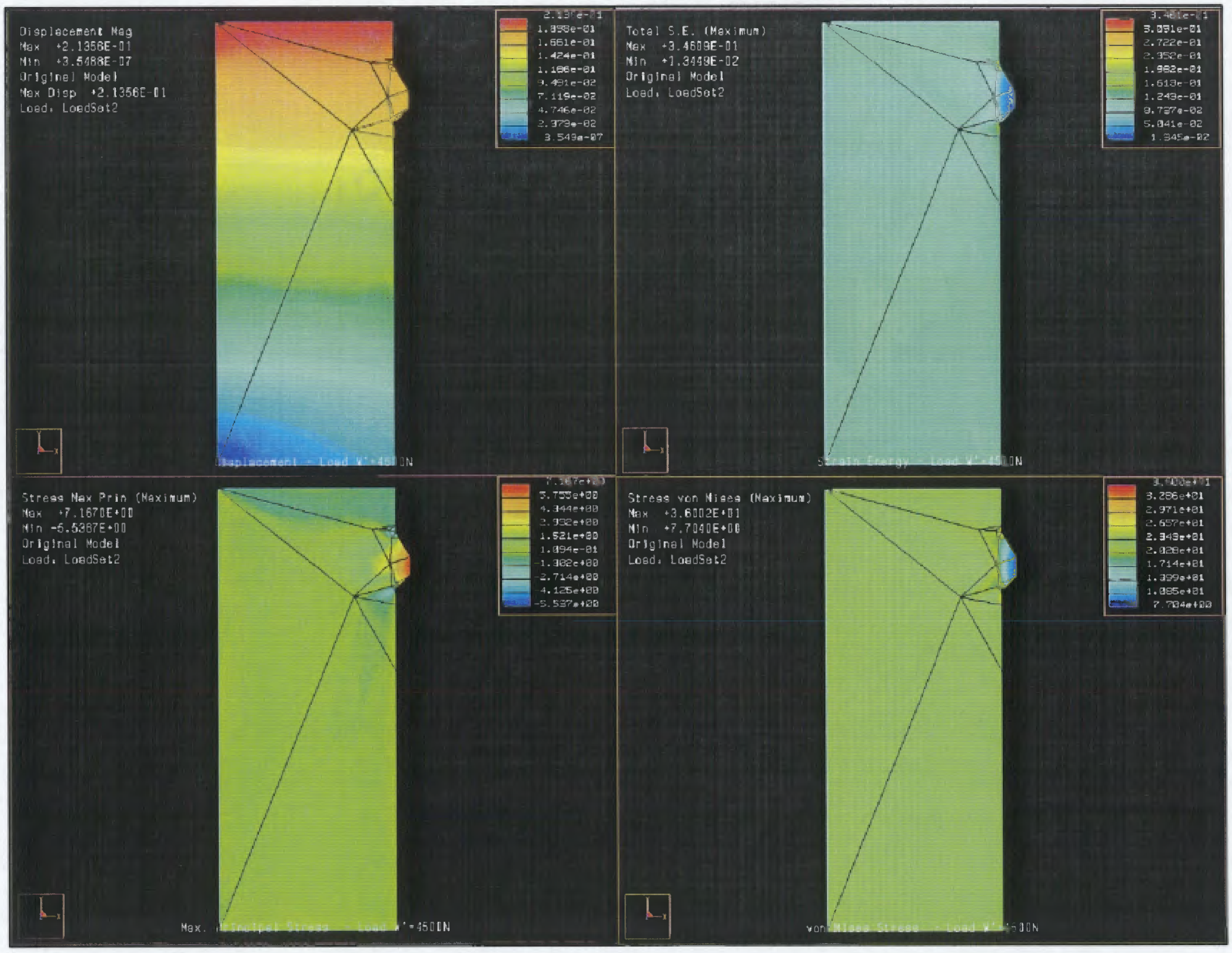


Figure E.20 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load W' of 4500N

Shaft - Case III

3. Load: W' = 6000N

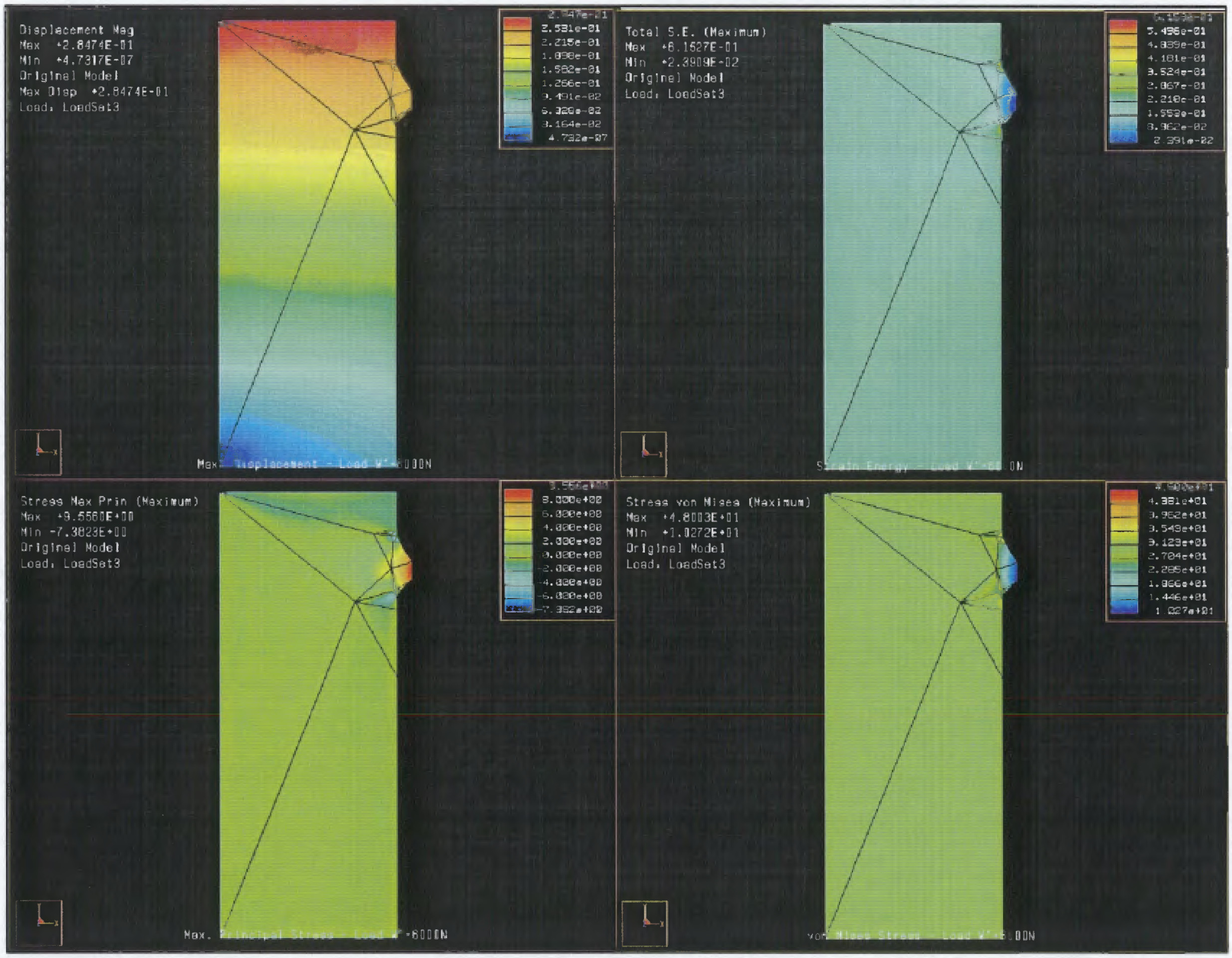


Figure E.21 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load W' of 6000N

Shaft - Case III

4. Load: W' = 10000N

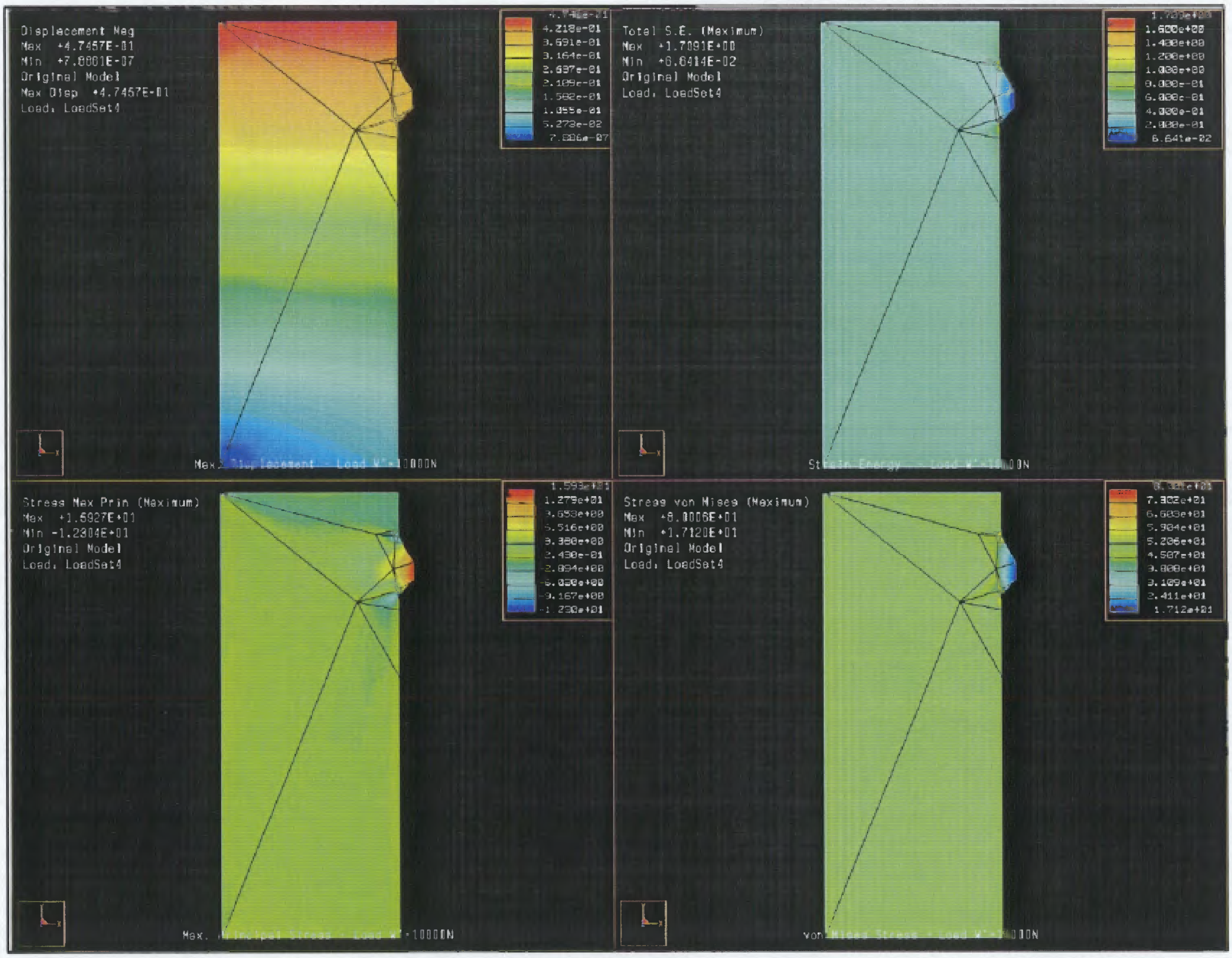


Figure E.22 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load W' of 10000N

Shaft - Case III

5. Load: W' = 15000N

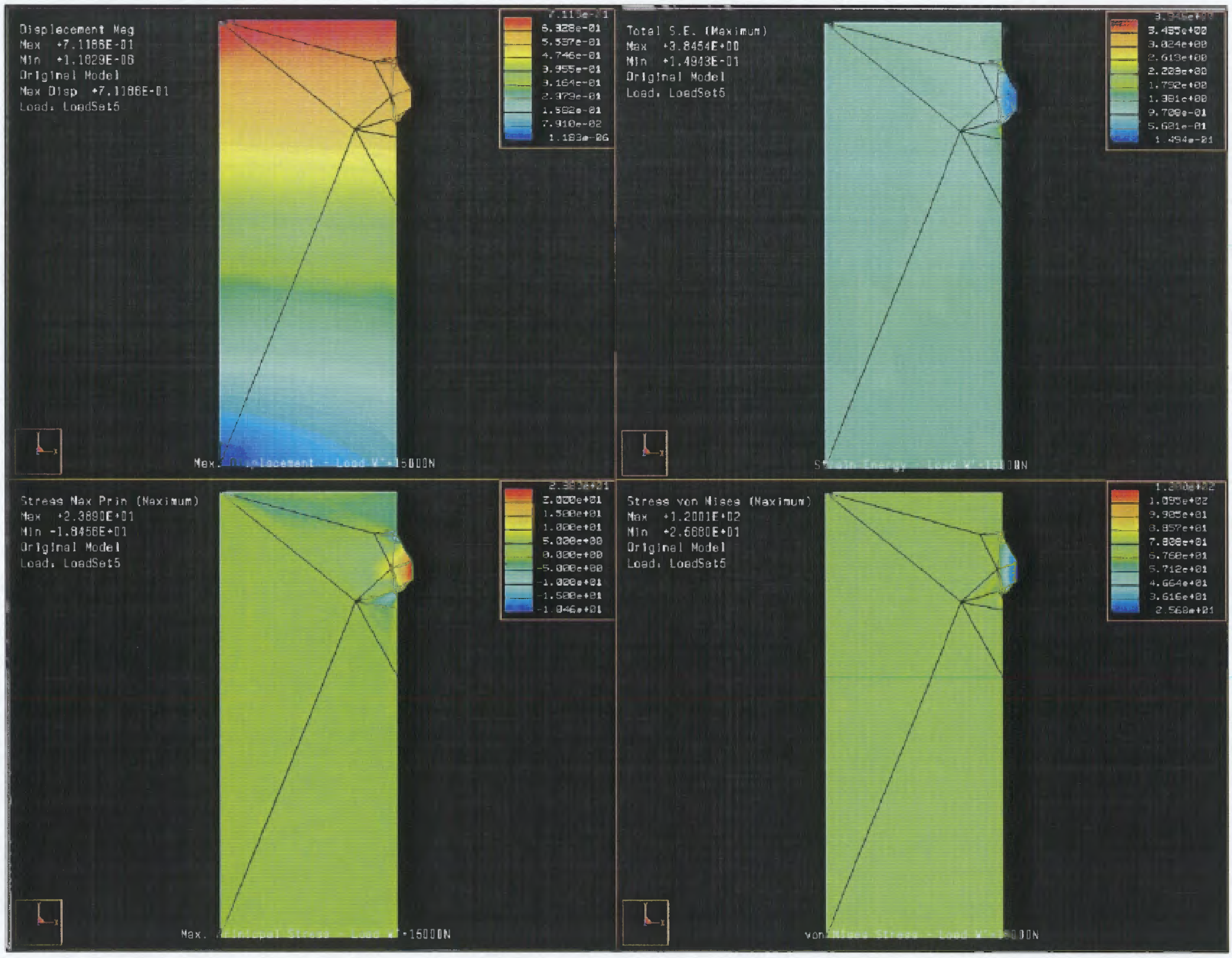


Figure E.23 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load W' of 15000N

SOUTHWORTH
Resumé
100% COTTON FIBER

Tube - Case III

1. Load: F = 1500N

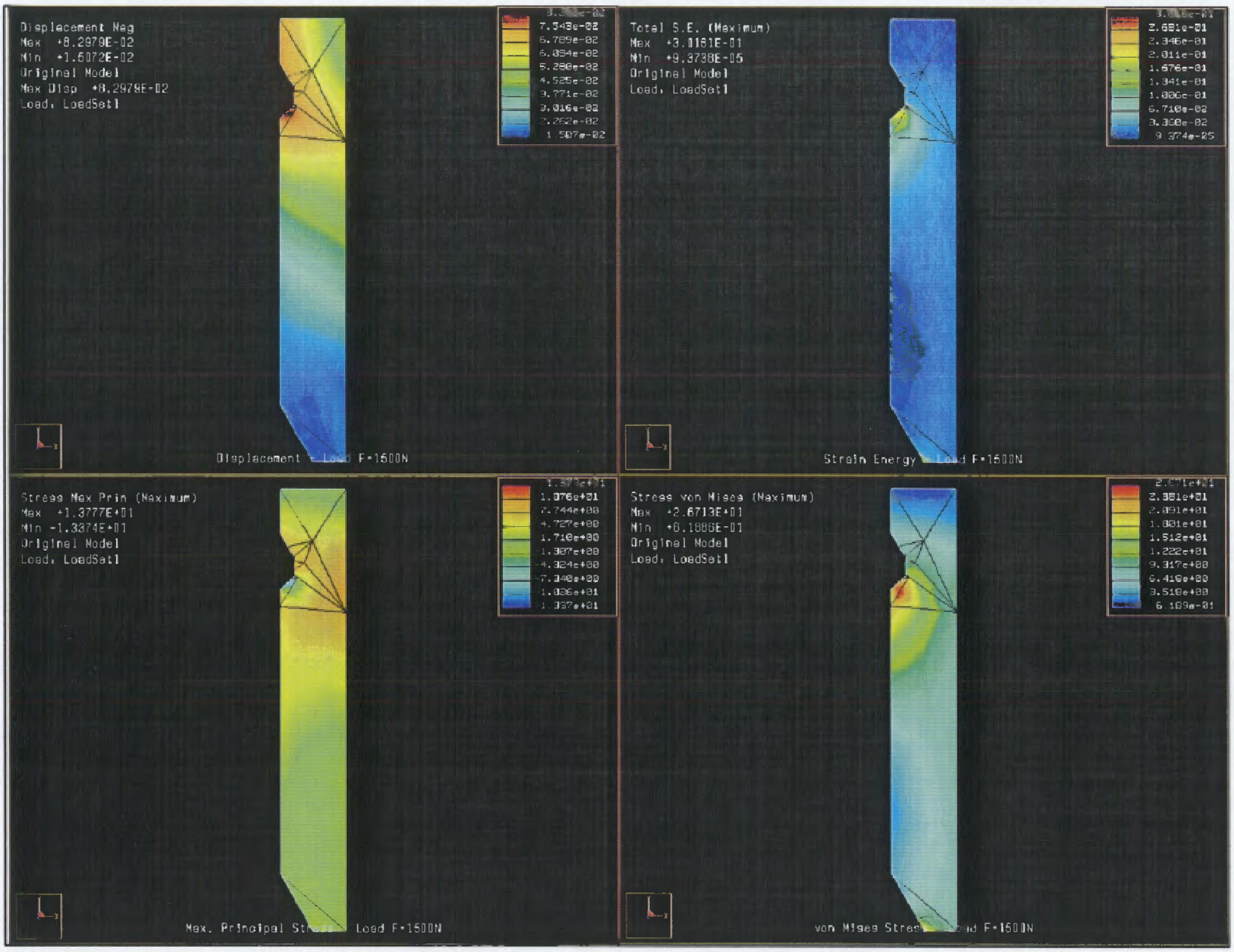


Figure E.24 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load F of 1500N

Tube - Case III

2. Load: F = 4500N

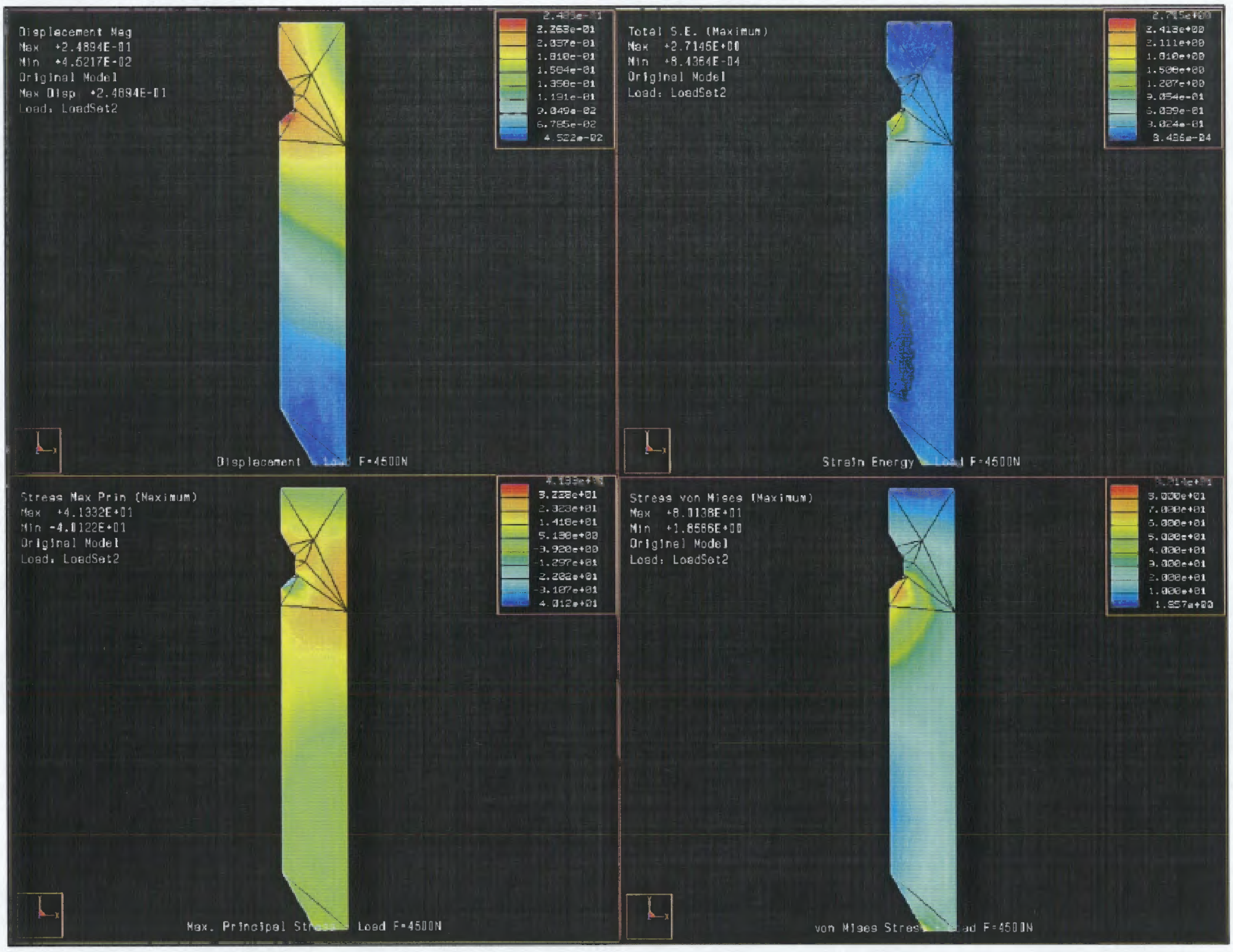


Figure E.25 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load F of 4500N

Tube - Case III

3. Load: F = 6000N

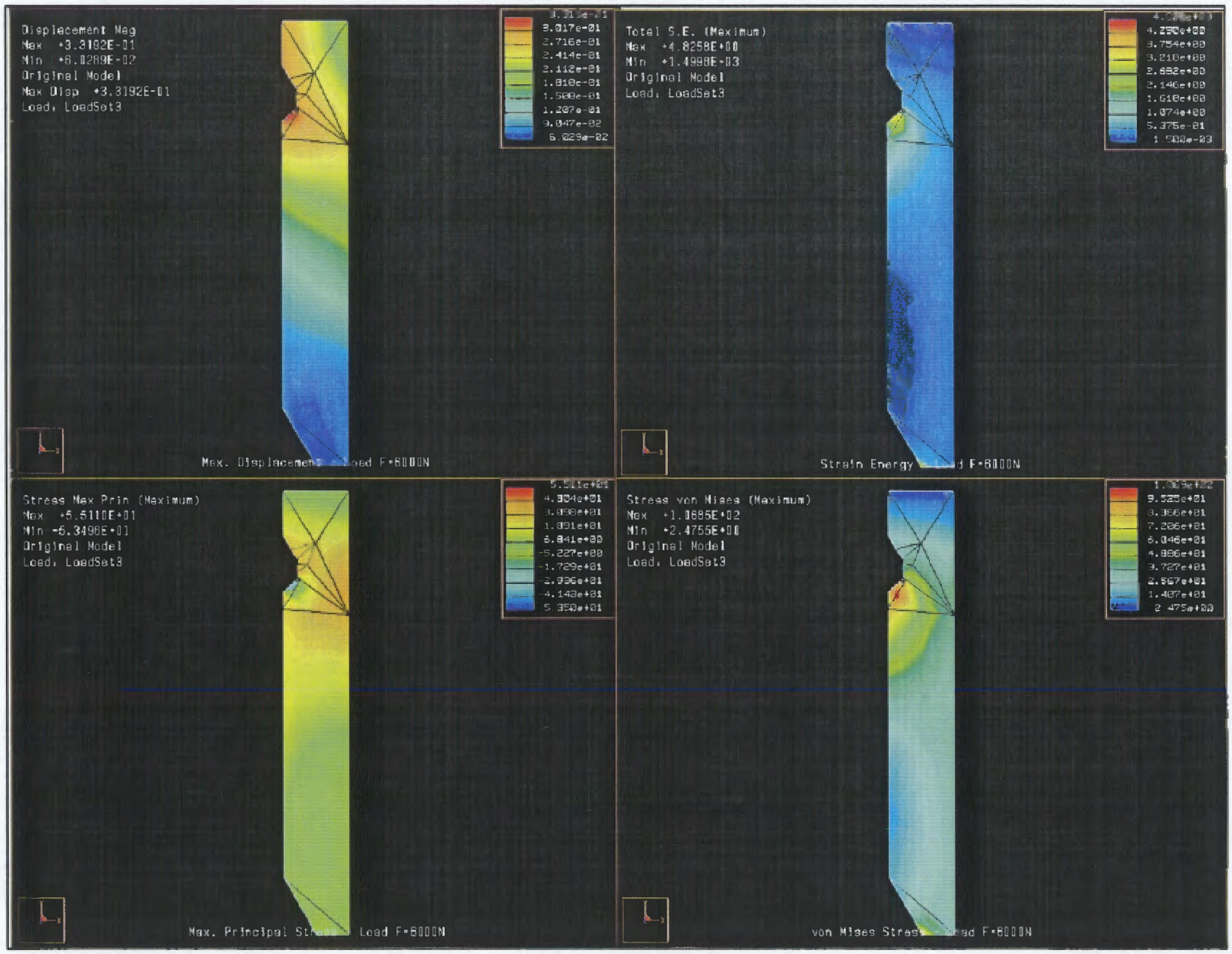


Figure E.26 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load F of 6000N

Tube - Case III

4. Load: F = 10000N

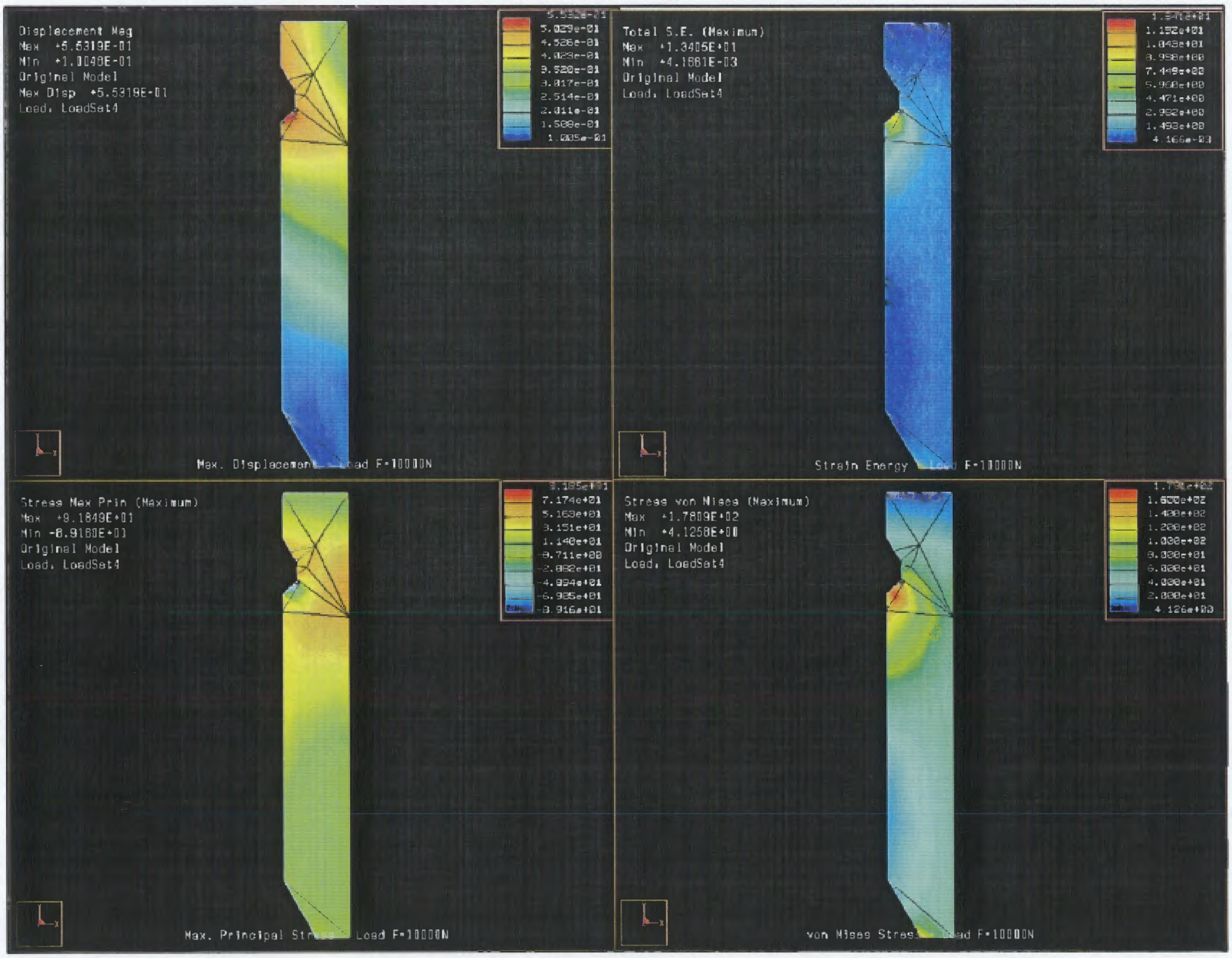


Figure E.27 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load F of 10000N

SOUTHWORTH
Résumé
100% COTTON FIBER

Tube - Case III

5. Load: F = 15000N

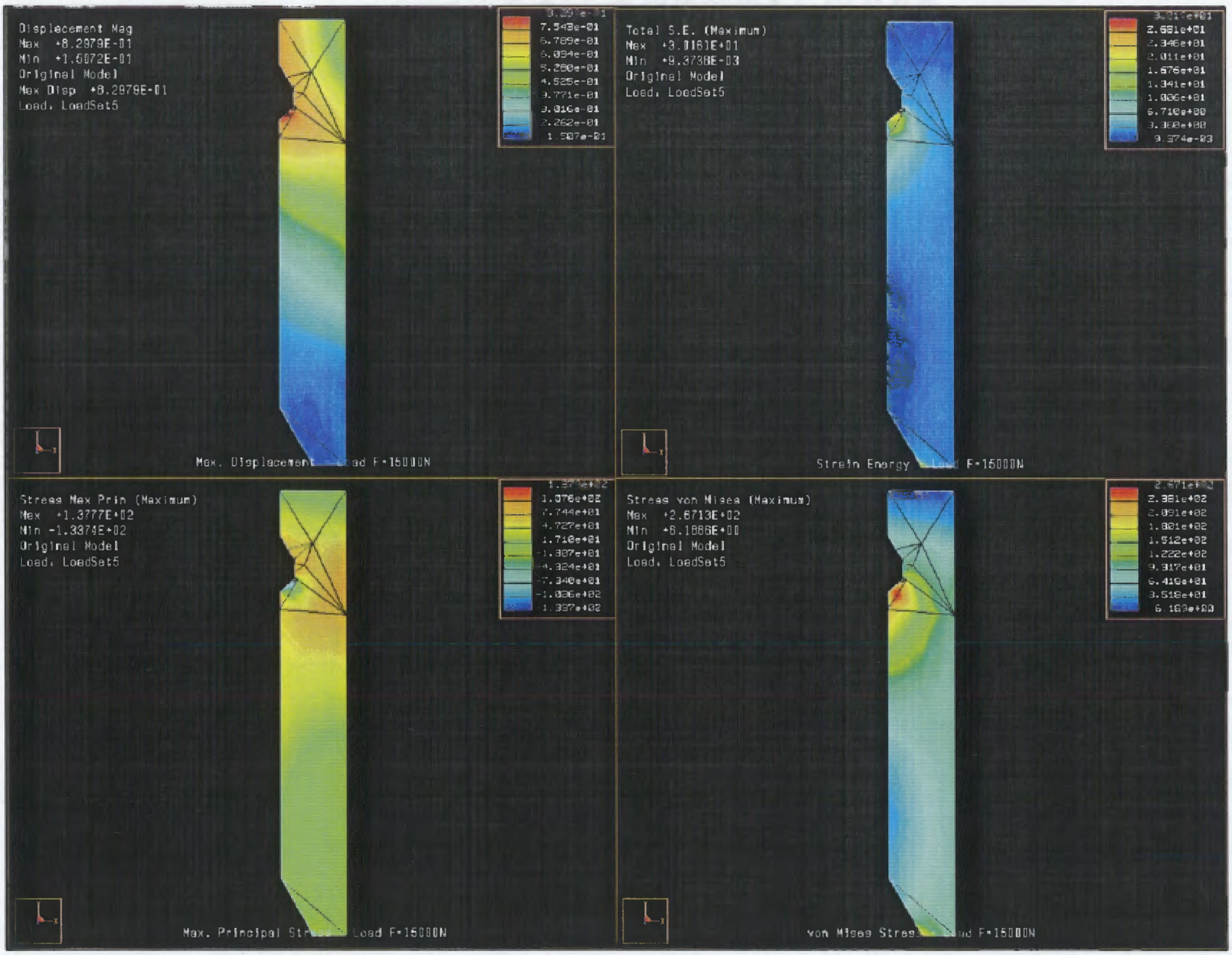


Figure E.28 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load F of 15000N

Shaft - Case IV

1. Load: W' = 3000N

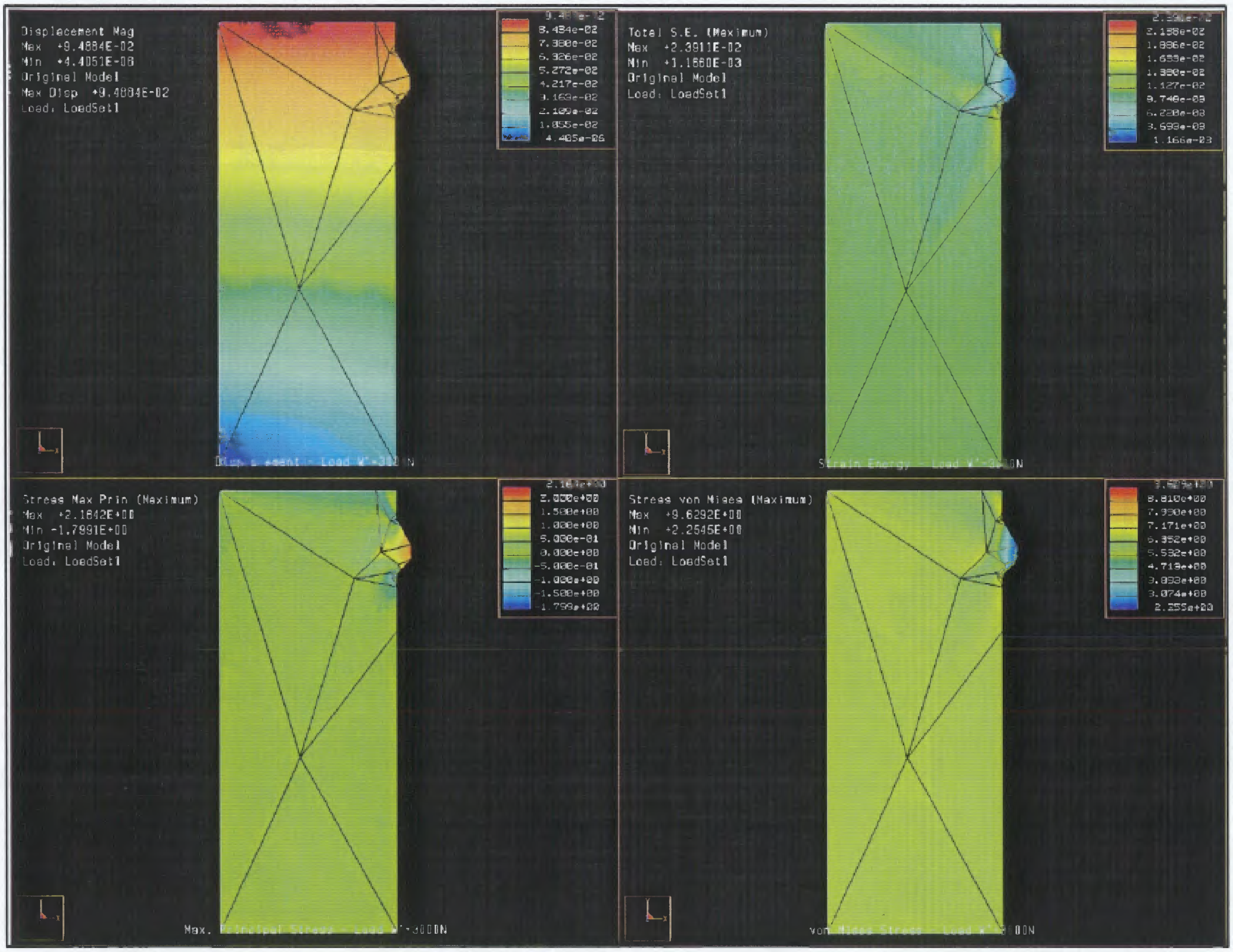


Figure E.29 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load W' of 3000N

Shaft - Case IV

2. Load: W' = 9000N

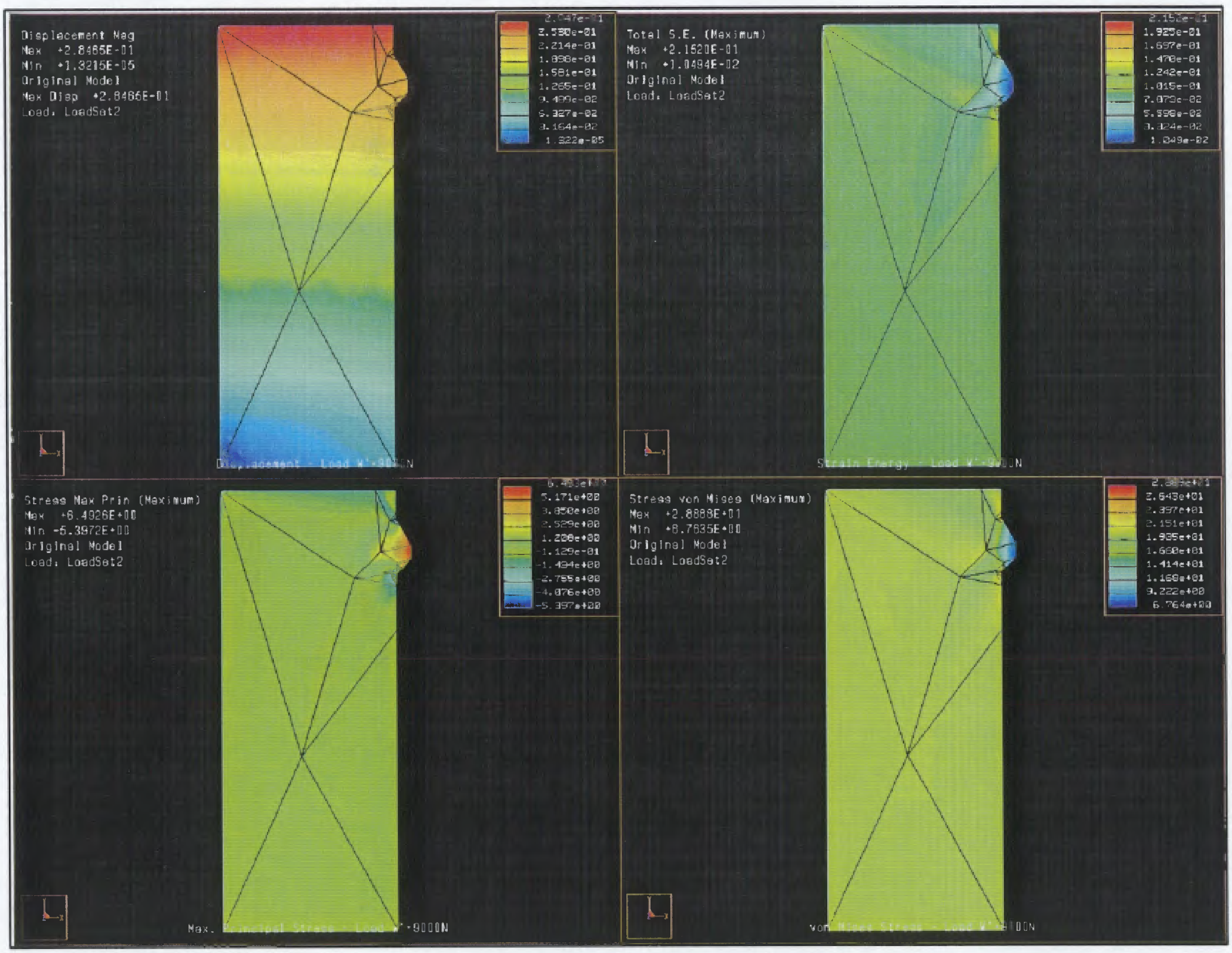


Figure E.30 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load W' of 9000N

Shaft - Case IV

3. Load: W' = 12000N

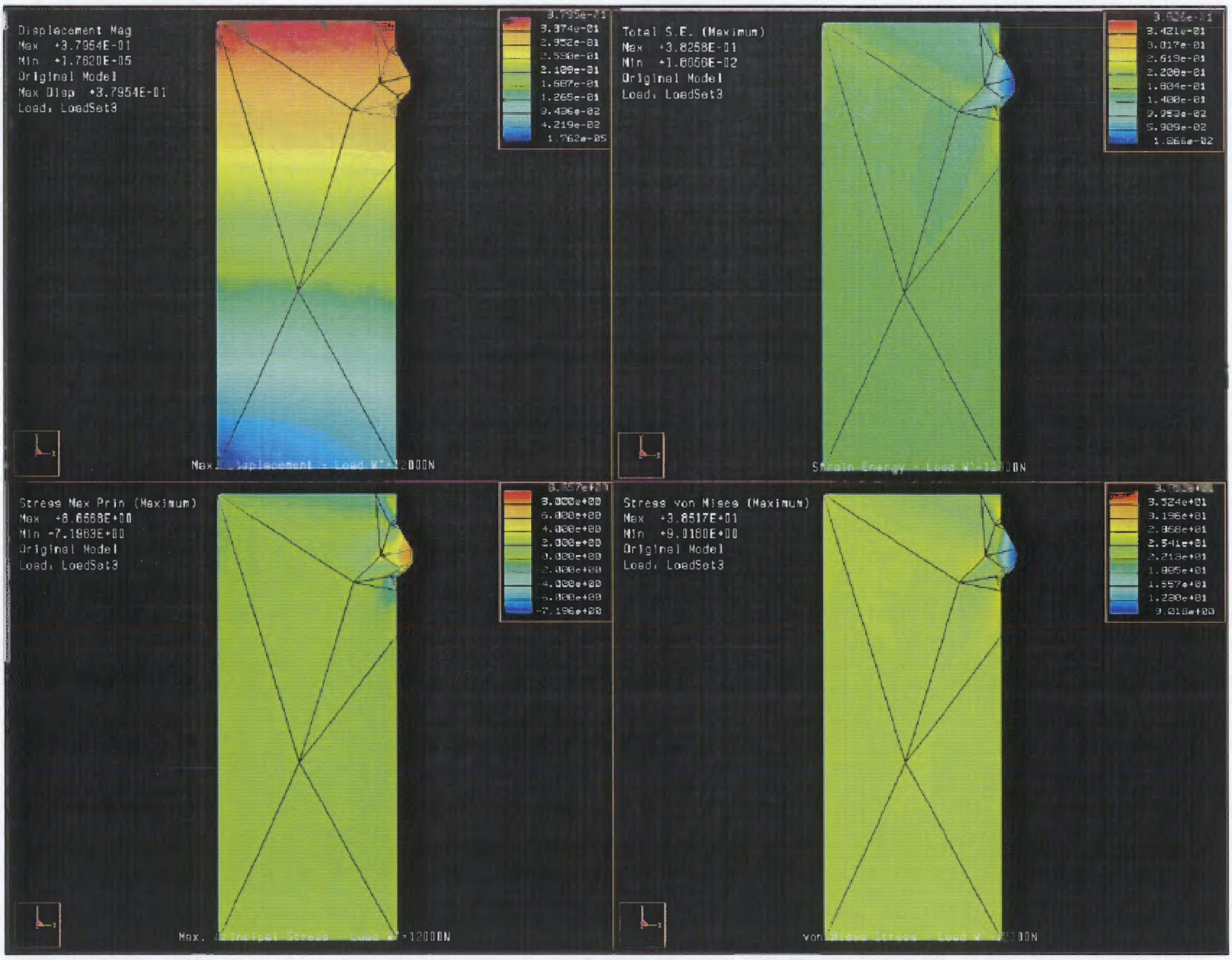


Figure E.31 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load W' of 12000N

SOUTHWORTH
Résumé
100% COTTON FIBER

Shaft - Case IV

4. Load: W' = 20000N

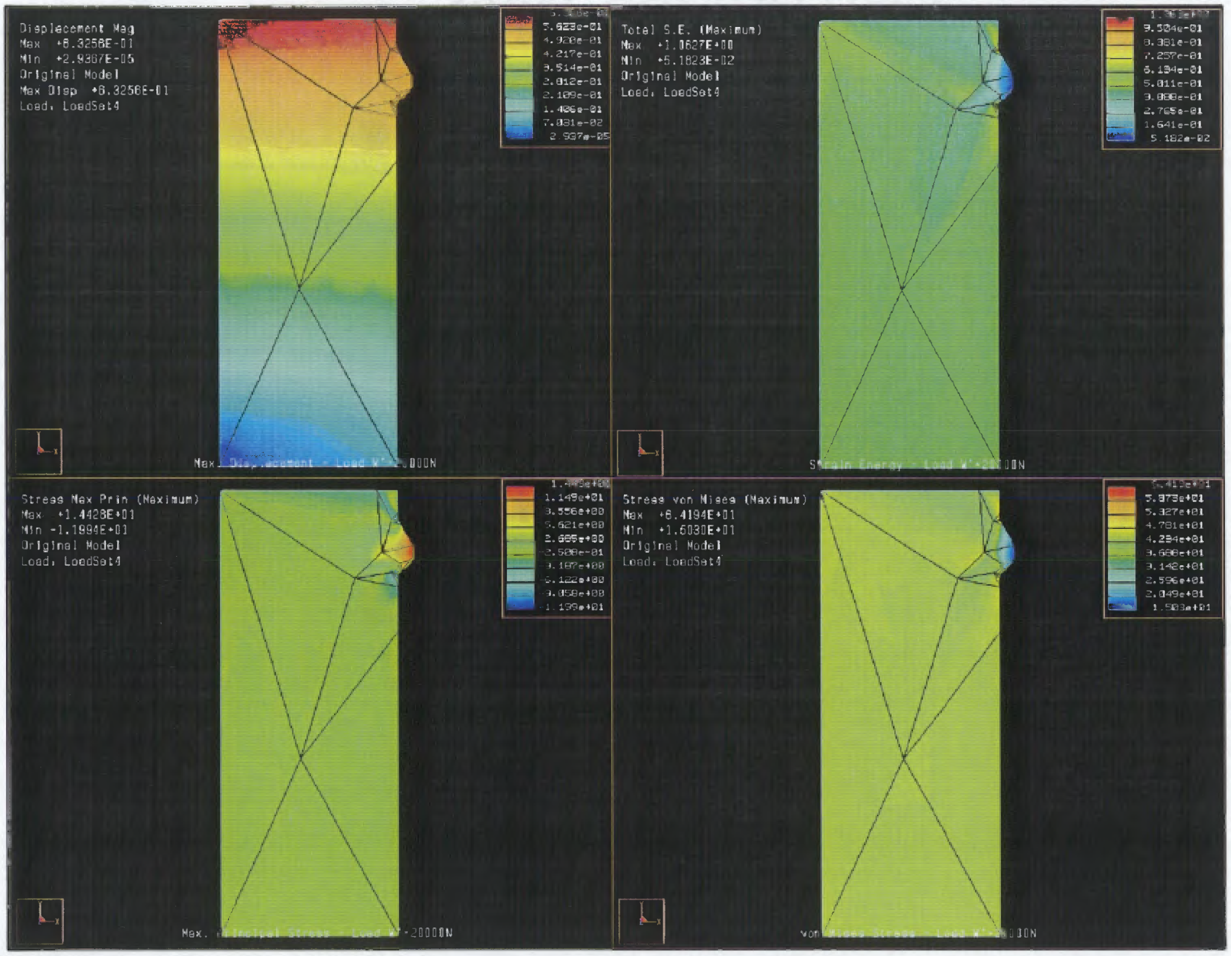


Figure E.32 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load W' of 20000N

Shaft - Case IV

5. Load: W' = 30000N

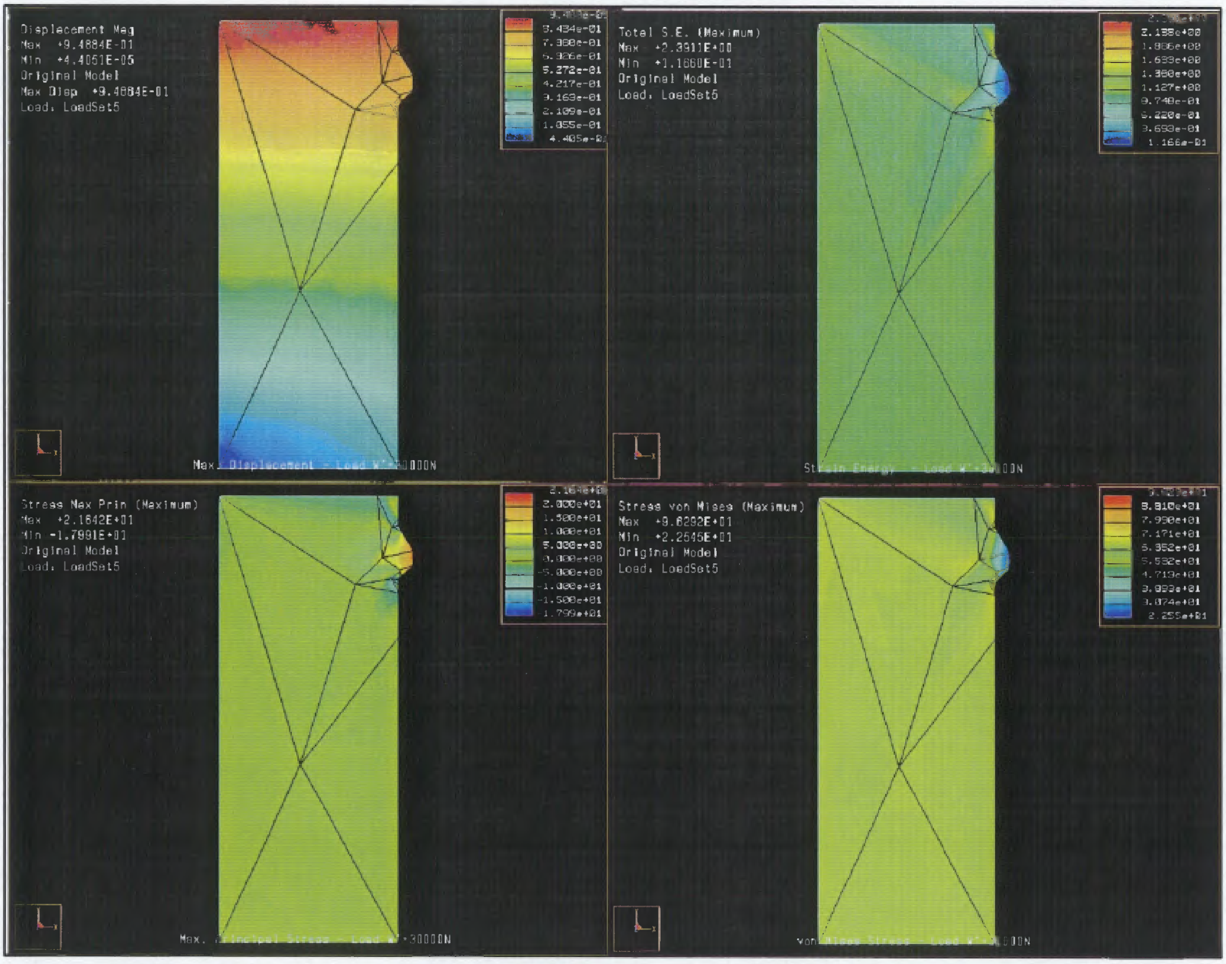


Figure E.33 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load W' of 30000N

Tube - Case IV

1. Load: F = 3000N

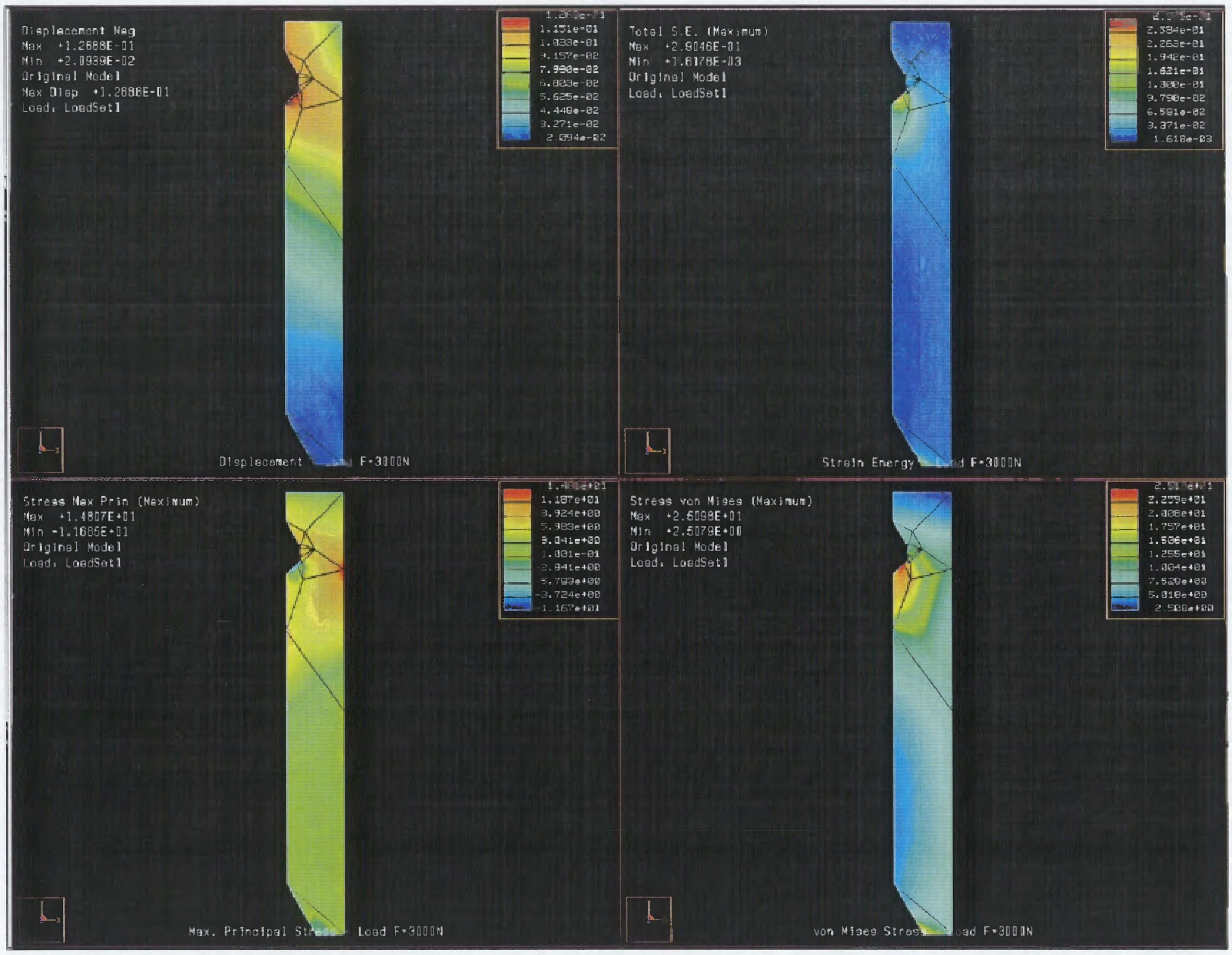


Figure E.34 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load F of 3000N

Tube - Case IV

2. Load: F = 9000N

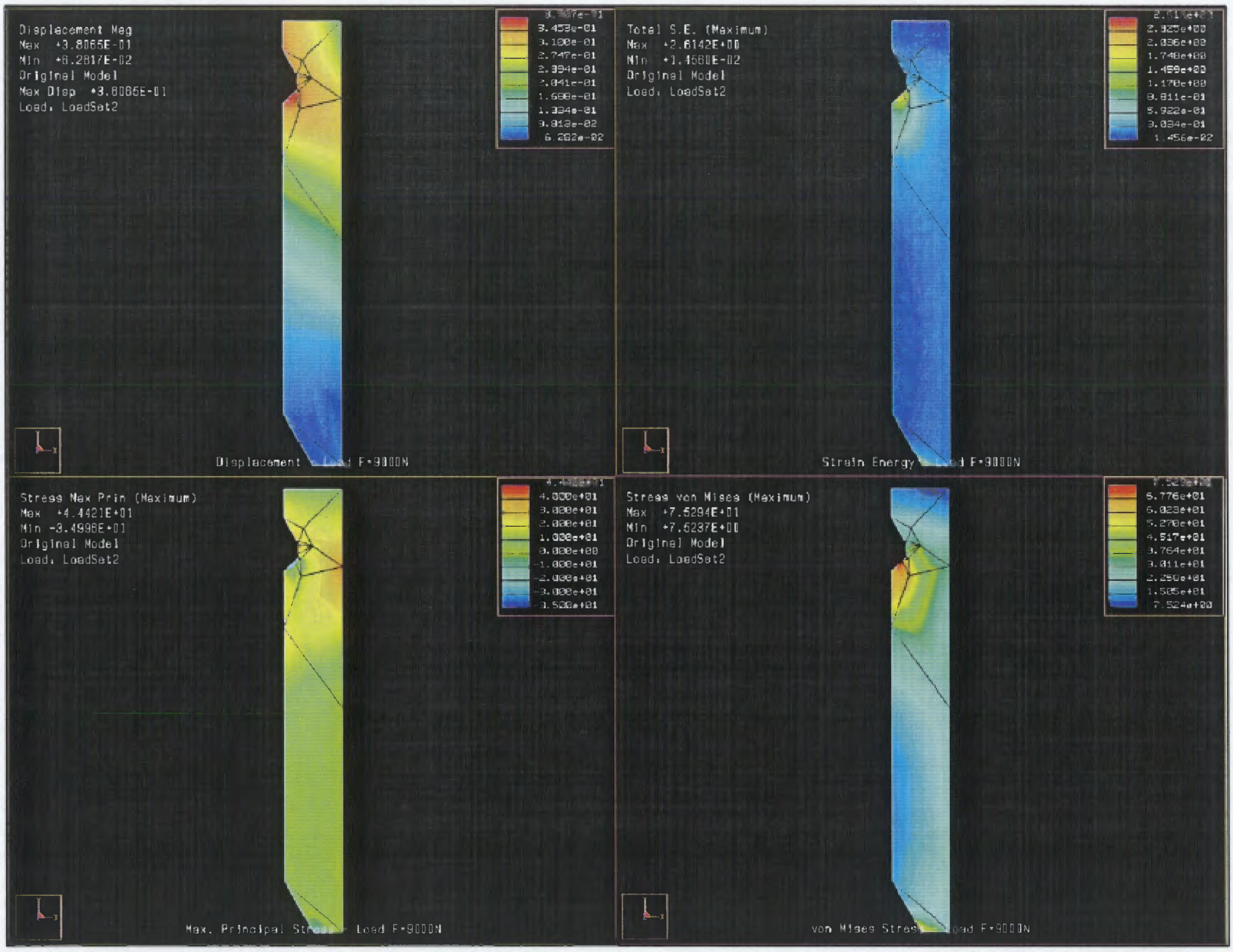


Figure E.35 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load F of 9000N

Tube - Case IV

3. Load: F = 12000N

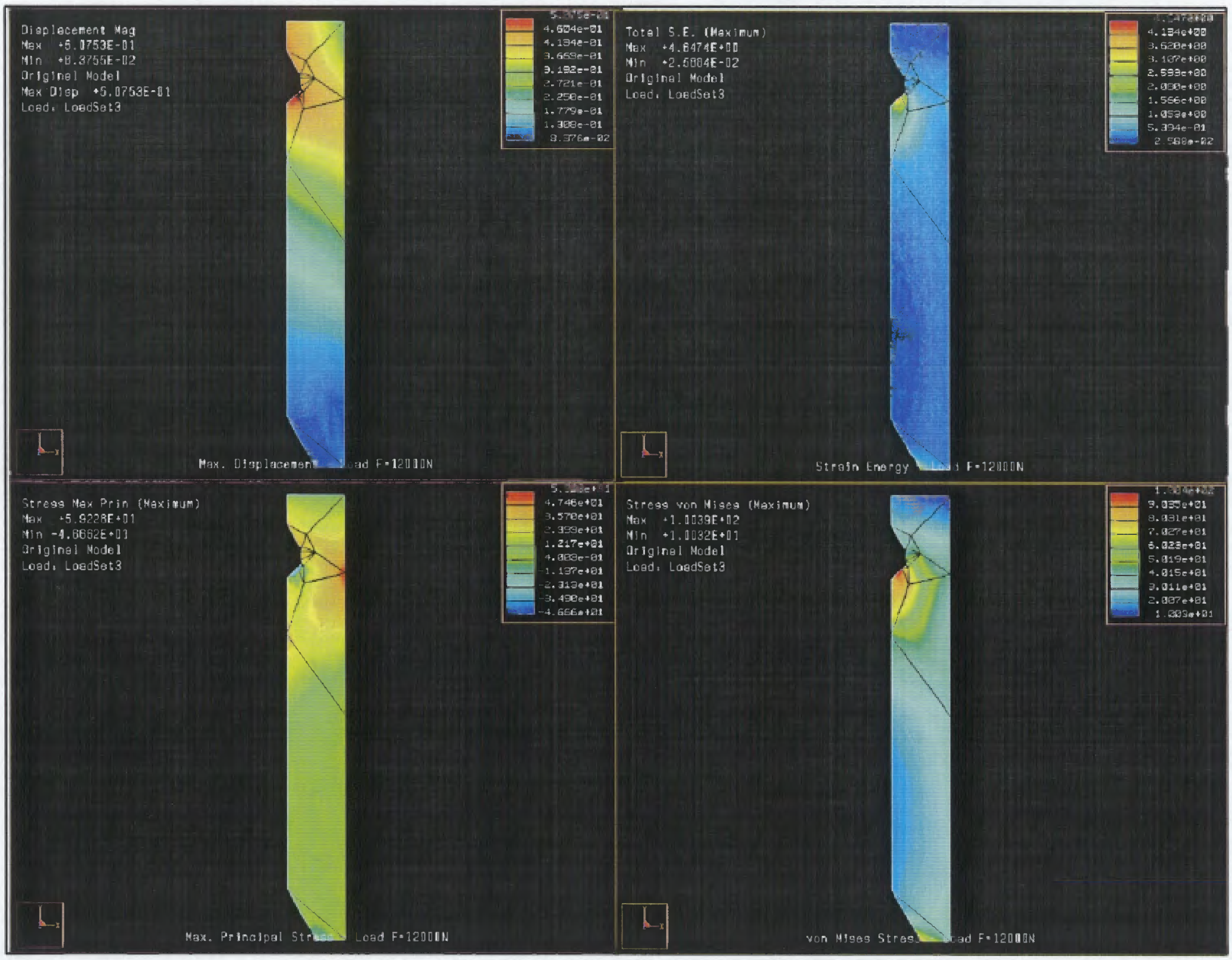


Figure E.36 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load F of 12000N

Tube - Case IV

4. Load: F = 20000N

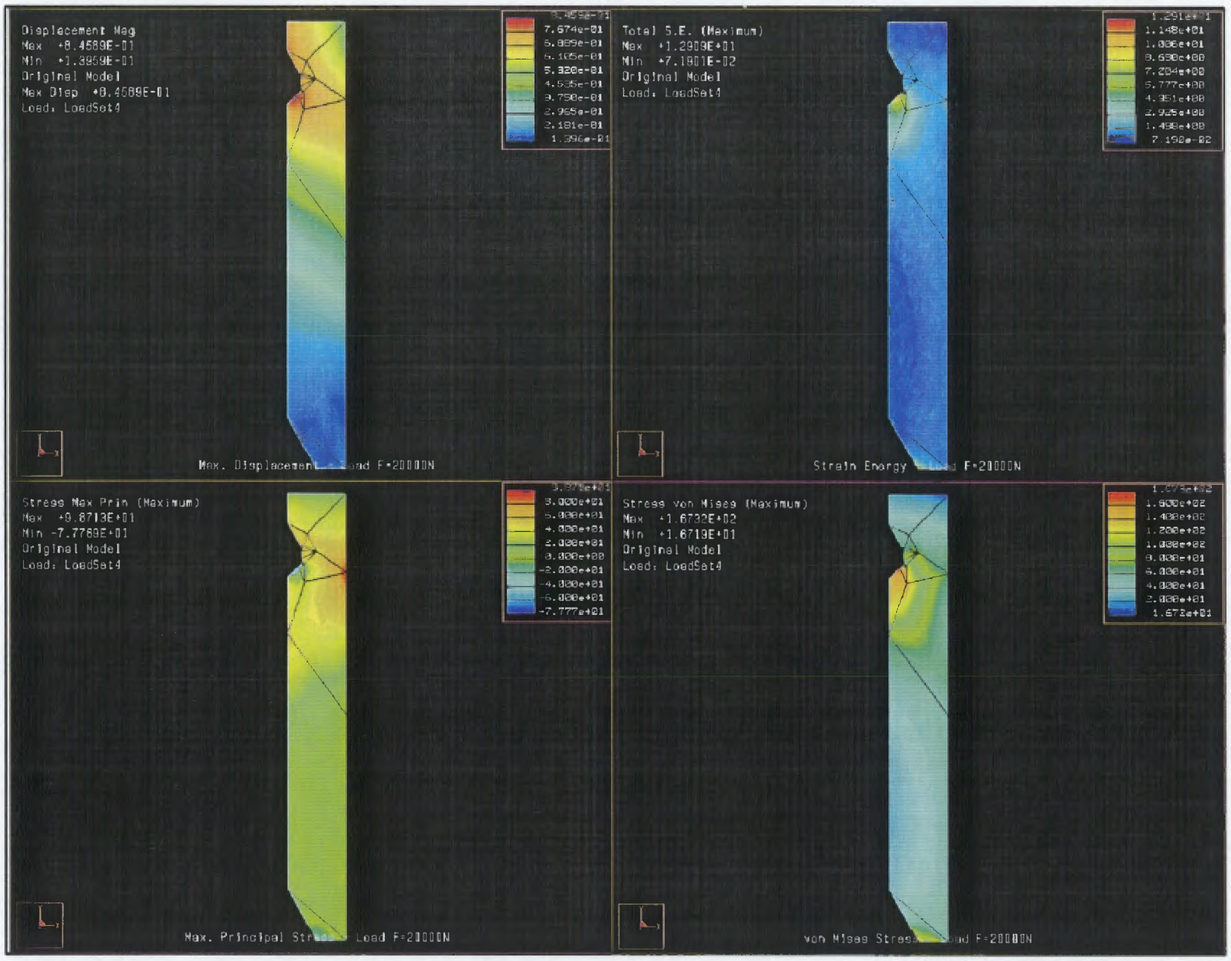


Figure E.37 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load F of 20000N

Tube - Case IV

5. Load: F = 30000N

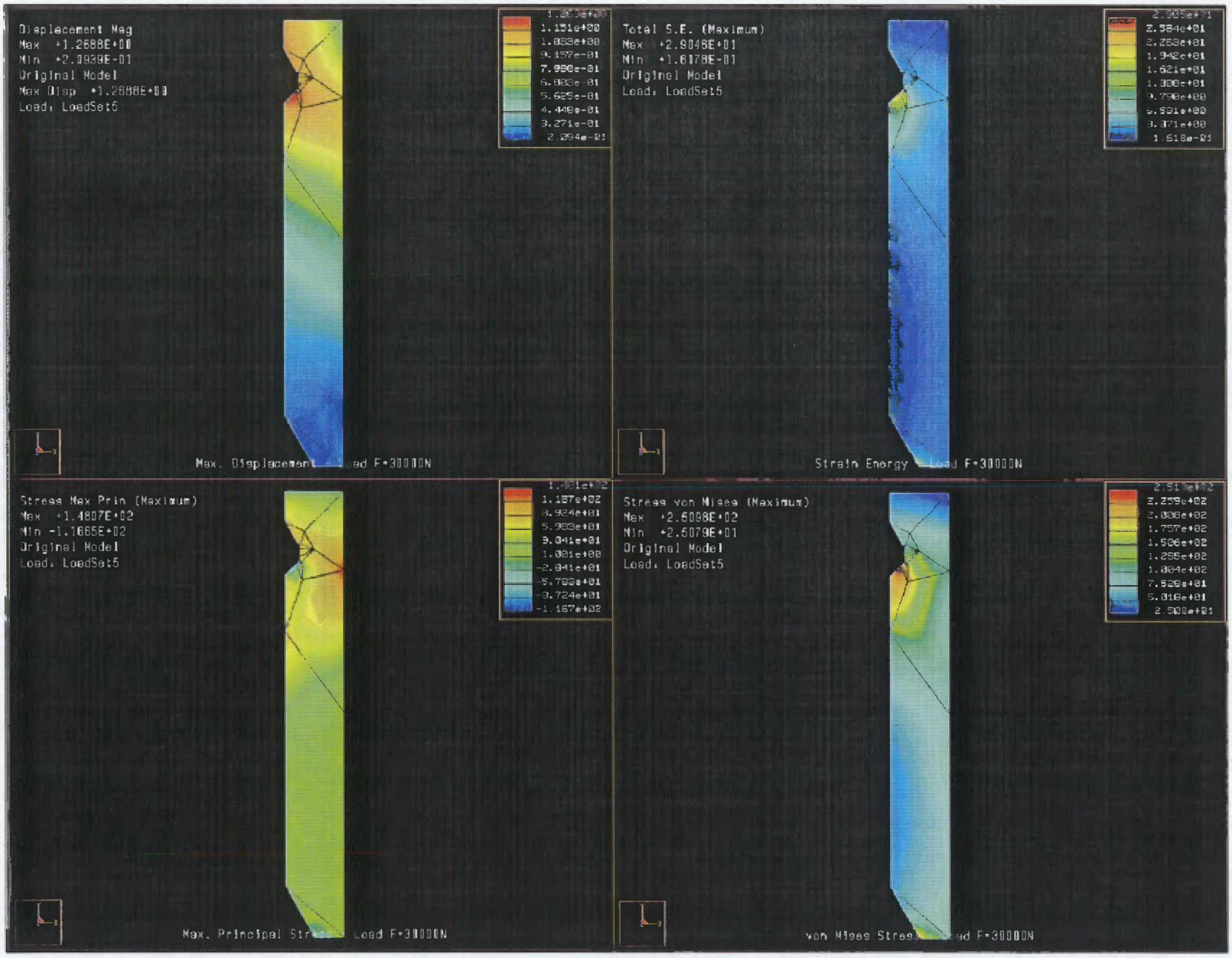


Figure E.38 Displacement, Strain Energy, Maximum Principal Stress, and von Mises Stress for a Load F of 30000N

APPENDIX F

RESULTS OF THE PARAMETER STUDY FOR THE CYLINDRICAL SNAP FIT WITH APPLIED PULLOUT FORCE W' AS REMOVAL FORCE FOR THE SHAFT AND RESULTING FORCE F FOR THE TUBE

Shaft - Case I

3. Load: W' = 500N

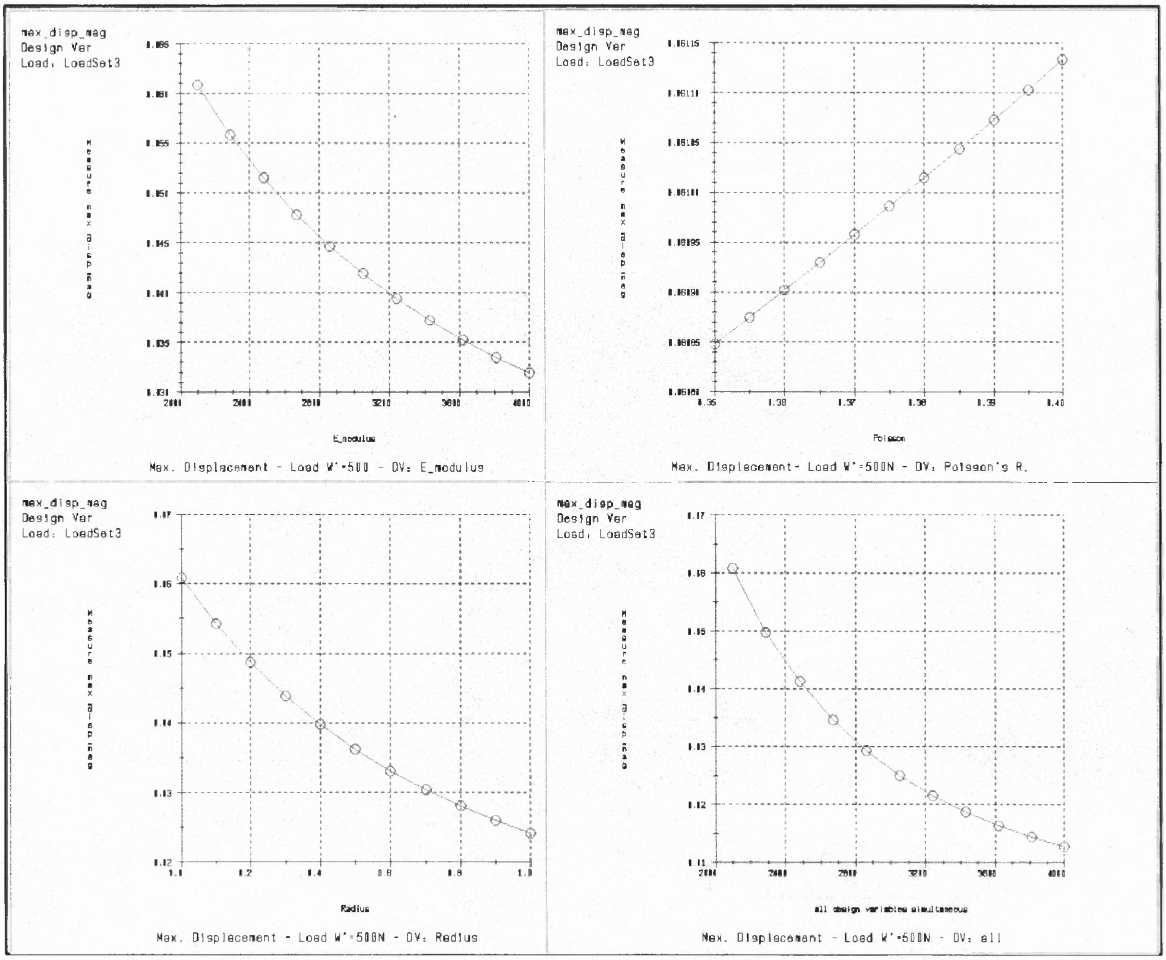


Figure F.1 Maximum Displacement Depending on Design Variable Modulus of Elasticity, Poisson's Ratio, Radius, and All Design Variables Simultaneous

SOUTHWORTH Resume 100% COTTON FIBER

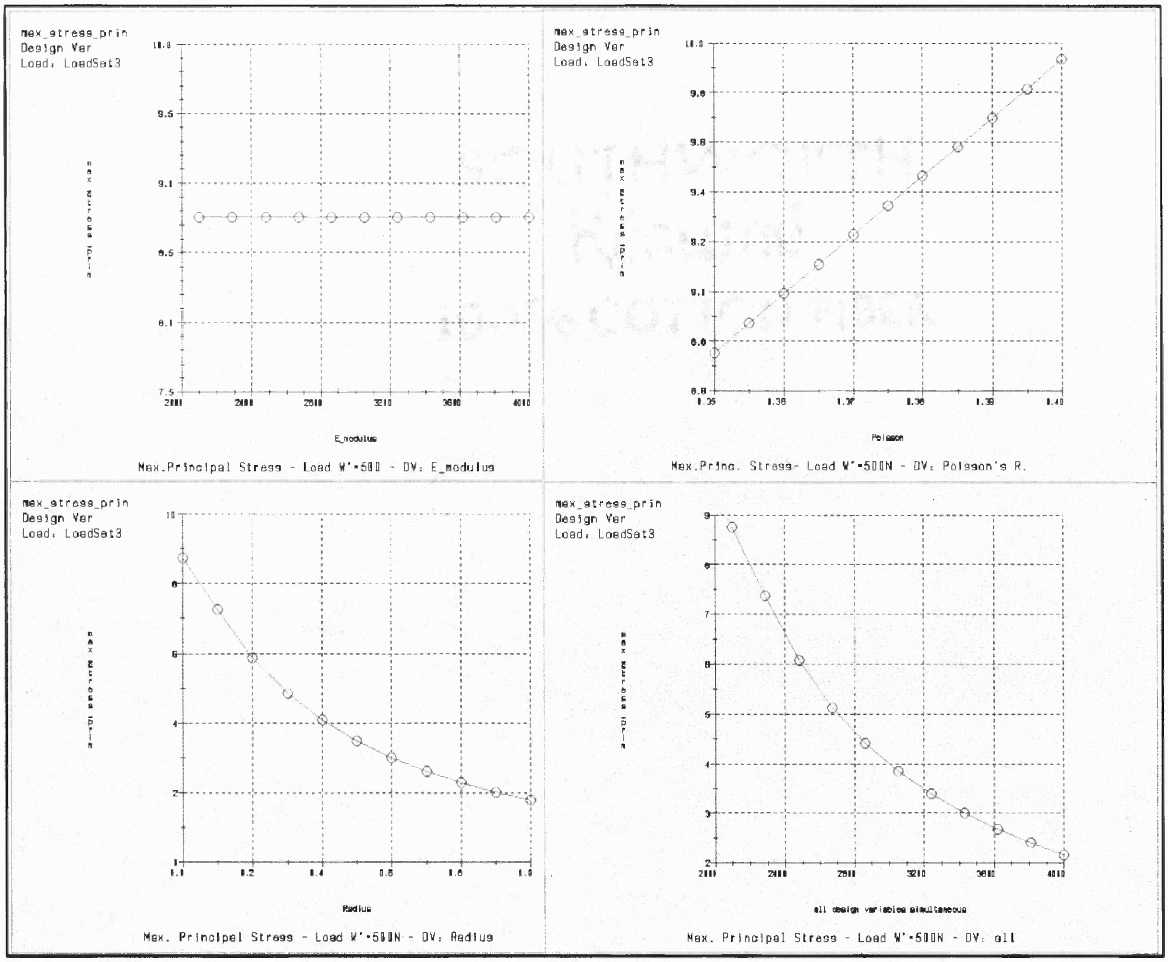


Figure F.2 Maximum Principal Stress Depending on Design Variable Modulus of Elasticity, Poisson's Ratio, Radius, and All Design Variables Simultaneous

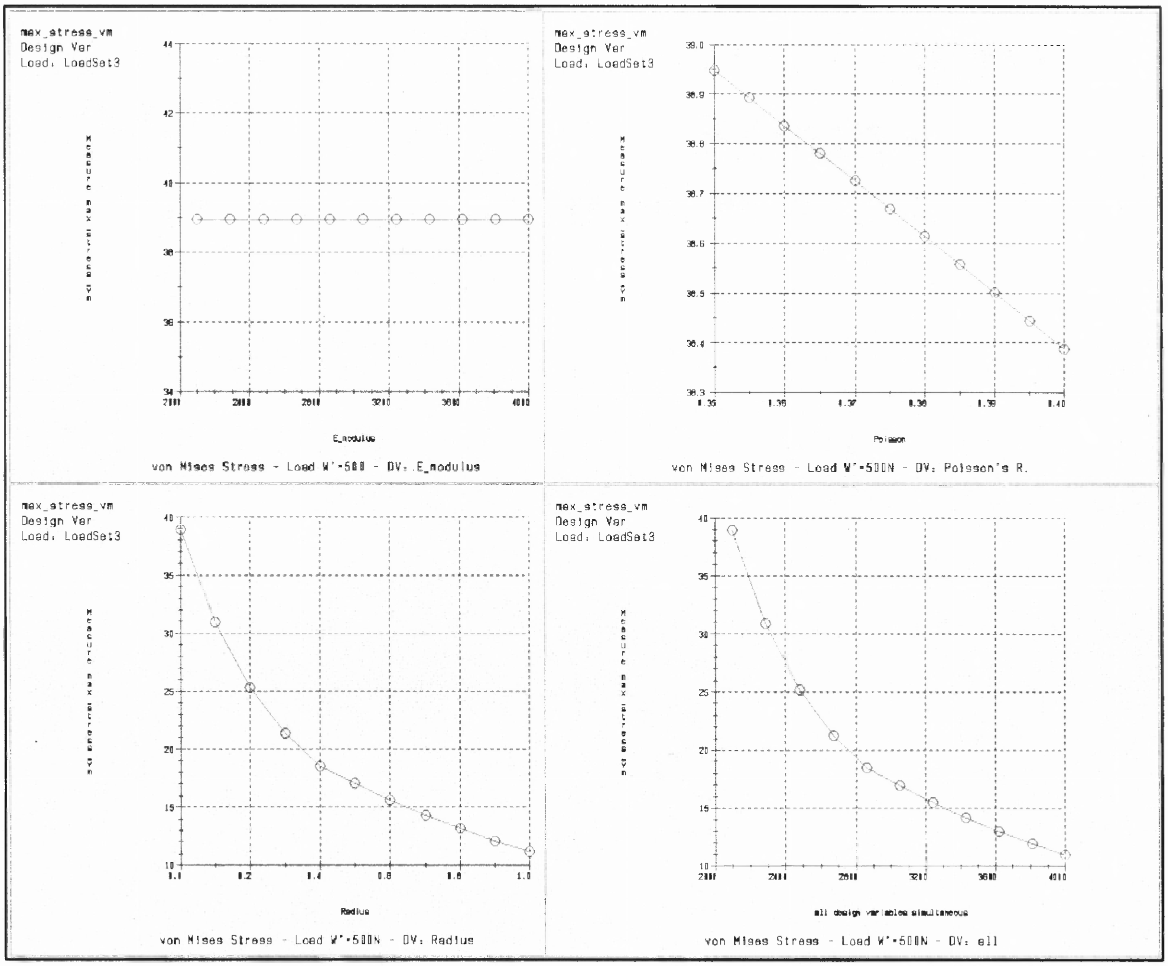


Figure F.3 Von Mises Stress Depending on Design Variable Modulus of Elasticity, Poisson's Ratio, Radius, and All Design Variables Simultaneous

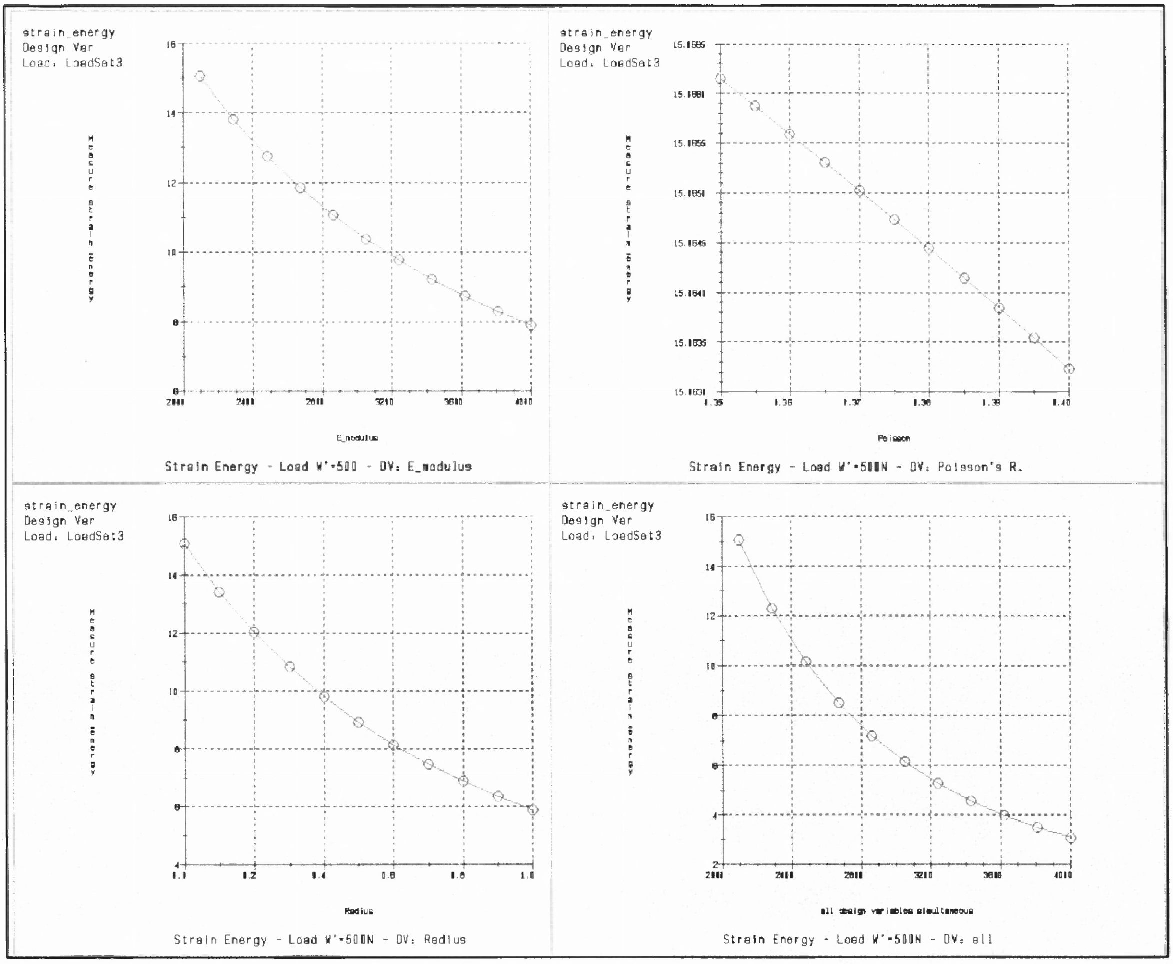


Figure F.4 Strain Energy Depending on Design Variable Modulus of Elasticity, Poisson's Ratio, Radius, and All Design Variables Simultaneous

Shaft - Case II

3. Load: W' = 2000N

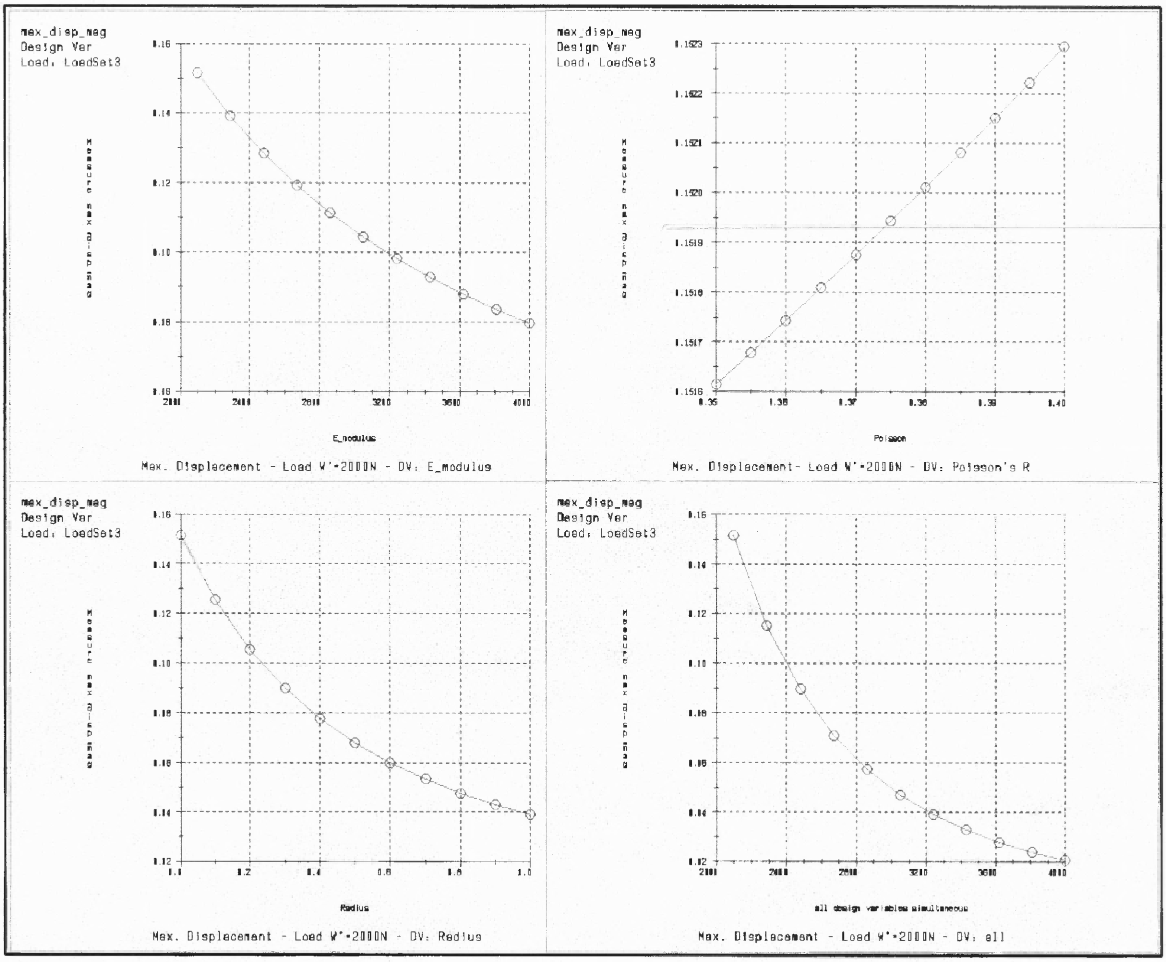


Figure F.5 Maximum Displacement Depending on Design Variable Modulus of Elasticity, Poisson's Ratio, Radius, and All Design Variables Simultaneous

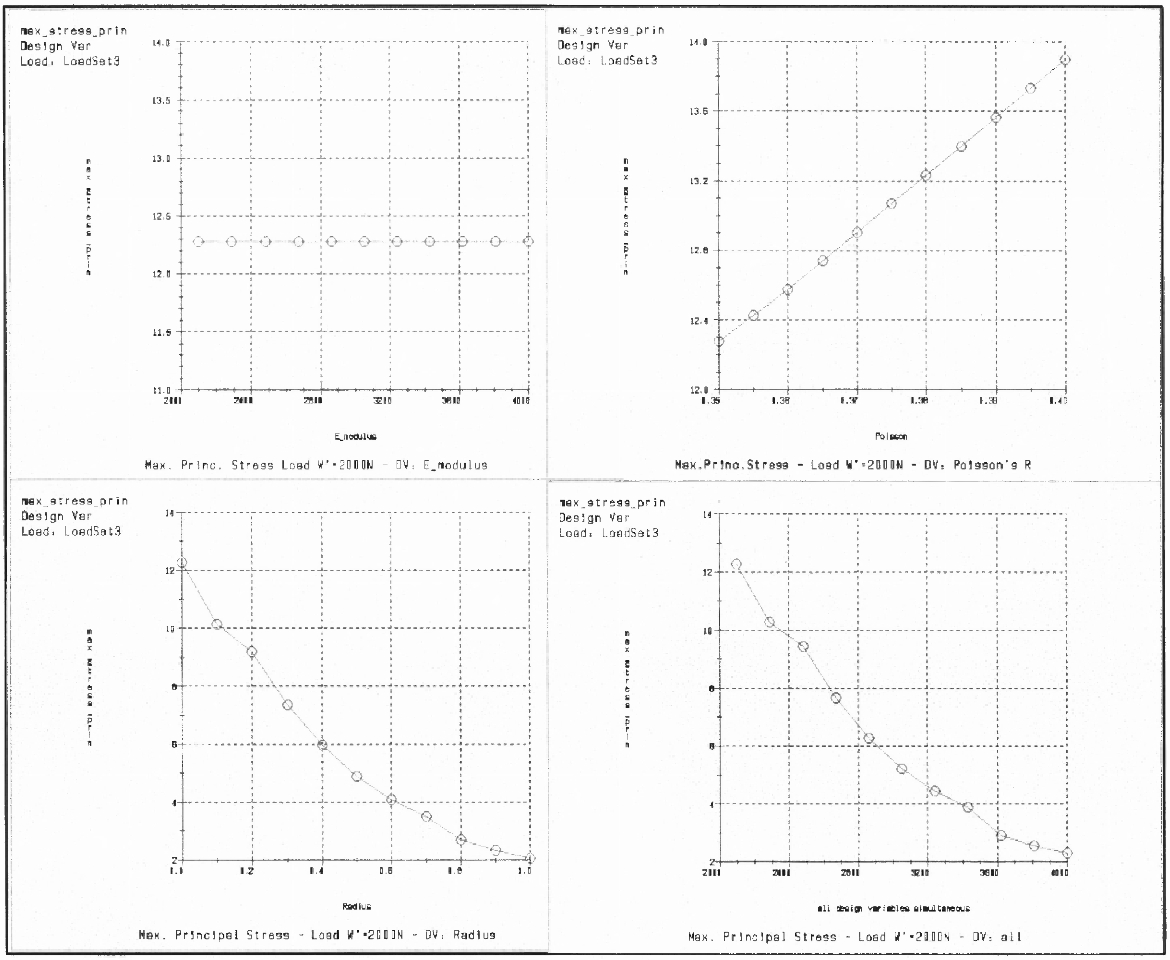


Figure F.6 Maximum Principal Stress Depending on Design Variable Modulus of Elasticity, Poisson's Ratio, Radius, and All Design Variables Simultaneous

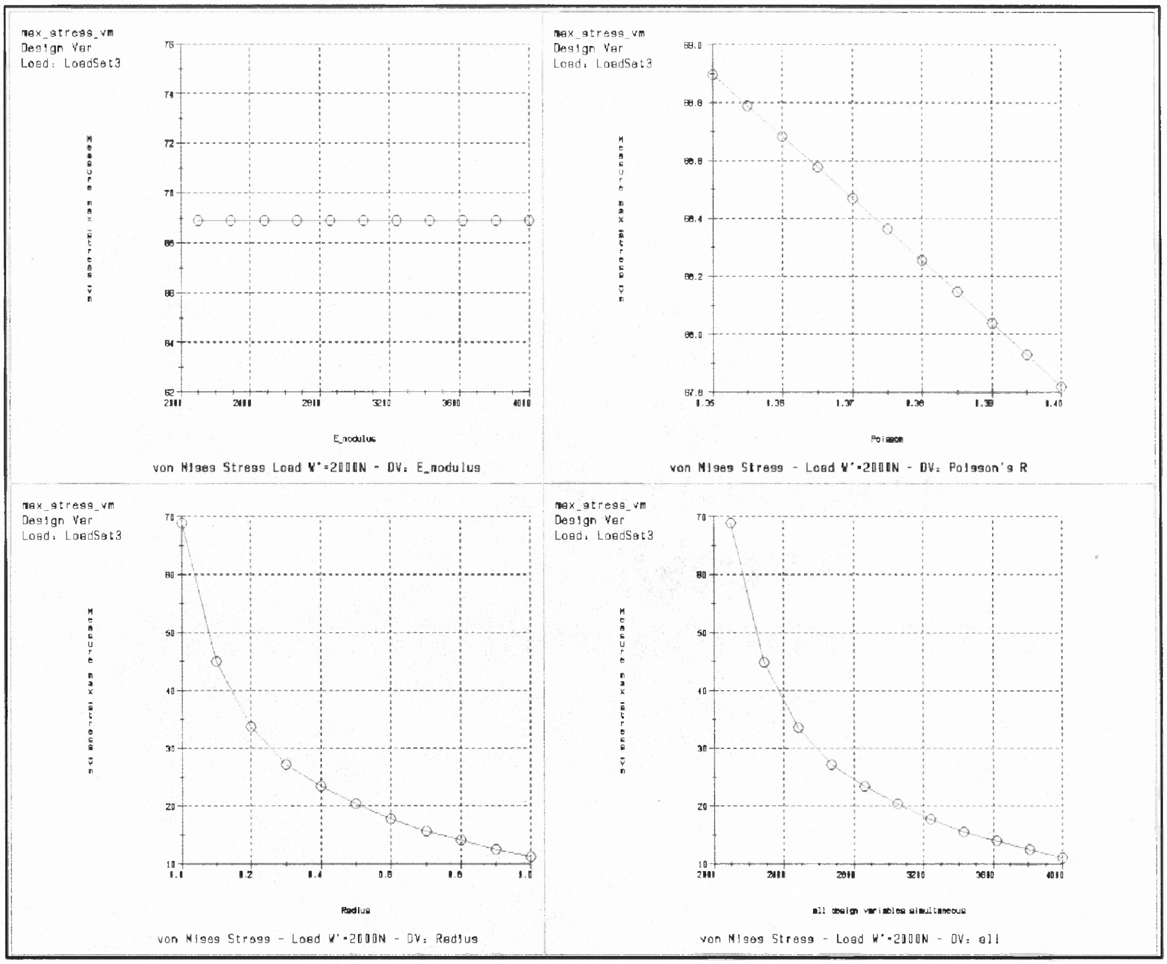


Figure F.7 Von Mises Stress Depending on Design Variable Modulus of Elasticity, Poisson's Ratio, Radius, and All Design Variables Simultaneous

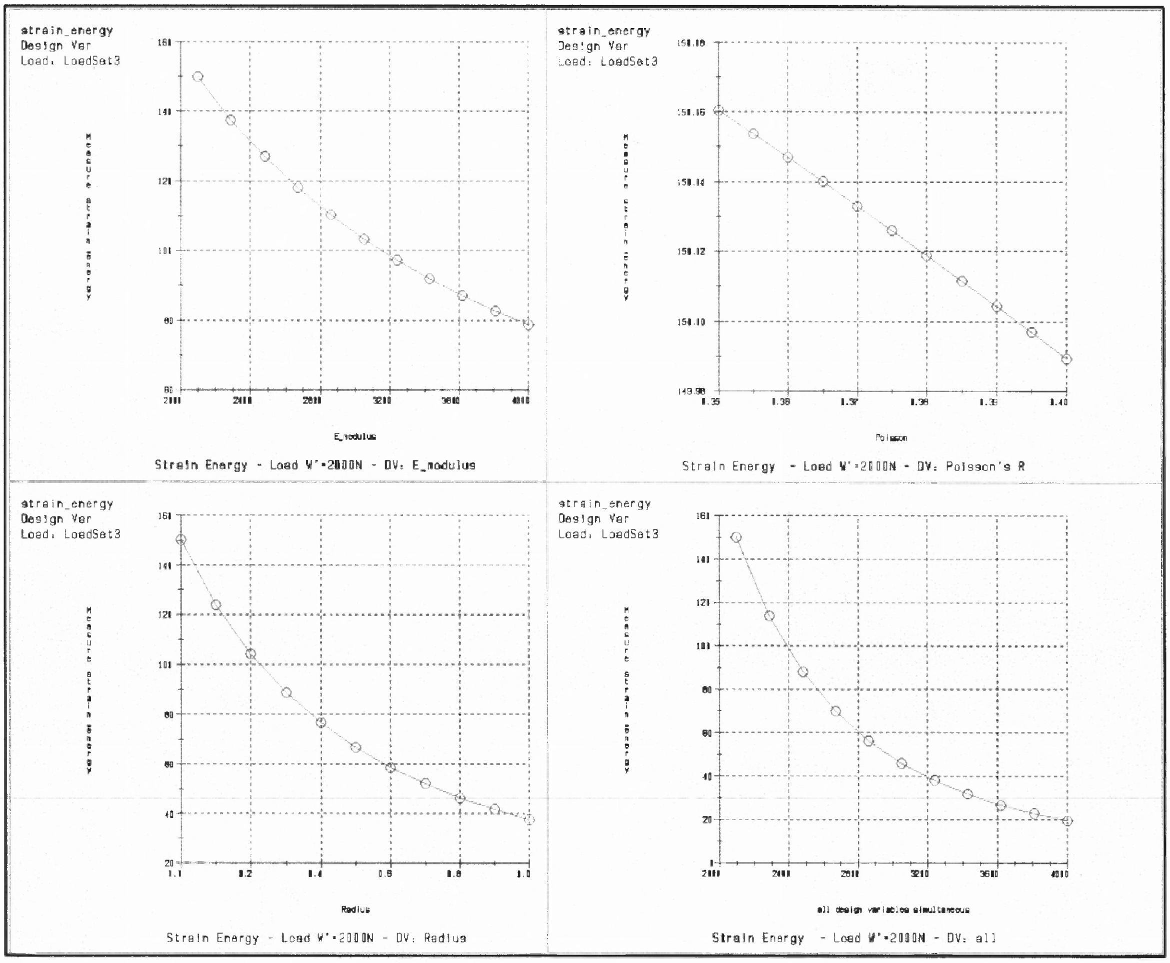


Figure F.8 Strain Energy Depending on Design Variable Modulus of Elasticity, Poisson's Ratio, Radius, and All Design Variables Simultaneous

Shaft - Case III

2. Load: W' = 4500N

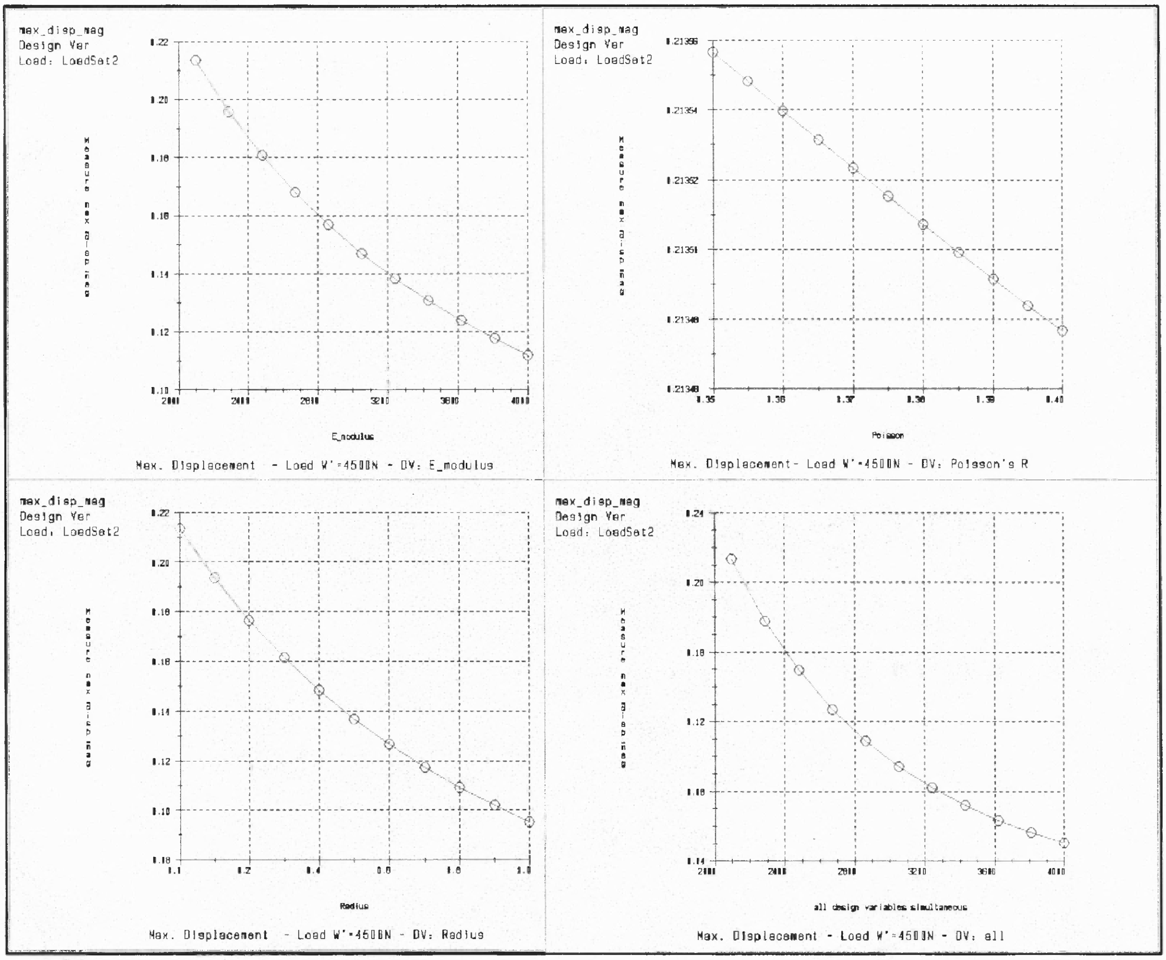


Figure F.9 Maximum Displacement Depending on Design Variable Modulus of Elasticity, Poisson's Ratio, Radius, and All Design Variables Simultaneous

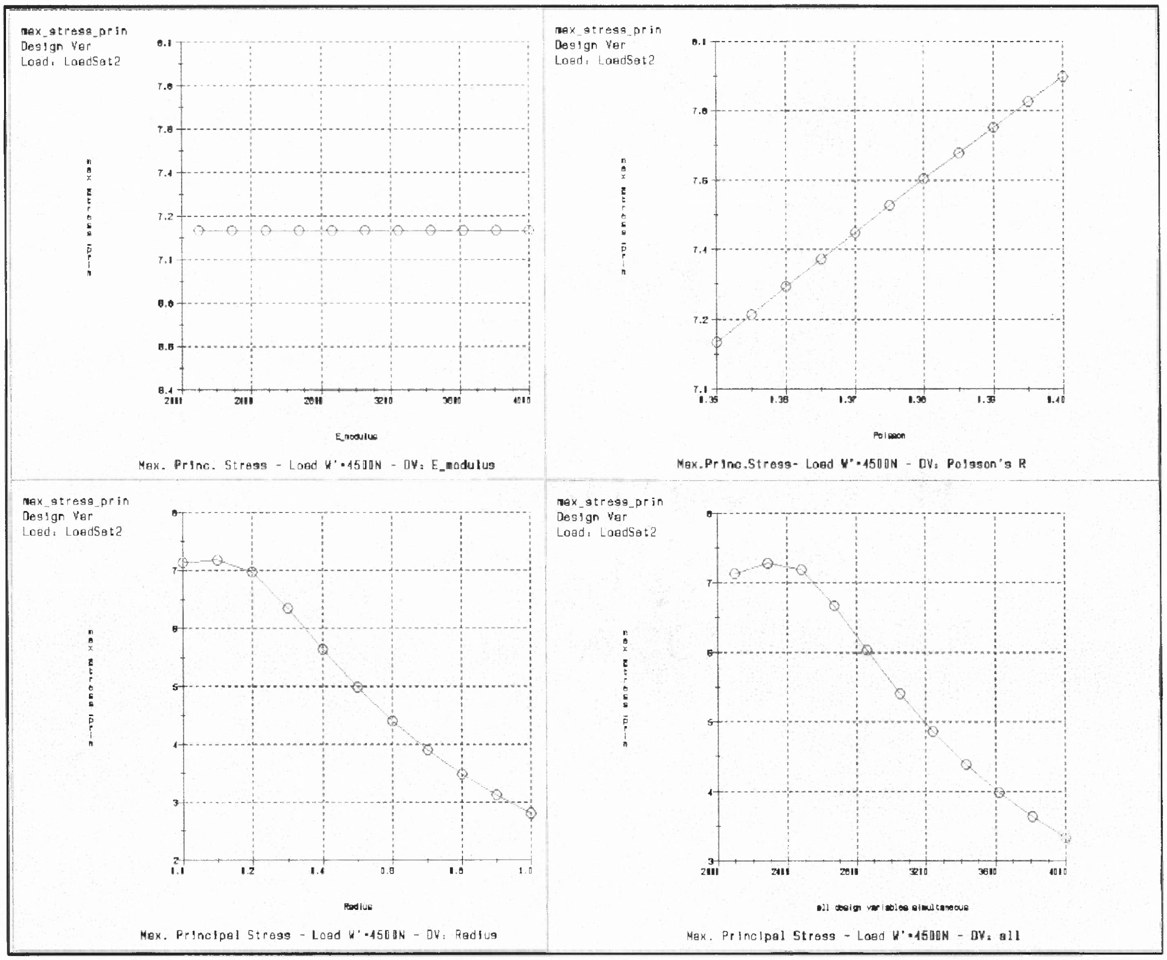


Figure F.10 Maximum Principal Stress Depending on Design Variable Modulus of Elasticity, Poisson's Ratio, Radius, and All Design Variables Simultaneous

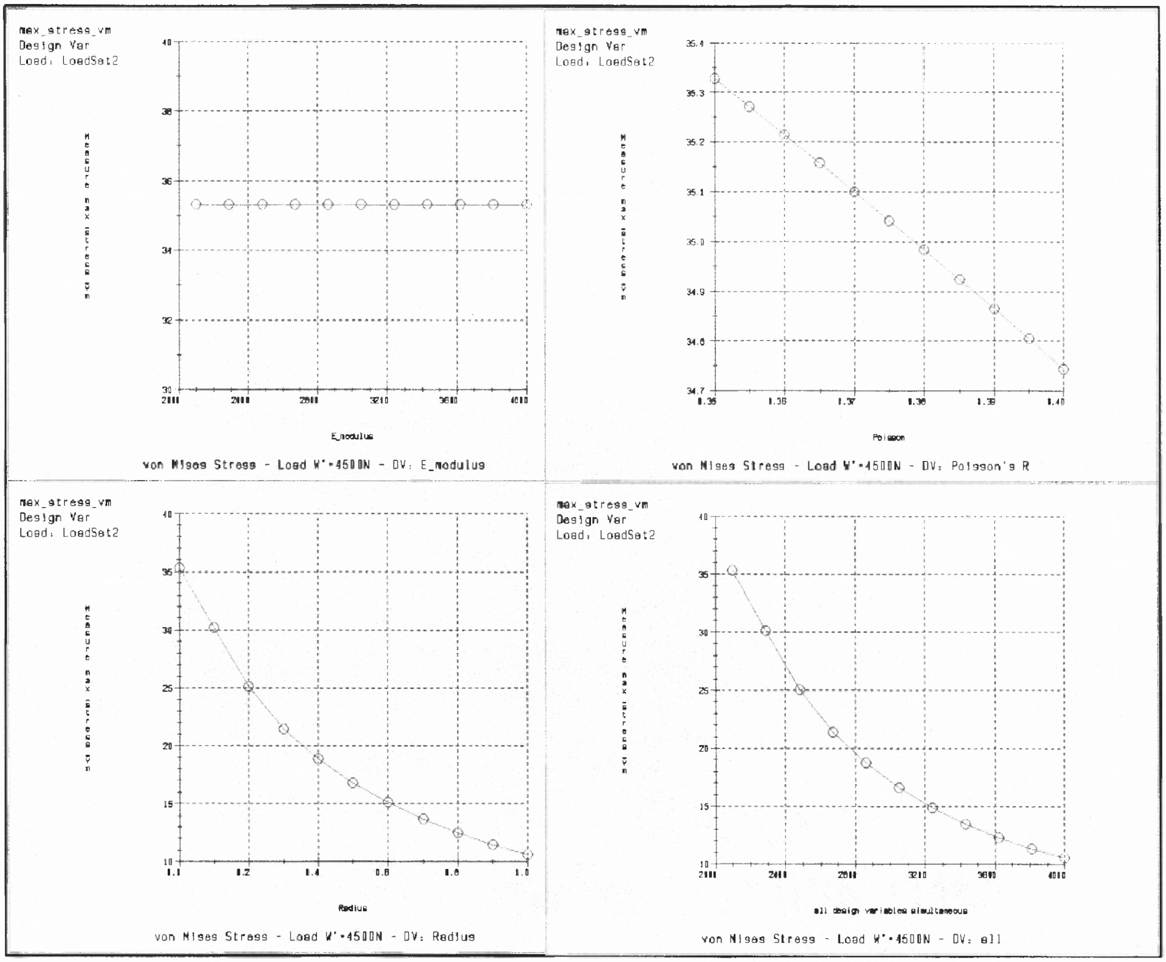


Figure F.11 Von Mises Stress Depending on Design Variable Modulus of Elasticity, Poisson's Ratio, Radius, and All Design Variables Simultaneous

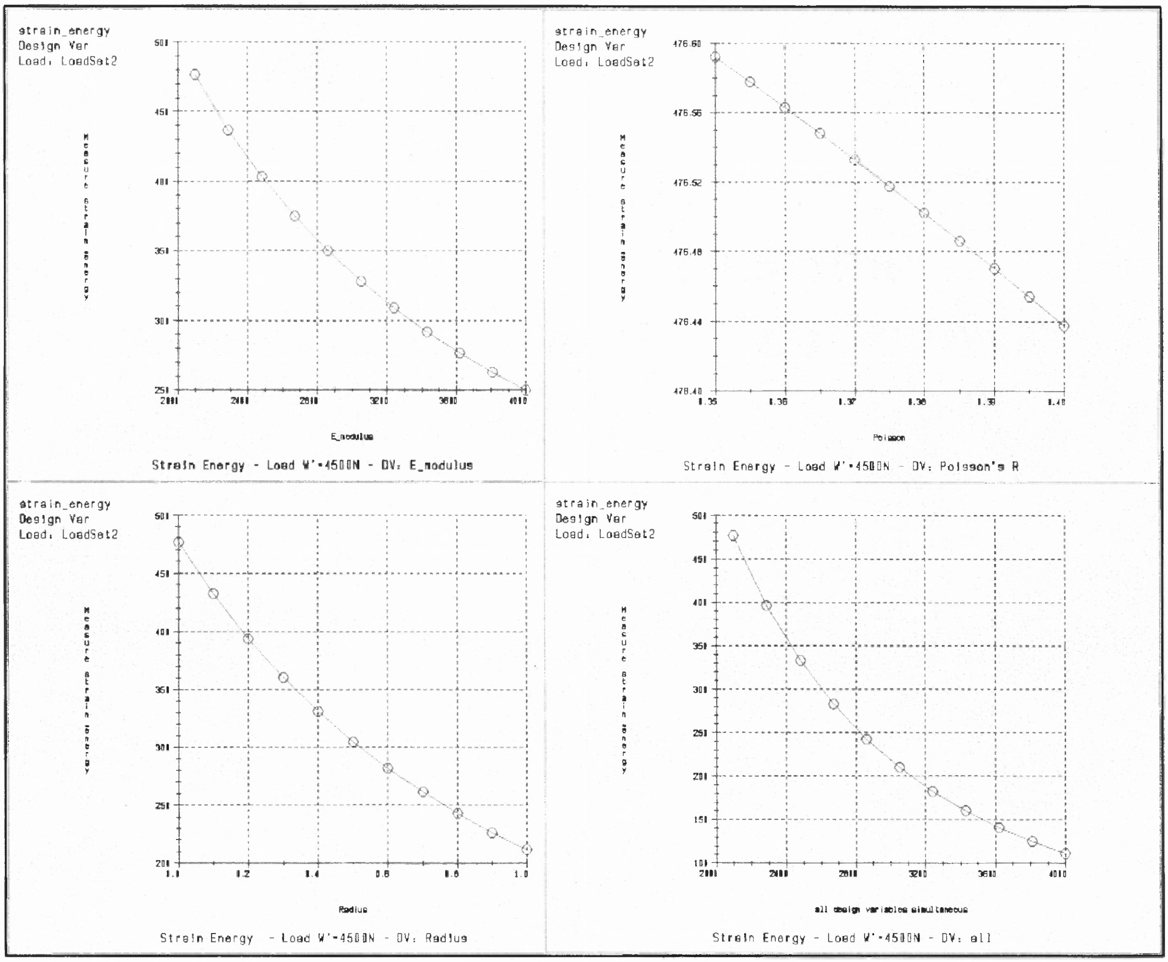


Figure F.12 Strain Energy Depending on Design Variable Modulus of Elasticity, Poisson's Ratio, Radius, and All Design Variables Simultaneous

Shaft - Case IV

2. Load: W' = 9000N

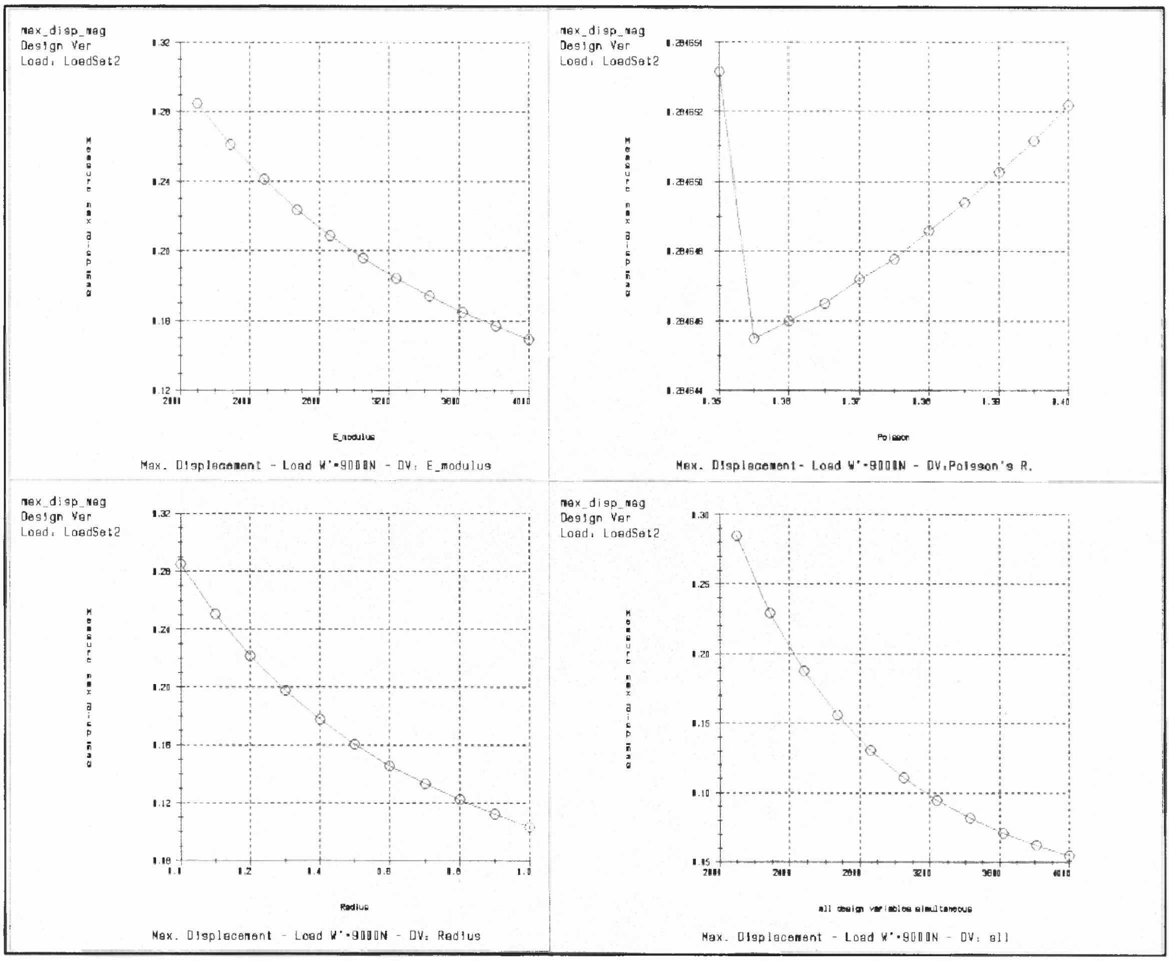


Figure F.13 Maximum Displacement Depending on Design Variable Modulus of Elasticity, Poisson's Ratio, Radius, and All Design Variables Simultaneous

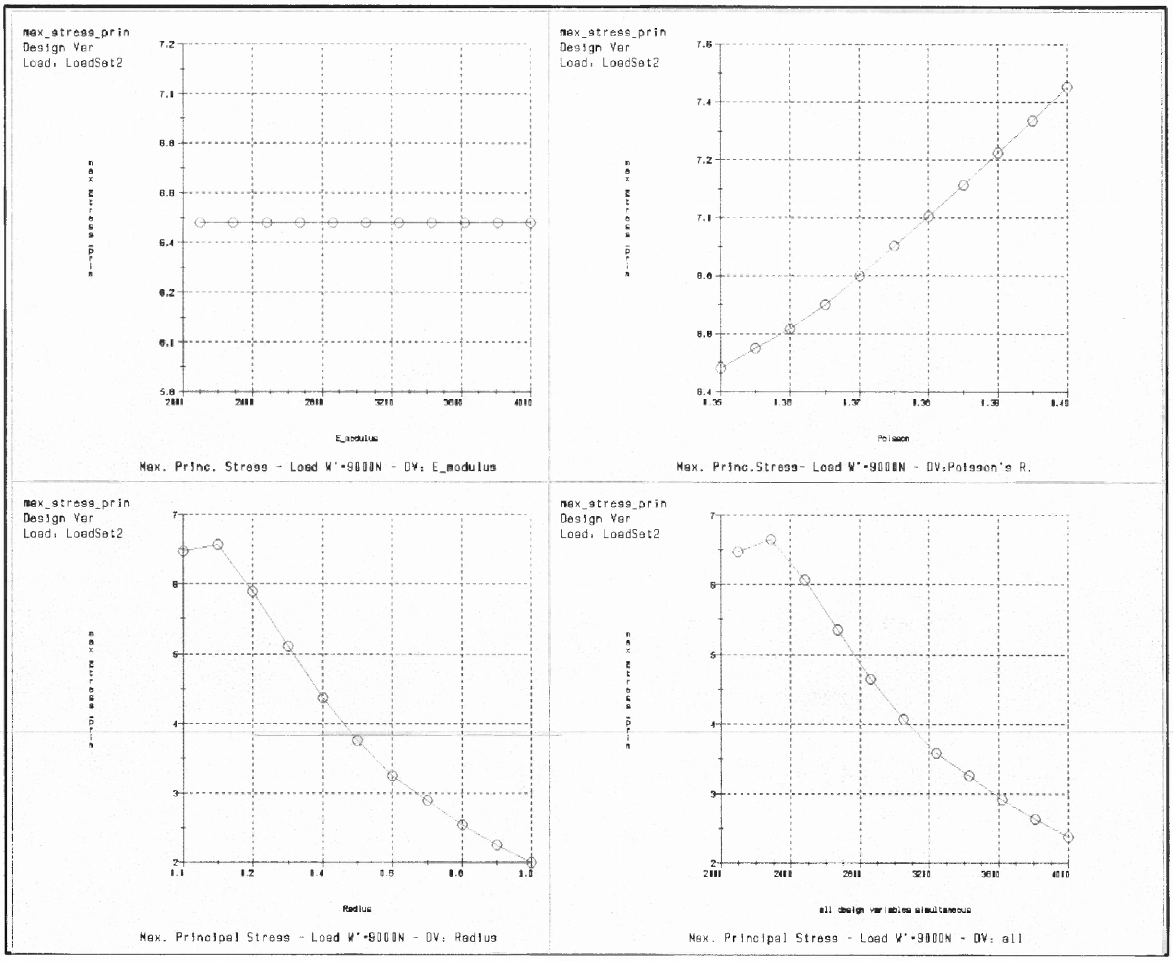


Figure F.14 Maximum Principal Stress Depending on Design Variable Modulus of Elasticity, Poisson's Ratio, Radius, and All Design Variables Simultaneous

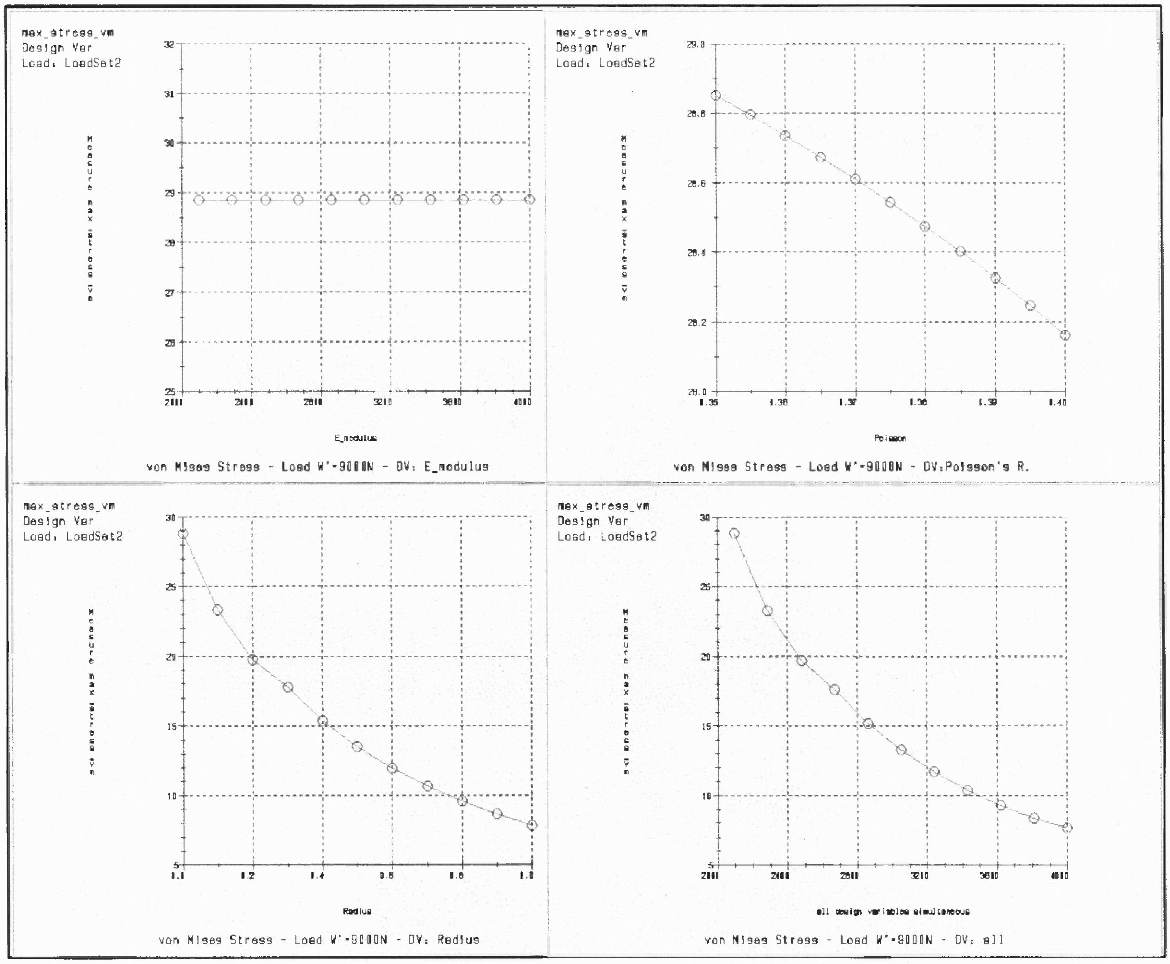


Figure F.15 Von Mises Stress Depending on Design Variable Modulus of Elasticity, Poisson's Ratio, Radius, and All Design Variables Simultaneous

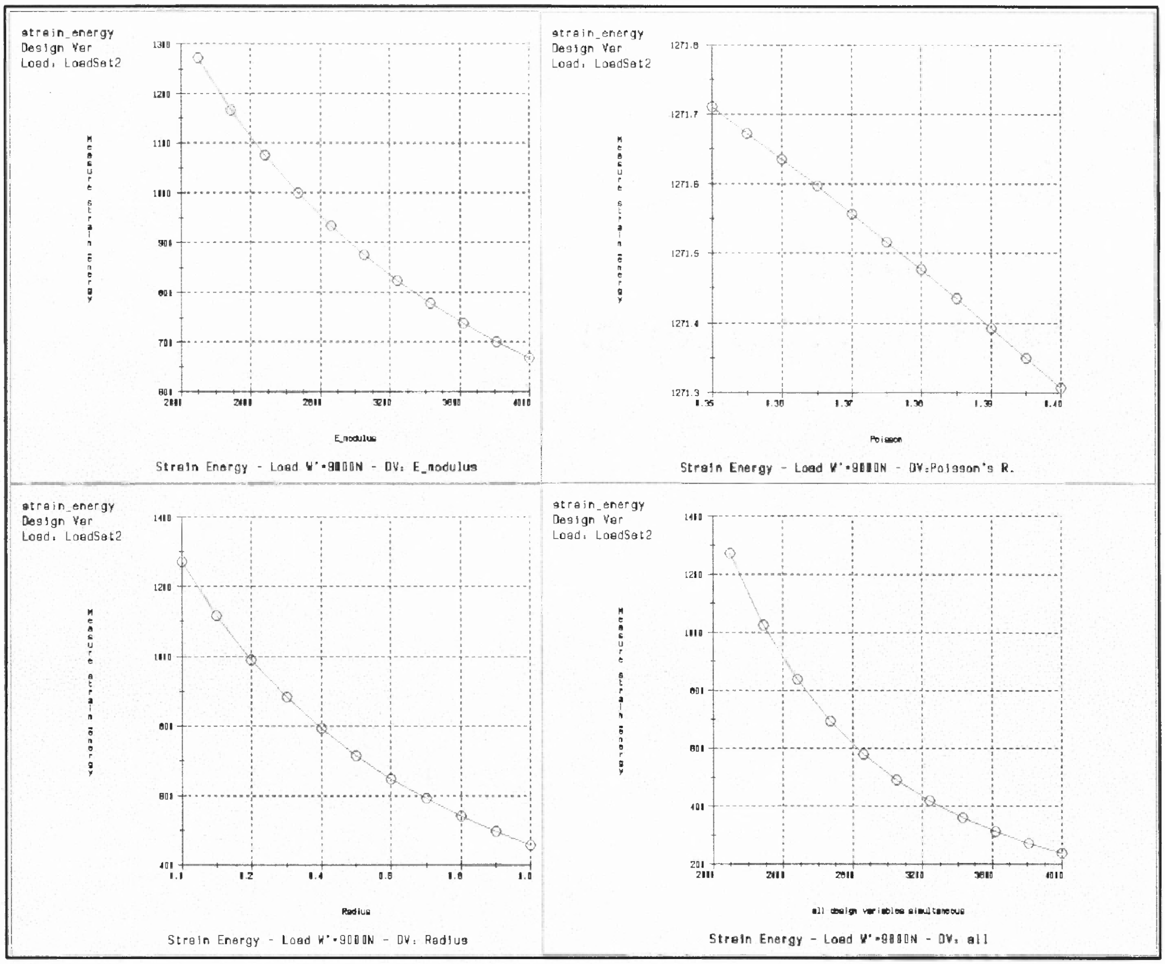


Figure F.16 Strain Energy Depending on Design Variable Modulus of Elasticity, Poisson's Ratio, Radius, and All Design Variables Simultaneous

Tube - Case I

3. Load: F = 500N

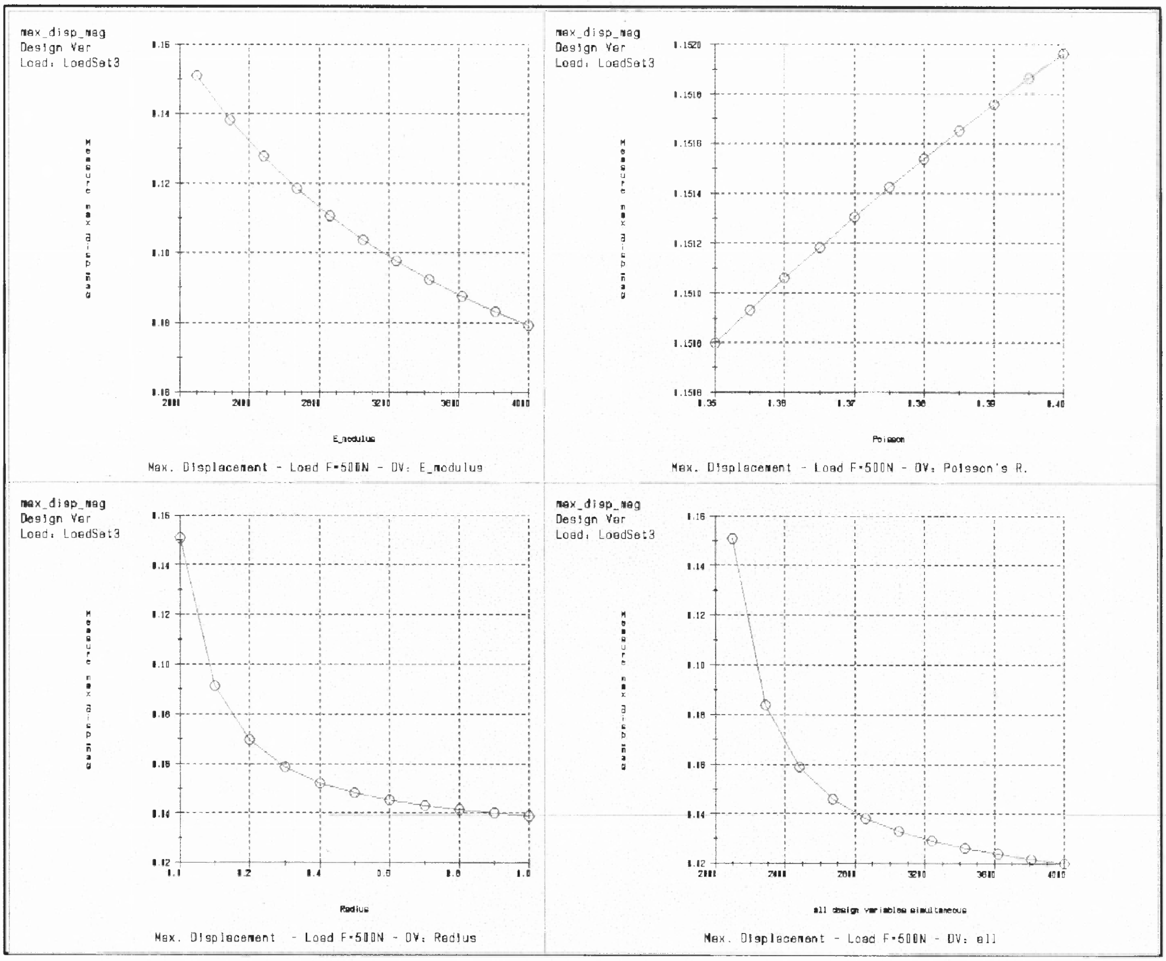


Figure F.17 Maximum Displacement Depending on Design Variable Modulus of Elasticity, Poisson's Ratio, Radius, and All Design Variables Simultaneous

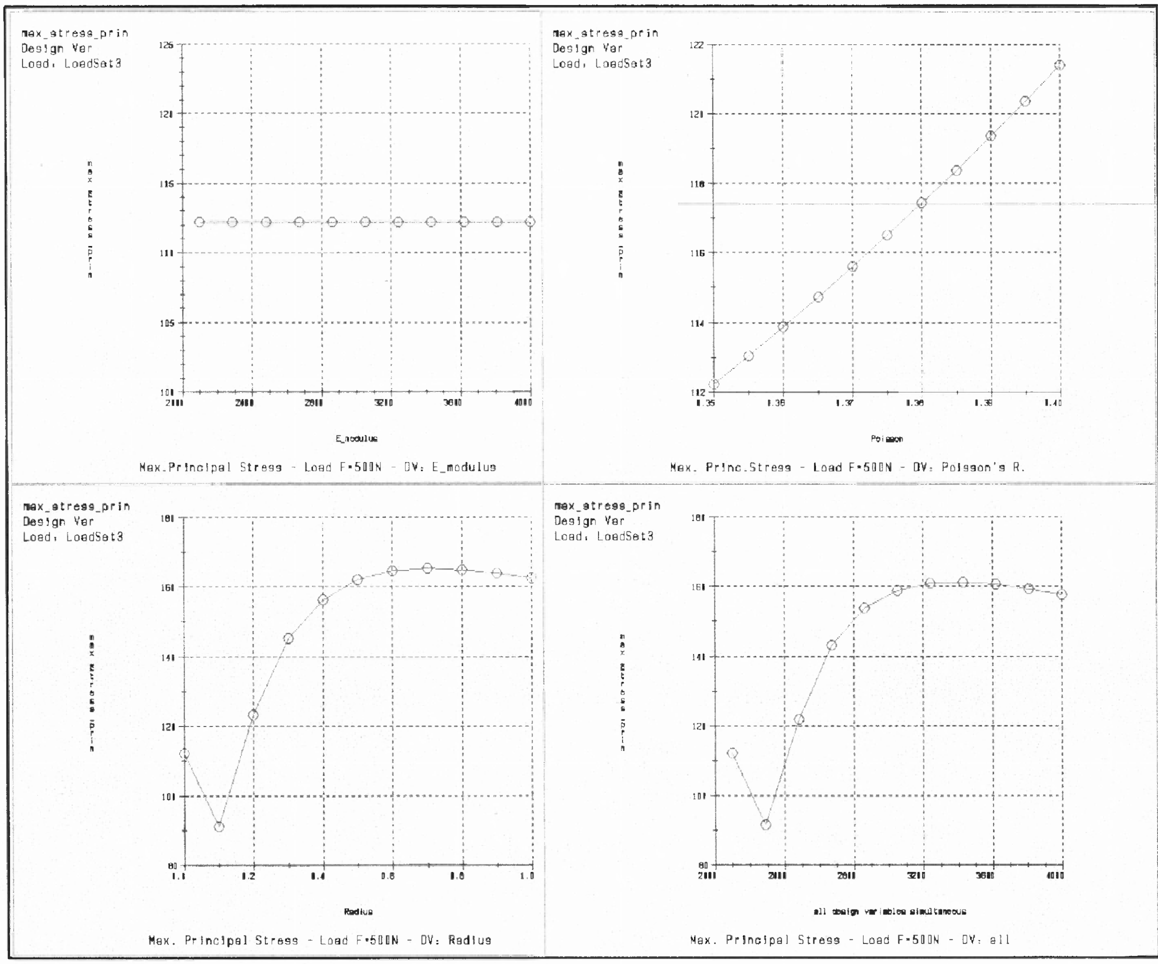


Figure F.18 Maximum Principal Stress Depending on Design Variable Modulus of Elasticity, Poisson's Ratio, Radius, and All Design Variables Simultaneous

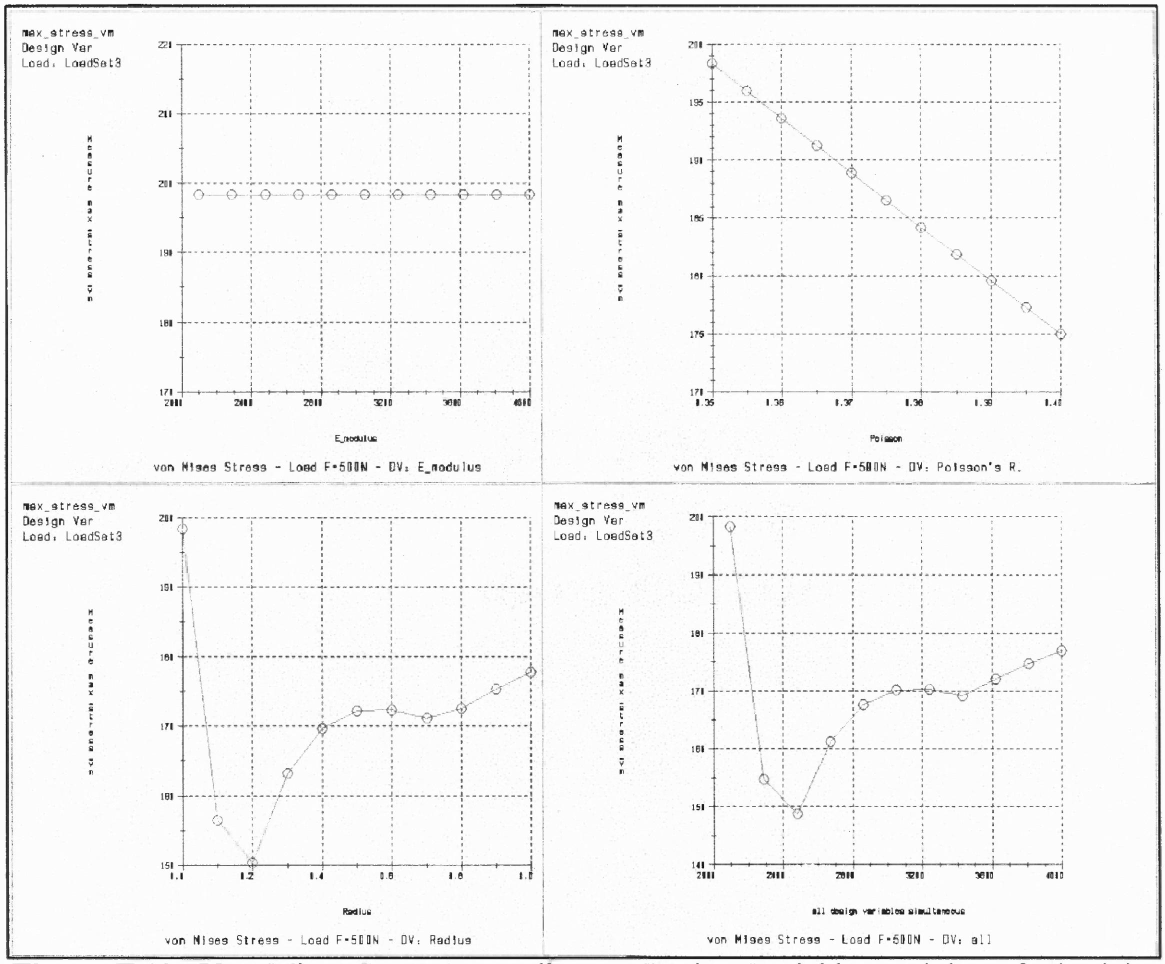


Figure F.19 Von Mises Stress Depending on Design Variable Modulus of Elasticity, Poisson's Ratio, Radius, and All Design Variables Simultaneous

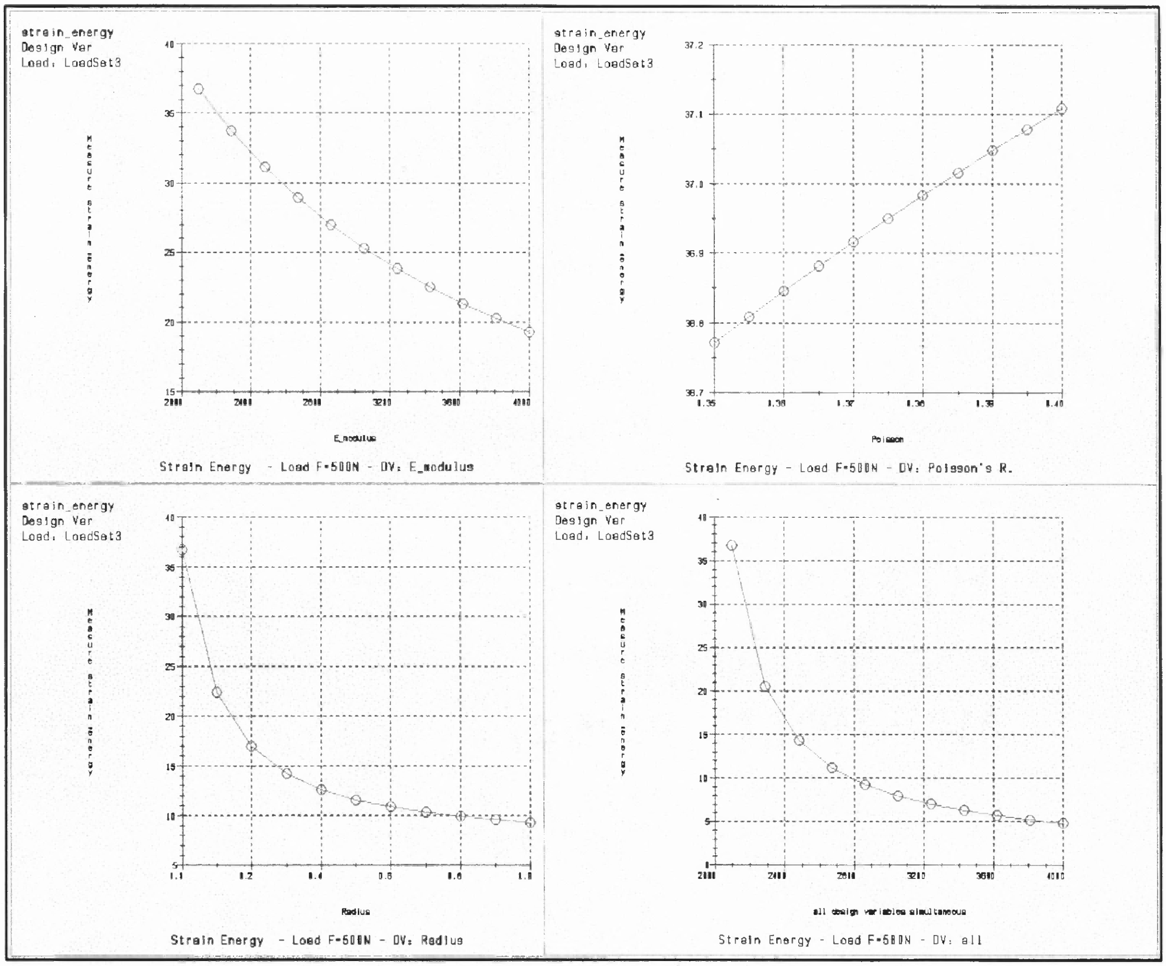


Figure F.20 Strain Energy Depending on Design Variable Modulus of Elasticity, Poisson's Ratio, Radius, and All Design Variables Simultaneous

Tube - Case II

3. Load: F = 2000N

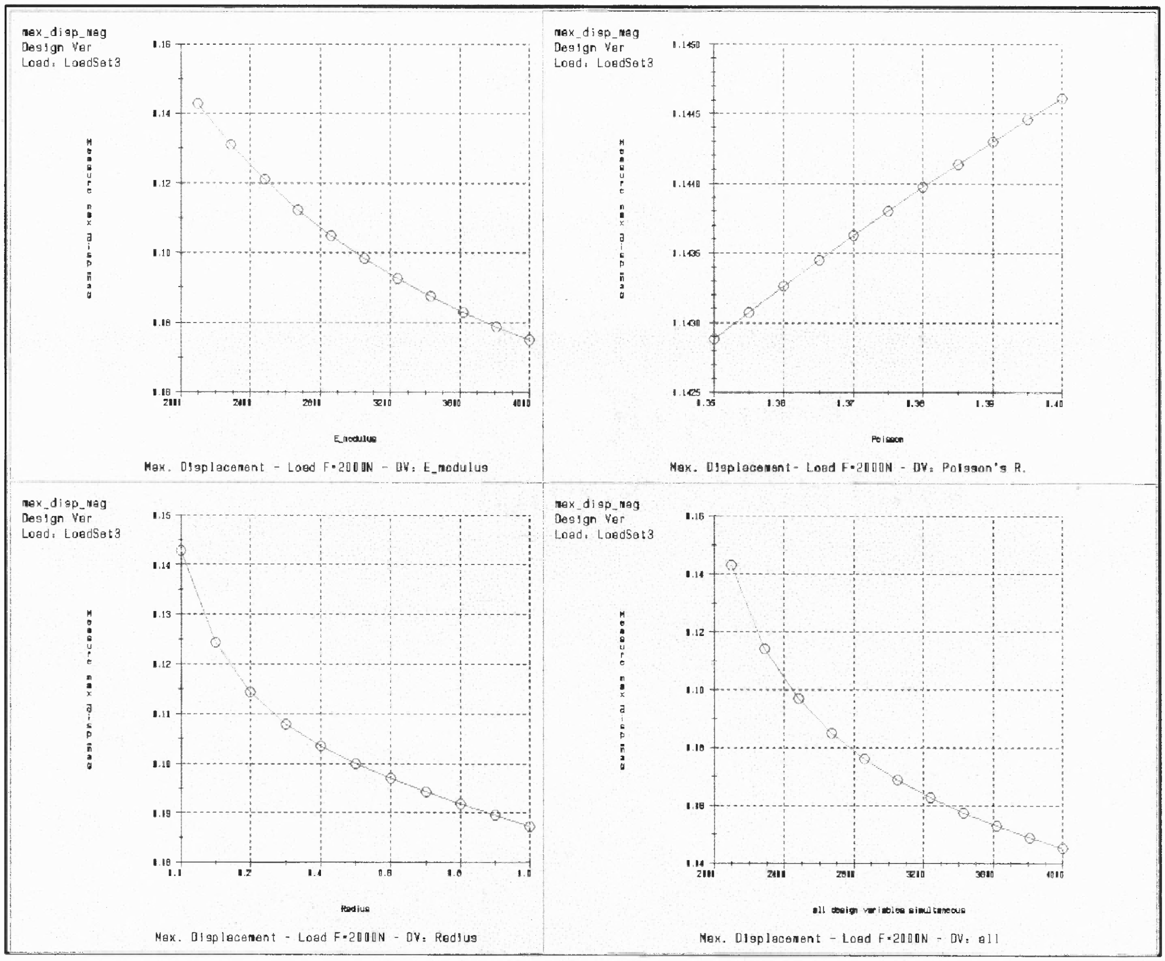


Figure F.21 Maximum Displacement Depending on Design Variable Modulus of Elasticity, Poisson's Ratio, Radius, and All Design Variables Simultaneous

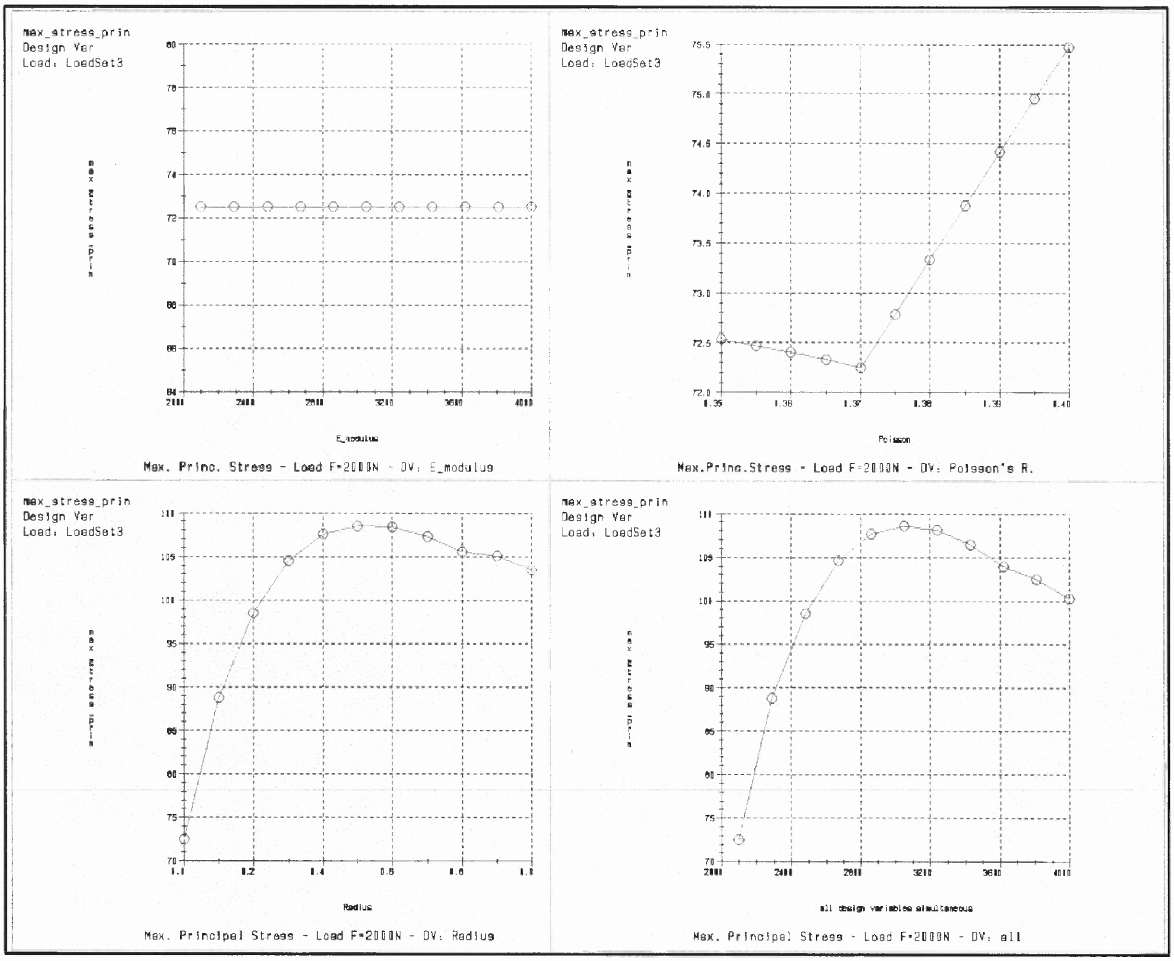


Figure F.22 Maximum Principal Stress Depending on Design Variable Modulus of Elasticity, Poisson's Ratio, Radius, and All Design Variables Simultaneous

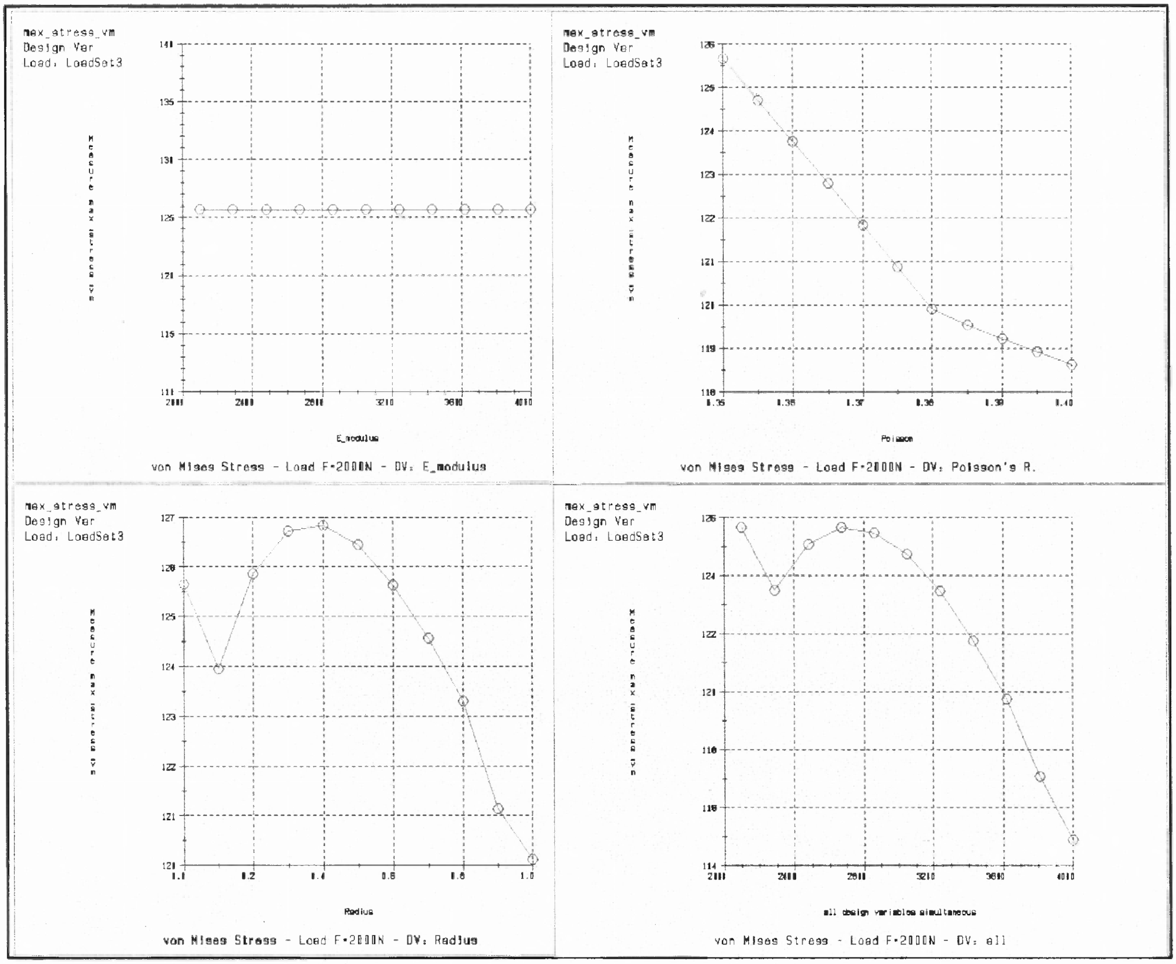


Figure F.23 Von Mises Stress Depending on Design Variable Modulus of Elasticity, Poisson's Ratio, Radius, and All Design Variables Simultaneous

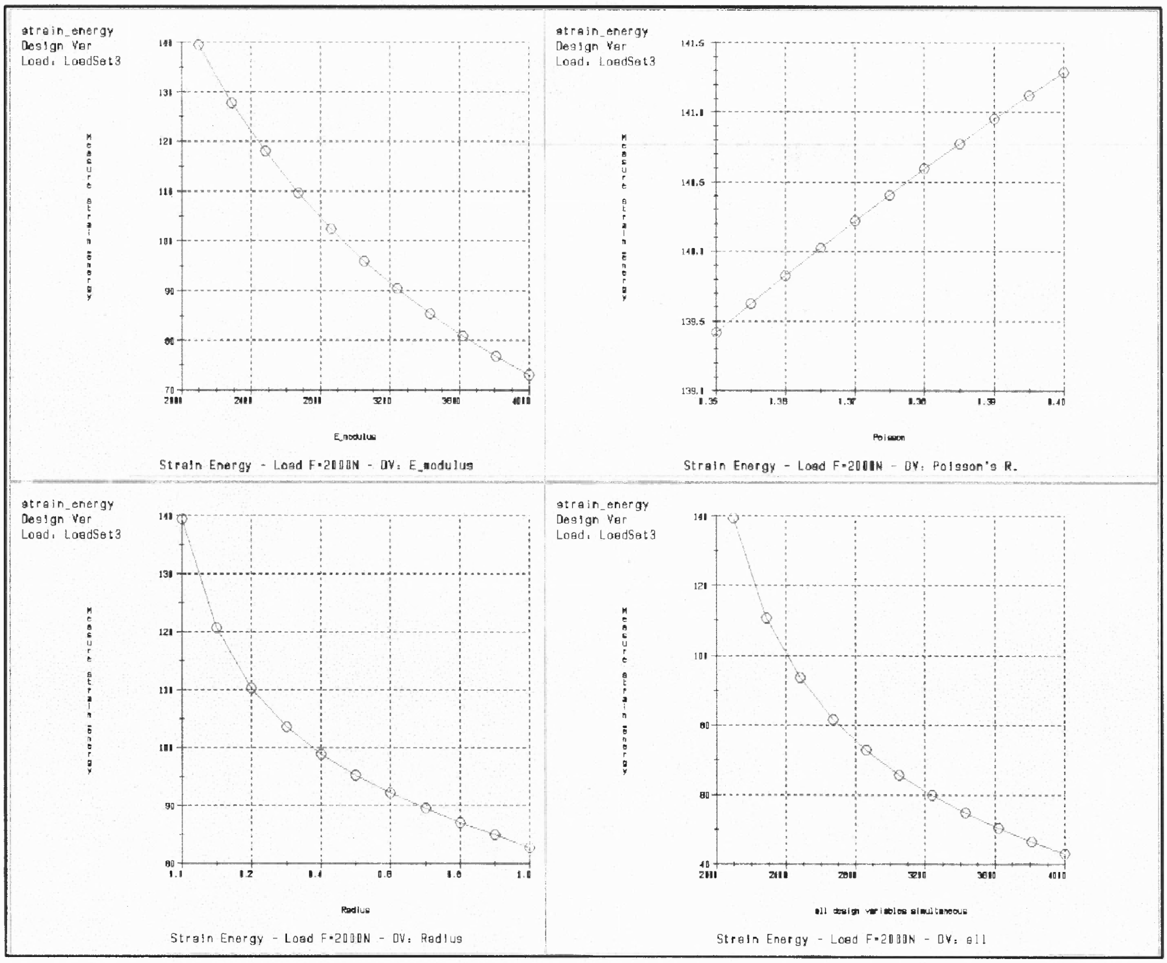


Figure F.24 Strain Energy Depending on Design Variable Modulus of Elasticity, Poisson's Ratio, Radius, and All Design Variables Simultaneous

Tube - Case III

2. Load: F = 4500N

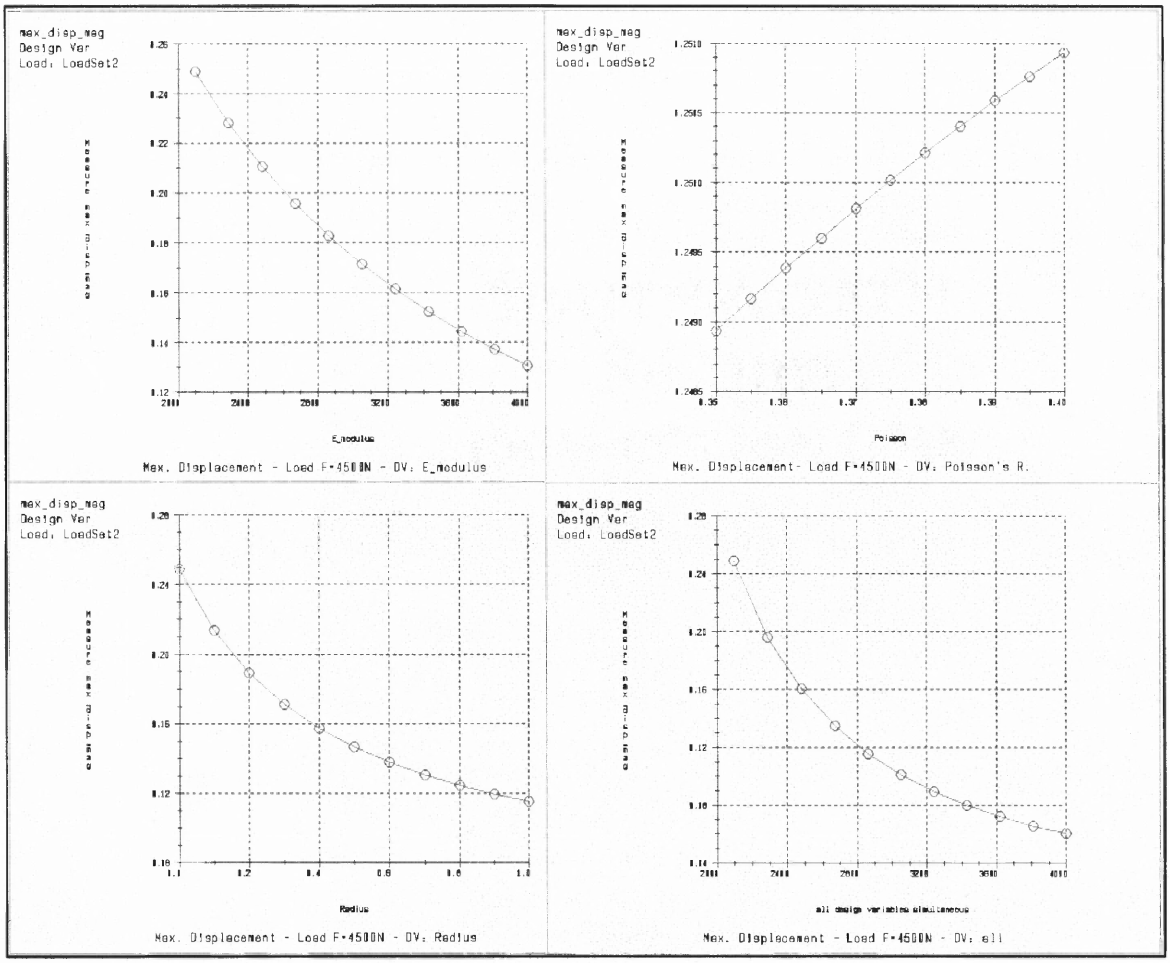


Figure F.25 Maximum Displacement Depending on Design Variable Modulus of Elasticity, Poisson's Ratio, Radius, and All Design Variables Simultaneous

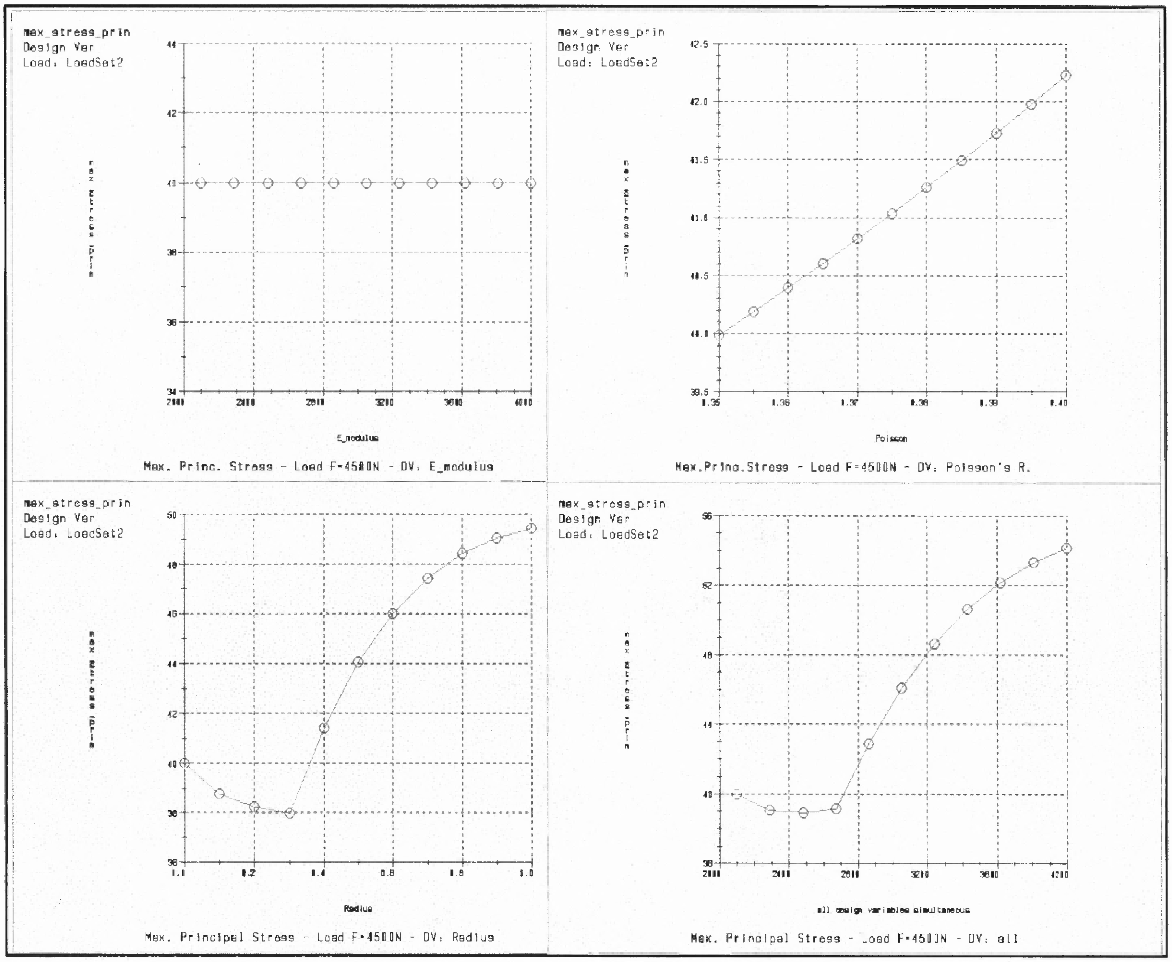


Figure F.26 Maximum Principal Stress Depending on Design Variable Modulus of Elasticity, Poisson's Ratio, Radius, and All Design Variables Simultaneous

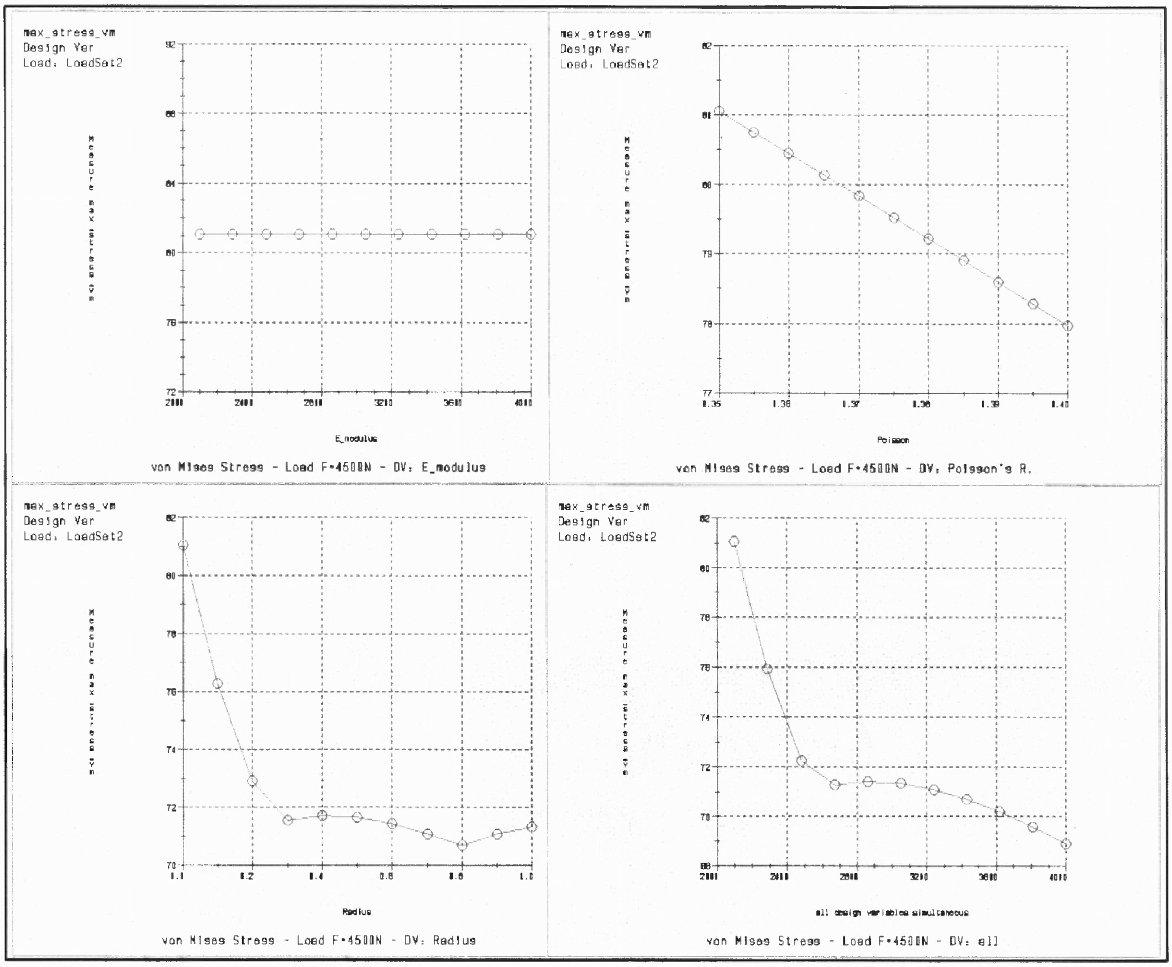


Figure F.27 Von Mises Stress Depending on Design Variable Modulus of Elasticity, Poisson's Ratio, Radius, and All Design Variables Simultaneous

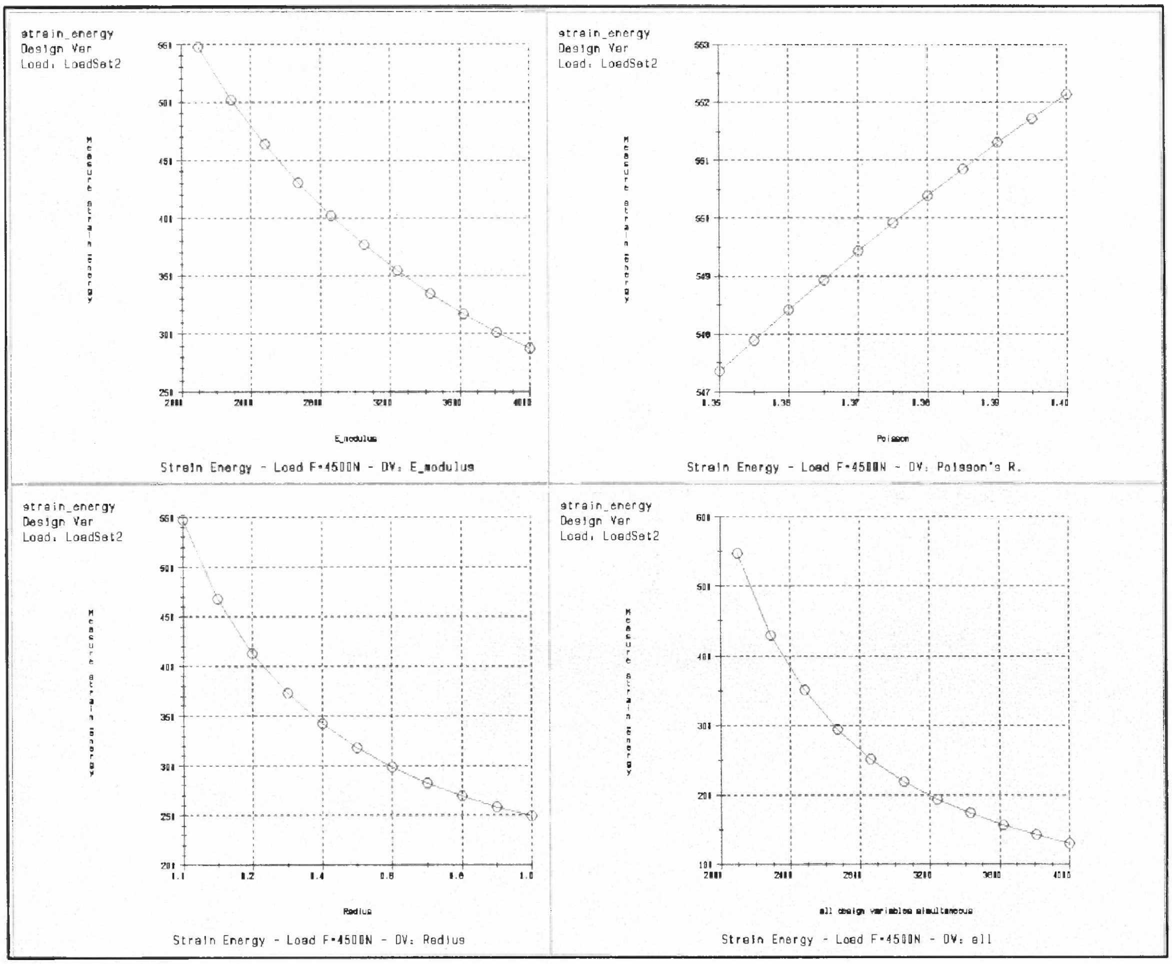


Figure F.28 Strain Energy Depending on Design Variable Modulus of Elasticity, Poisson's Ratio, Radius, and All Design Variables Simultaneous

Tube - Case IV

2. Load: F = 9000N

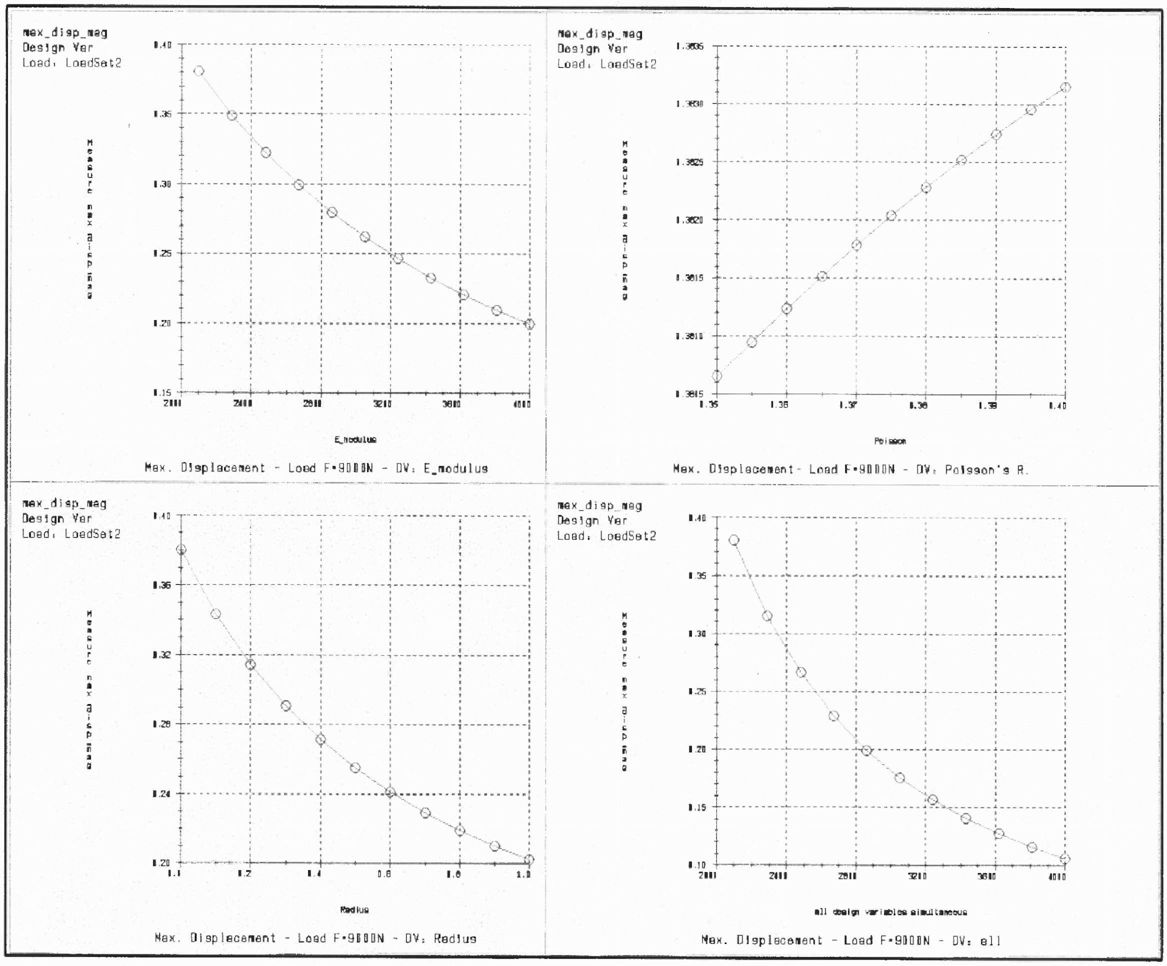


Figure F.29 Maximum Displacement Depending on Design Variable Modulus of Elasticity, Poisson's Ratio, Radius, and All Design Variables Simultaneous

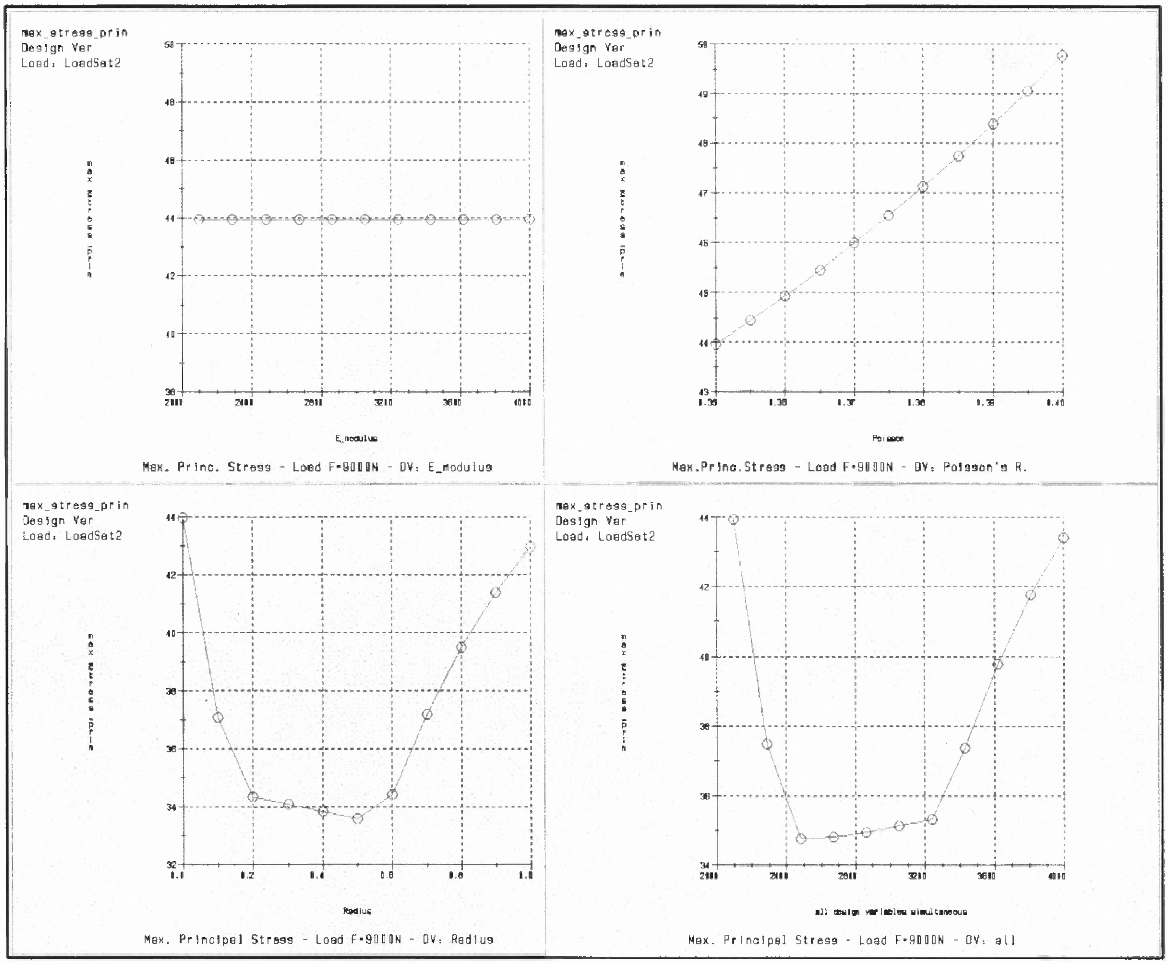


Figure F.30 Maximum Principal Stress Depending on Design Variable Modulus of Elasticity, Poisson's Ratio, Radius, and All Design Variables Simultaneous

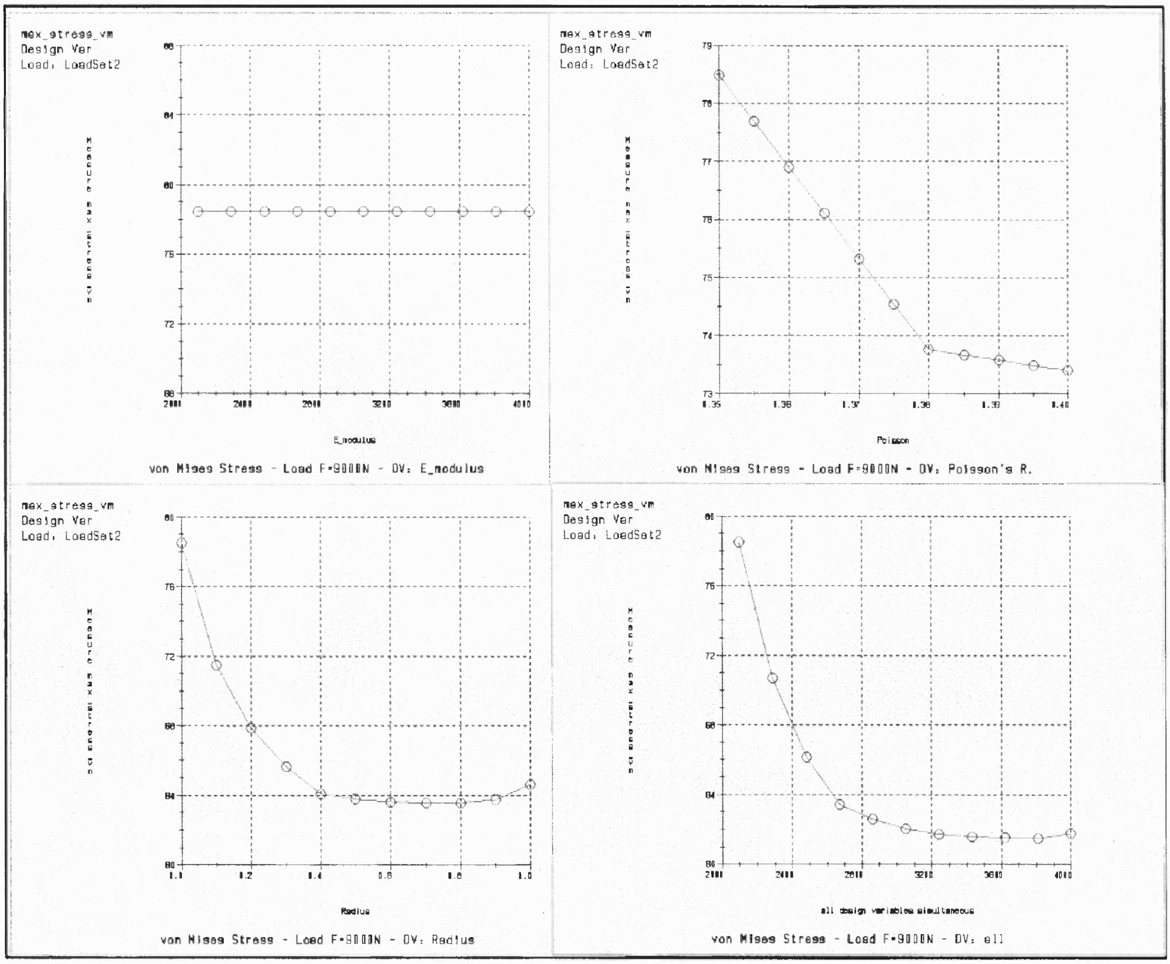


Figure F.31 Von Mises Stress Depending on Design Variable Modulus of Elasticity, Poisson's Ratio, Radius, and All Design Variables Simultaneous

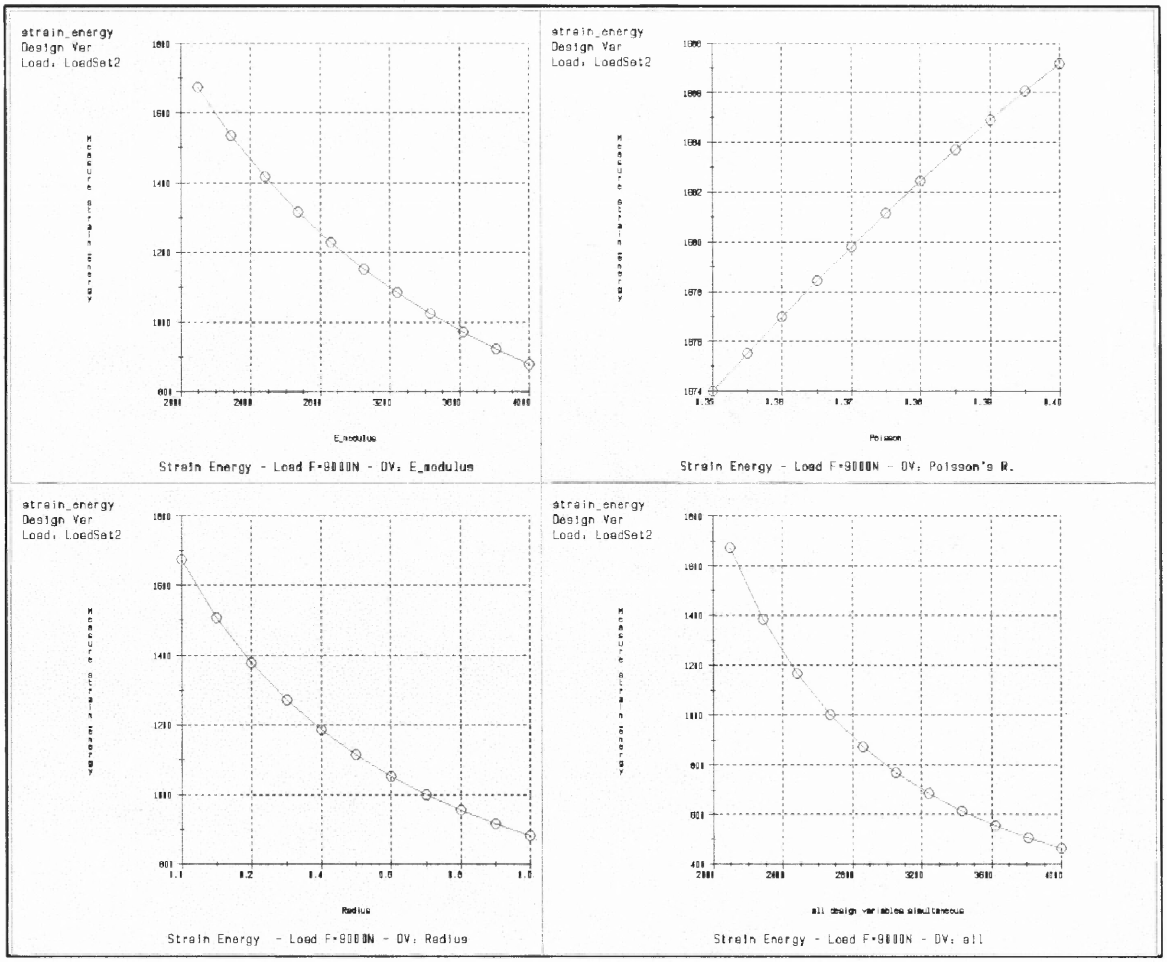


Figure F.32 Strain Energy Depending on Design Variable Modulus of Elasticity, Poisson's Ratio, Radius, and All Design Variables Simultaneous

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