NATIONAL COOPERATIVE
HIGHWAY RESEARCH PROGRAM REPORT 248

ELASTOMERIC BEARINGS DESIGN, CONSTRUCTION, AND MATERIALS

TRANSPORTATION RESEARCH BOARD
NATIONAL RESEARCH COUNCIL

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AREAS OF INTEREST:

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TRANSPORTATION RESEARCH BOARD
NATIONAL RESEARCH COUNCIL
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NATIONAL COOPERATIVE HIGHWAY RESEARCH PROGRAM

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FOREWORD

By Staff
Transportation
Research Board

This report is recommended to bridge engineers, construction engineers, materials engineers, researchers, specification writing bodies, and others concerned with elastomeric bridge bearings. It contains the findings of a comprehensive assessment of the performance of elastomeric bearings based on a review of current domestic and foreign codes of practice, research findings, and performance data. Recommended specifications for elastomeric bridge bearings, accompanied by a commentary, are included in this report.

Because of the desirable performance characteristics, maintenance-free durability, and first-cost economy of elastomeric bearings, there has been a burgeoning of applications and a proliferation of sizes for bridge bearings. From the initially small, unreinforced elastomeric bearing pads for short-span prestressed beams, applications for laminated elastomeric bearings have grown—especially abroad—to such an extent that today designers are considering bearing capacities of 1,500 tons. The current AASHTO specification for elastomeric bearings has, in large measure, been a catalyst for the accelerated growth of elastomeric bearing applications in the United States; yet it no longer reflects the best information available. The full potential of elastomeric bridge bearings will not be realized until these specifications are modified and expanded.

The present AASHTO specifications are limited in a number of ways. For example: axial load capacity for steel-reinforced bearings is below that allowed by other codes; axial loads are not related to rotation or translation, separately or in combination; shear forces that are generated by temperature-induced translation are undefined; laminate reinforcement is not related to load levels; and the relationship between compression stress and compressive strain is not specified.

Innumerable elastomeric bearing research projects have been completed during the last two decades both in the United States and abroad. The findings of much of this research have been presented in detail in published papers receiving broad distribution. But much other work, especially that by commercial concerns and independent research organizations, has received only limited circulation or none at all. Some performance data are available on actual bearings in the United States and abroad. Because most design engineers have not had the time or opportunity to assimilate this information, it is not presently reflected in the AASHTO specifications and has not had much effect on the design of elastomeric bearings.

NCHRP Project 10-20 was initiated at the University of Washington in 1981 with the objective of developing design, construction, and materials specifications for unconfined, plain and reinforced, elastomeric, bridge bearings, and this report contains the findings of that study. The first six chapters comprise a logical development of the information available on elastomeric bridge bearings. The major conclusions are summarized in Chapter 7, and the proposed specifications (Method A) are presented in Appendix D.

Method A, along with its accompanying commentary, deserves special attention in that it is recommended for immediate adoption as part of the *AASHTO Standard Specifications for Highway Bridges*.

A second phase of research with the objective of developing further improvements in the specifications for elastomeric bearings is expected to begin early in 1983. Additional information might make it possible for the adoption of a more sophisticated approach, similar to Method B in Appendix E, for use when additional design effort is justified. For the present, however, only Method A is recommended based on the best information available.

CONTENTS

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The research reported herein was performed under NCHRP Project 10-20 by the Department of Civil Engineering, University of Washington. The work was supervised and performed by the co-principal investigators, Charles W. Roeder, Associate Professor of Civil Engineering, and John F. Stanton, Assistant Professor of Civil Engineering. Two former graduate students, Mr. Soterious Demetriou and Mr. Kenneth Gottleaber made extensive contributions.

The authors are particularly grateful for the advice and assistance of their Advisory Panel; Mr. Charles Dolan, ABAM Ltd., Consulting Engineers, Ontario, Canada; Professor James Kelly, University of California, Berkeley; Dr. Peter Lindley, Malaysian Rubber Producers Research Association, England; Mr. Leland Mayo, and Mr. Anthony Schloss, du Pont Company, Stow, Ohio; Mr. Geoff Stranaghan, Private Consultant, Melbourne, Australia; Mr. Umesh Vasishth, Washington State DOT; and the late Mr. Lawrence Bell, du Pont Company. This panel provided valuable advice and guidance throughout the study, and they were particularly helpful in reviewing reports and draft specifications before they were submitted to NCHRP.

ELASTOMERIC BEARINGS DESIGN, CONSTRUCTION, AND MATERIALS

SUMMARY

The objectives of the study were to obtain the most up-to-date information on elastomeric bearings, to evaluate the data, and to propose modifications for the AASHTO Specifications. The major findings were:

1. *General*—Elastomeric bearings are effective and economical. Those problems that do occur are generally attributable to poor materials or workmanship or to misuse of the design specification. Current stress limits are too severe on some classes of bearings and too liberal on others. Raising allowable stress levels should be done with caution because, although the high stress levels allowed by foreign codes are attractive, they are generally dependent on higher shape factors and more stringent quality control than are customary in the United States.

2. *Materials*—Suitable elastomeric compounds can be made from Natural Rubber (polyisoprene) or Neoprene (polychloroprene). Other materials may also be suitable, but lack of a universal material performance specification and scanty field experience inhibit their use. Elastomers stiffen at low temperatures; therefore, a compound must be chosen that is suitable for the environmental conditions at the bridge site. This is particularly important with polychloroprene rubbers. Conflicting opinions exist over which material is best for particular applications, but the question is probably not of major importance because both Natural Rubber and Neoprene appear adequate for most applications.

3. *Mechanics*—Small-deflection theories have been extensively developed, but are generally too cumbersome for use in design. Simplifications based on the shape factor are less accurate, but appear adequate for present-day design. A serious need exists for a reliable, usable, large deflection theory that includes consideration of combined loadings.

4. *Experimental Work*—Many experimental programs have been carried out, but their usefulness as a whole is limited by the diversity of materials used, the conditions of testing, and, in a number of cases, failure to control important variables and to record all the necessary details. Thus correlation between different test series is not in general good, and verification of theories is difficult. The experimental corroboration that does exist, generally does so only over a rather limited range of parameters. There is a serious lack of information on fatigue loading, rotation, and combined stresses.

5. *Design*—The present AASHTO Specifications do not have a particularly rational basis, make no mention of some important issues, and, in particular, do not distinguish between plain pads and reinforced bearings. As a result, bearings in use in the United States today, while satisfying the AASHTO Specifications, have widely varying margins of safety and reliability. Thus, a limited number of cases have occurred of bearings not performing satisfactorily in the field, and some of them have been serious enough to cause litigation. Such cases have shown that the cost of replacement far exceeds that of the original bearing. Therefore, it makes sense economically to use only high quality reliable bearings.

Two proposals for change to the Specifications are presented in this report. They are based on an improved understanding of bearing behavior and are intended to lead to bearings that have an adequate and more uniform level of

reliability without incurring excessive cost. Method A is safe, more rational than the present specifications, and can be incorporated now. The major changes lie in the separate treatment of plain pads and reinforced bearings, revised provisions for allowable compressive stress, deflection, and anchorage, and selection of material, and the introduction of provisions for rotation and strength of the reinforcement. Method B is complex and will permit more sophisticated designs at the expense of increased design effort. It is presented only in skeletal form because some of the details in its provisions can be established only after more research has been done in certain specific areas—notably fatigue, stability, large deformation theories and combined stress. Tighter material quality control standards will also be needed if the full potential of elastomeric bearings for high load applications is to be met.

CHAPTER ONE

INTRODUCTION AND RESEARCH APPROACH

GENERAL

Elastomeric bearings are widely used as supports for bridge girders of many types and as seating pads in precast concrete construction. Today they are also finding increasing use as acoustic or seismic isolation devices for whole structures. Additionally, they have provided excellent service in mechanical applications, such as machine vibration mountings, shaft bearings in helicopters and wind turbines, and automotive suspension systems and engine mounts. Their main advantages are that they have no moving parts to corrode or seize, they are inexpensive to manufacture, and they require no maintenance.

Typical bridge bearings are either plain pads (Fig. 1) or bearings that are reinforced with horizontal laminates of steel or fabric (Fig. 2). The reinforcing laminates are bonded to the rubber and restrict the bulging of the bearing. Thus, reinforcement increases the compressive and rotational stiffness and controls the vertical deflection and rotation of the bearing. Horizontal movement is permitted by shearing deformation of the rubber, and, for a given total rubber thickness, it is unaffected by the presence of reinforcement. When very large horizontal movements must be accommodated, a polytetrafluoroethylene (PTFE) slider is added (Fig. 3). A stainless steel plate slips with very little friction on the sheet of PTFE that supports it, and that is secured to the bearing below it.

Pot bearings (Fig. 4) are used when large vertical loads are accompanied by large horizontal movement but little rotation. A piston rides on the elastomer, which is fully contained within a shallow cylinder and behaves like an incompressible fluid. Horizontal movements are taken on a PTFE slider, while the pot bearing allows rotation in any direction.

This study concentrates on plain pads and reinforced bearings. Throughout the world the most common elastomers used in bearings are compounds of Natural Rubber (NR) or polychloroprene (CR). Reinforced bearings are usually made by hot molding, with steel plates completely encased in the elastomer. Some states in the United States have successfully used mats of fabric (e.g., fiberglass) in place of steel, permitting large sheets of bearing material to be made and then cut to size without the risk of corrosion at the edge of the reinforcement. In Europe, similar economies have been sought with steel reinforced bearings by making sheets of steel–rubber–steel "sandwiches," cutting to size, and cementing the pieces on top of each other to form multilayer bearings, the exposed surfaces of which are then painted with a protective coating.

A wide variety of uses, configurations, materials, and design methods are thus associated with elastomeric bearings. In the United States, they have been used as bridge bearings since the late 1950's and were first recognized by the American Association of State Highway and Transportation Officials (AASHTO) in 1961 (*1, 2*). The original specifications were written in response to the developmental work of the du Pont Company (*3*) and others, and were intended for plain

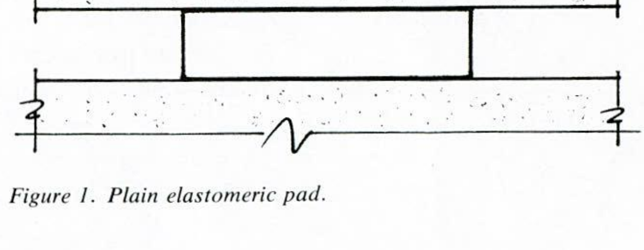


Figure 1. Plain elastomeric pad.

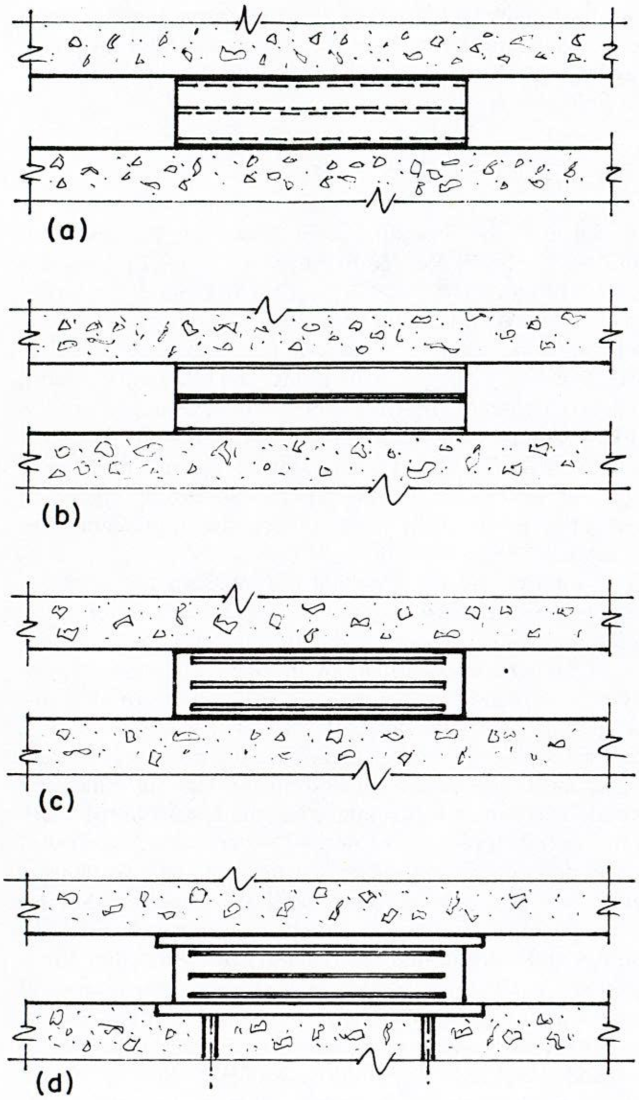


Figure 2. *Reinforced elastomeric bearings: (a) glass fiber reinforced bearing, (b) multiple rubber-steel sandwich, (c) hot-molded steel-reinforced bearing, and (d) reinforced bearing with external plates (dowels optional).*

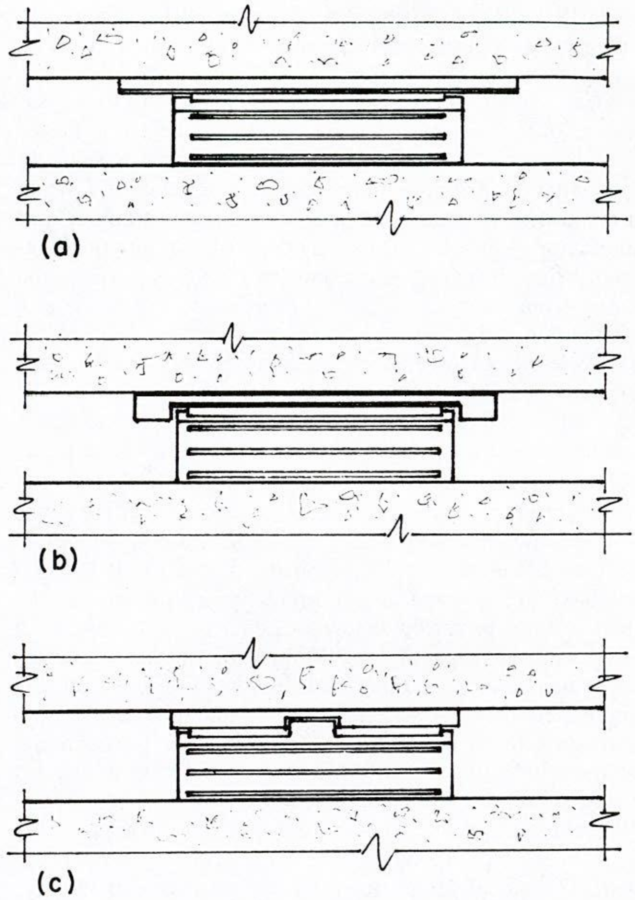


Figure 3. *Elastomeric bearings with PTFE sliders: (a) unguided sliding bearing, (b) externally guided sliding bearing, and (c) internally guided sliding bearing.*

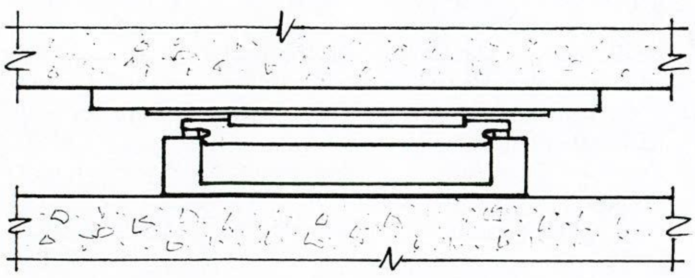


Figure 4. *Pot bearing with PTFE slider.*

CR pads. However, the specifications have been used for both plain pads and reinforced bearings made from CR and NR. In 1961, elastomeric bearings were not widely used, but their application has increased dramatically in recent years because of their economy. Increased usage has resulted in a wide spectrum of bearing applications that were not envisioned in the 1961 AASHTO Specifications. For example, the only major changes in the AASHTO elastomeric bearing design provisions since 1961 are the elimination of the 80-ft span limitation and the reduction of maximum permissible compressive strains from 15 percent to 7 percent.

Lack of communication is the most probable cause for this lag in the development of the AASHTO Specifications. It is understandable because the implementation of elastomeric bearings requires the skills of several parties with quite different backgrounds and interest, such as the bridge engineer, rubber producer, bearing manufacturer, and contractor. The differences can best be seen by considering the steps involved from initial conception to final installation of such a bearing. The raw polymer (NR or CR) is produced and sold by a small number of organizations that focus on the chemical properties, behavior at the molecular level, and related problems such as resistance to aging. They have only a secondary interest in the mechanical behavior of the rubber and the response of the bearing to load, because the majority of their products are used in nonstructural applications. The bearing manufacturer purchases the raw rubber from the producer, compounds it, assembles the bearing to the specifications of the designer, and vulcanizes it. The properties of the rubber and the behavior of the bearing are strongly dependent on the compounding and vulcanization, but, while the manufacturer knows much about rubber technology and manufacturing process, he usually knows less about the function of bridge bearings. The contractor's job is to order the bearing from the manufacturer according to the designer's specification and to install it in the bridge. He generally has no knowledge of elastomeric bearings, and is unable to judge whether the design that he passes to the manufacturer will prove satisfactory or whether the bearing delivered to site is what he asked for.

The interests and skills of those who contribute to the finished product are clearly diverse, and so the bearing needs to be specified in a way that is clear and unambiguous to all concerned. Yet, the bridge engineer, whose job it is to do this, has interests that are different again from any of the others—lying primarily in the mechanical performance of the bearing. He typically has little interest or knowledge of rubber technology, and he speaks a very different language from the rubber producer and bearing manufacturer. These contrasting needs and interests, combined with the complex, nonlinear behavior of elastomers have created problems for designers and the AASHTO Specifications.

It is thus reasonable to enquire if these problems result in any failures in the field, because if they do not, the existing AASHTO Specifications must in some sense be satisfactory, and the real question of interest is how much further their provisions can be extended without significant risk of trouble. The answer is that bearings do fail. However, accurate information is extremely difficult to obtain both because engineers understandably do not like to advertise such events, and because failure of an elastomeric bearing is hard to define, being less sudden and catastrophic than, say, the collapse of a beam. Despite this, the authors know of several cases in the United States and abroad in which problems have arisen that are severe enough to cause distress in the structure and litigation over it. In addition, inspection of a random selection of bearings in Western Washington (where the research was performed) suggests that there are a significant number of bearings in place that are not performing the way their designers intended them to.

Most of these cases concern plain pads that have deformed excessively, split, or slipped out of place.

Rubber is seen by most people as a tough, resilient material that can take phenomenal abuse, and many engineers view elastomeric bearings in the same light. Further, the fundamental mechanics that describe its behavior are different from those of conventional structural materials and are consequently less well understood by many engineers. This combination of unusual structural behavior, lack of dramatic failure, unfamiliar mechanics, and a simplistic design specification have caused some engineers to treat elastomeric bearings in a rather "cavalier" fashion, viewing overstress of a bearing less seriously than overstress of a steel beam. The authors believe that this attitude has also pervaded field inspections, which in some cases have become rather cursory operations, and which, were they performed more stringently, would reveal many more difficulties than are presently known. This opinion is backed up by the investigators' own field observations.

SCOPE AND OBJECTIVES

Because of the foregoing difficulties, the National Cooperative Highway Research Program (NCHRP) funded a research program (NCHRP Project 10-20) to study the problem and propose a new specification. The objectives of this study were to gather knowledge in the form of research reports, test data, design specifications, and other information; to analyze it; and to present a state-of-the-art summary of the work. Recommendation for further study and a proposal for a new AASHTO Elastomeric Bearing Specification were also to be developed. A final report was to be prepared to present the results of the work and propose modifications to the AASHTO Specifications.

The authors and the University of Washington were selected to perform this work. The authors have gathered many research articles, reports and publications, which are recorded alphabetically in an unabridged bibliography, included in Appendix C. They have read and interpreted this information, and they have discussed the problems with many researchers—rubber technologists, bearing manufacturers, and bridge engineers both in the United States and abroad. This report is a summary of the results of the work on this research project. Chapter Two contains a review of present practice in elastomeric bearings. Chapter Three is an elementary description of those material properties that are most relevant to the designer and manufacturer of bridge bearings and a discussion of the influence of the compounding process. Chapters Four and Five contain a review of theoretical developments and experimental results. Chapter Six contains a summary of the major foreign design specifications and compares them with the AASHTO Specifications, and the final chapter contains conclusions, proposals for modification of the AASHTO Specifications, and recommendations for further work.

RESEARCH APPROACH

As noted earlier, the purpose of this research was to investigate thoroughly the state of the art of elastomeric bearings and to propose a new AASHTO specification.

Thus, it was necessary to gather and evaluate all available information before attempting to write a specification. The diversity of both the subject matter and the sources of information called for special efforts to ensure that the research was conducted in a thorough, rational manner, and two particular steps were taken towards this goal. First, an advisory panel of specialists was selected by the researchers to assist in the investigation. The subject is sufficiently complex that no one individual has a detailed grasp of all aspects of it; therefore, the members of the advisory panel were chosen because of their detailed knowledge in particular areas, and they included producers, rubber chemists, researchers, state bridge engineers, and consultants. They gave initial advice on where to find information, they provided solutions to particular problems on a continuing basis throughout the project, and, in particular, they made a critical review of draft specifications and the interim and final reports prior to submission to NCHRP.

The second step was to divide the subject into broad areas so that the presentation and evaluation of the information could be as orderly as possible. The chapters of this report reflect this division.

The planned activities could be broken down into the four main areas of acquisition of information, evaluation of data, formulation of a specification, and writing of reports. The gathering of data was carried out on three fronts. First a telephone survey of bridge engineers and others was conducted within the United States to try to establish the state of current practice and to find out the main problems in the field from those actually working in it. Then an extensive search of published literature was conducted, providing information on materials, fundamental theories of mechanics, laboratory tests, field behavior, other codes of practice, etc. Lastly, the wealth of unpublished knowledge was tapped by visits to manufacturers, researchers, and engineers both in the United States and abroad. Many more contacts were established by mail or by telephone.

The information was frequently conflicting, so the evaluation phase of the project was vital. In some cases, what purported to be fact turned out to be opinion, derived from a misunderstanding of the work of others. Elsewhere, the results of one test appeared to contradict those of another, even allowing for the difficulties involved in achieving precision when testing rubber.

The formulation of the specification was based on a synthesis of all the information gathered and evaluated in the prior phases, but particular attention was paid to the existing AASHTO Specifications and to foreign codes of practice. Because the mechanics of elastomeric bearings are complicated and because failure criteria for rubber are much less clearly defined than those for conventional structural materials, the stress limits proposed by different authorities vary widely; and so considerable judgment was required in this phase of the project. As a result the proposed specification does not follow closely any one other, but rather attempts to draw together the best of the existing specifications and research results in a document that is rational, self-consistent, and in accordance with the most up-to-date knowledge.

CHAPTER TWO

REVIEW OF PRESENT PRACTICE

GENERAL

Because of the communication gaps noted in Chapter One of this report, a survey was made of bearing manufacturers, bridge engineers, researchers, and elastomer producers in order to identify the problems in present practice. Most contacts were in the United States, but discussions were also held with a few key authorities abroad. The objective of the survey was to gather information relevant to bearing design but not available in the literature. The communications were made by phone, letter, or personal contact, and they were directed toward evaluating field experience, existing AASHTO code provisions, code provisions of other countries, difficulties encountered in the design, manufacture and installation of elastomeric bearings, and other similar concerns. Often, a wide variety of opinions were found on the same subject. Because of time limitations, not all the possible sources were contacted, so this cannot be regarded a true statistical survey, but rather an expression of concerns and observations relevant to elastomeric bearing design and use. This chapter summarizes these comments and conversations.

FIELD EXPERIENCE IN THE UNITED STATES

A series of general and technical questions was directed toward the bridge engineers of each state or the person nominated as most knowledgeable in the area of elastomeric bearings. The responses revealed that elastomeric pads and bearings are widely used in the United States, and their use is increasing rapidly. Only one of the respondents indicated that they were not used in his state, while nearly 50 percent of the respondents indicated that they had been using them for less than approximately 5 years. Nearly all respondents indicated that they used elastomeric pads or bearings on virtually all short-span, reinforced or prestressed concrete bridge girders. These were most often plain pads 1 in. thick or less. Occasionally, bearings with cotton fabric reinforcement were reported, and glass fiber or steel plate reinforcement was used for thicker pads (usually ¾ in. or thicker). Elastomeric bearings were used less frequently for steel or long-span concrete bridges, and approximately 35 percent of the respondents indicated that their state had virtually no experience with large bearings for such structures. These states used mechanical bearings because they did not fully understand the behavior of larger elastomeric bearings and they were not confident of their ability to design and select them. Generally, such states had been using elastomers for fewer years (5 years or less). They typically believed that elastomeric bearings were economical and maintenance free, and they expected their states' usage to increase significantly in the coming years. However, they were still restricting use at this time.

Other states reported much wider use of elastomeric bearings. Approximately 25 percent indicated that they used them on all but the most unusual bridges. These states typically had a longer history of experience with elastomers (12 to 15 years or more) and employed better defined design procedures, most of which were originally developed with the help of individual manufacturers or fabricators. They noted a few problems in their early experience, mostly attributable to poor quality materials, fabrication or installation, but said that over the years these problems had been resolved and now elastomeric bearings were providing relatively trouble free service for a wide range of applications. The experienced users had confidence in their grasp of the design and behavior of the bearings, and they were willing to extend the usage to new applications and bigger sizes. For example, they have pursued options such as PTFE coatings to permit slippage, glass fiber for reinforcement, and the use of bearings with large areas and load capacities (i.e., surface areas of approximately 500 in.^{2} or more, 400 kips load capacity or larger). The remaining states generally fell somewhere between these two extremes.

The respondents were almost unanimous in their praise of elastomeric bearings, noting particularly that their initial price and maintenance costs were lower than those for mechanical bearings. They indicated that bearings designed under the present AASHTO (*1*) procedure have very seldom shown distress. Only a very few elastomeric bearings have required replacement, and most of these were replaced for indirect reasons such as repair of bridge piers or excess shortening of prestressed concrete girders because of concrete creep. Respondents who had worked with elastomers for a number of years frequently indicated that several problems occurred during their early experience, but that these early problems were now resolved. Separation of the rubber from the reinforcement was a common problem during these early years, but it was largely resolved through improved manufacturing procedures, and the almost universal use of bearings that are hot bonded during vulcanization.

Although there was agreement that elastomeric bearings function well, the discussions clearly illustrate several major problem areas in their use. One relates to the wide range of applications and sophistication in the use of elastomeric bearings. Because some states have the confidence of many years of experience, while others do not, the needs of the designer vary greatly from state to state. Ideally the AASHTO Specifications should satisfy these needs, but the present edition appears to fall short. This difficulty could possibly be solved by a multilevel design approach. A very simple, conservative method could be used by the designer less knowledgeable in rubber, and a more sophisticated technique could be employed by the engineer who is familiar with elastomers and wants to stretch the design to a more realistic limit. The 800-psi compressive stress limit also appears to be a major concern of the bridge engineers. They generally believe it is too conservative and results in excessively large bearings. These have a larger horizontal stiffness than necessary, cause large forces in the substructure, and increase the total cost of the bridge.

Concerns over quality assurance were also expressed. In U.S. practice many bearings are shipped without any sort of test, and the only checks that can reasonably be made on site are on the external dimensions. These give scant assurance that the rubber has the properties specified or that the reinforcement has been accurately located. Early failures because of poor quality materials served to exacerbate such concerns.

Installation problems also caused concern. For example, bearings have occasionally been installed the wrong way up (with steel plates vertical). Further, excessive rotation caused by differences in slope between the pier and girder due to camber or deflection of the girder or construction tolerances was often quoted as worrisome. These rotations cause vertical loads to act eccentrically, risk overstressing the rubber in the bearing, and may cause slippage of unreinforced pads. Several details were suggested for minimizing these problems, as shown in Figure 5, but many of the suggestions employed details whose behavior was not well understood, such as tapered elastomer layers. Some engineers noted that in extreme cases the bearing may slip out of position because of the eccentricity and dynamic loading of the bridge. They were unsure about the true level of friction between the bearing and the bridge piers and girders, and they were interested in finding attachment details for various degrees of end restraint. Several engineers had reasonable suggestions for attaching laminated bearings, but few for unreinforced pads.

Several respondents reported that they frequently used bearings of odd shapes or with holes in them. For example, it is sometimes necessary to cut corners from bearings used on skewed crossings because of space limitations, and holes are frequently made through the bearing to accommodate dowels that restrict translation. Several engineers questioned the wisdom of using these details because they did not un-

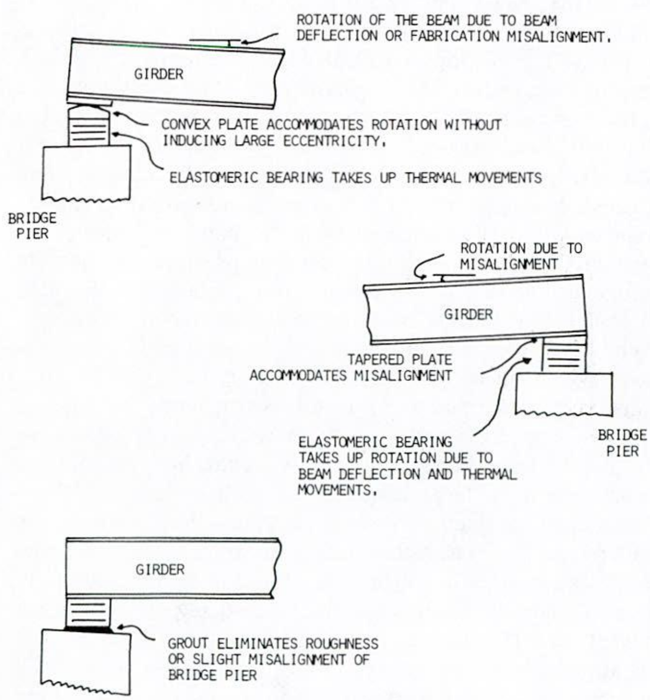


Figure 5. Typical details to prevent misalignment damage to the bearing.

derstand how the changed shape influenced the stress and strain of the bearing. Other questions were raised about the validity of the design aids, material specifications, and manufacturing tolerances employed by individual states. In summary, most bridge engineers were generally very happy with the performance of elastomeric bearings as presently used in their state, but some engineers had doubts about individual aspects of their design procedures, or the confidence that could be placed in the existing methods for use with bigger bearings.

Although bridge engineers were nearly unanimous in their praise of elastomeric bearings, there were a very few indications that bearings were under distress in present practice. Details of these distressed bearings were sketchy because of a natural hesitancy to discuss matters that are pending litigation. However, it is sufficient to note that there a few concrete examples of damaged bearings due to overstress, excessive deformation, or other cause, and a careful evaluation of present design procedures is warranted.

COMMENTS OF BEARING MANUFACTURERS

Several rubber bearing manufacturers were also contacted to determine their thoughts concerning present U.S. practice and design procedures. The subject was discussed with several major firms as well as smaller organizations. All of the firms interviewed had principals with extensive experience in the elastomeric bearing industry. Universal agreement was noticeable by its absence; but a number of their remarks are worthy of discussion.

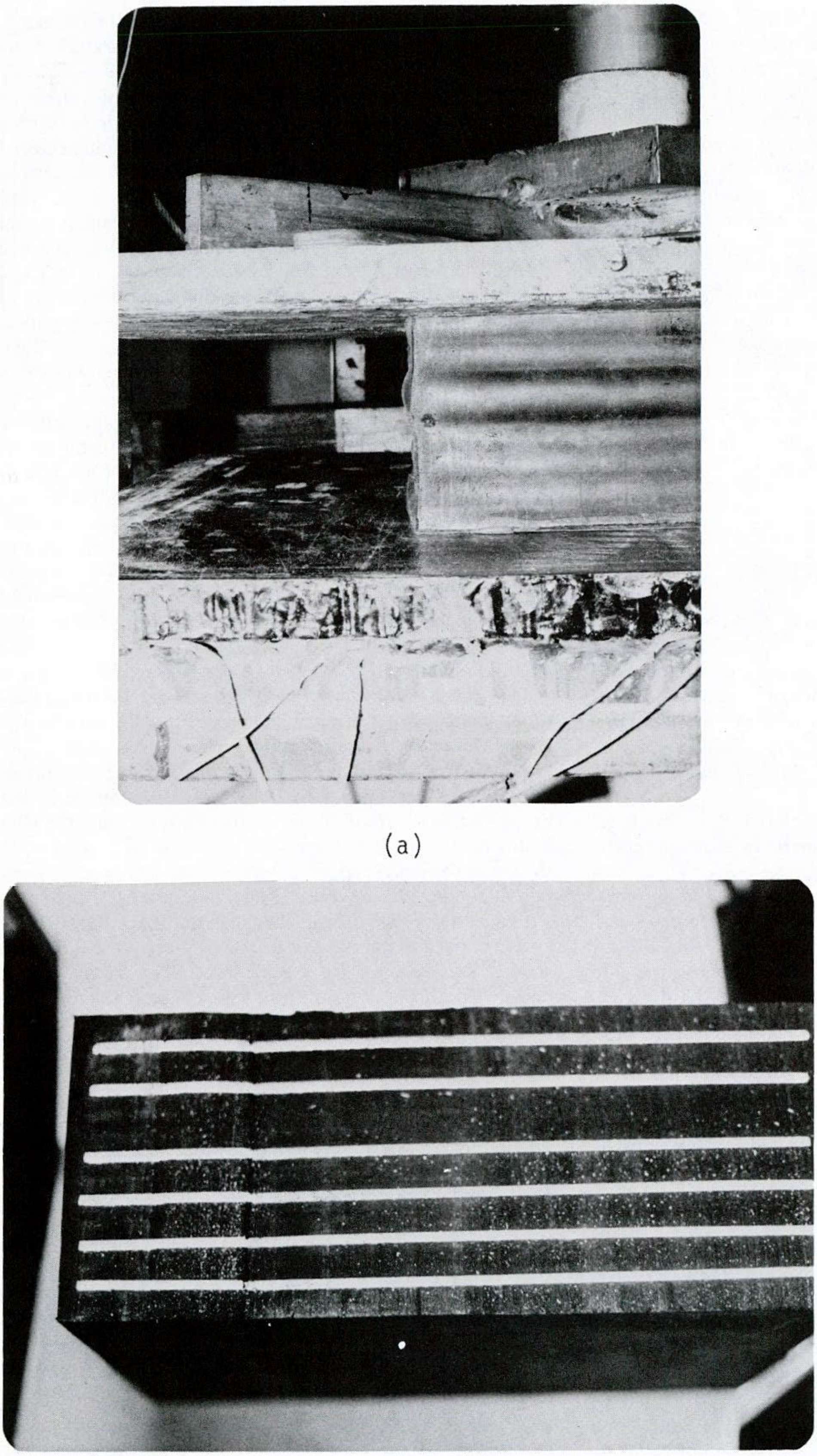
The bearing manufacturers all agreed that there has been a rapid increase in the use of elastomeric bearings in recent years, with virtually all being made from NR or CR. They indicated that these bearings were almost always in the hardness range of 50 to 60, with plain pads and reinforced bearings both enjoying increased use. They noted that rubber with a hardness up to about 70 was occasionally used for plain pads, but was usually specified because the designer did not understand the material. Most manufacturers believed that the communication gap between the bridge engineer and the bearing manufacturer caused them considerable difficulty. They reported that designers who do not understand the complex behavior of rubber bearings sometimes produced poor designs. These mistakes were difficult to correct because the bearing manufacturer usually deals directly with the bridge contractor rather than the design engineer. A few manufacturers would prefer to design their own bearings because of that difficulty, but others wanted no responsibility for the design process. For example, they noted that many state design specifications (and AASHTO) require A36 steel for the laminating plates, but 14 or 16 gage laminates are specified in the bearing. A36 steel is usually not available in these thicknesses except on large special orders, but many other equivalent mild steels are readily available. This communication lapse frequently results in expensive and time consuming change orders or the use of more expensive steels.

The manufacturers generally believed that their major contribution to the finished bearing was the correct compounding of rubber, the bonding of the rubber to its laminate, and the vulcanization of the final product—operations demanding considerable skill. They indicated that separation of the laminate from the rubber is the mode of failure most often caused by poor work on their part. The manufacturers all agreed that quality control in general and cleanliness during bonding in particular are essential to the production of a good bearing. Several felt that cleaning of the steel reinforcement by sand blasting, careful and clean application of the adhesive, and well-controlled application of heat and pressure during vulcanization were equally important. They all noted that the rubber bridge bearing industry had a steady influx of new manufacturing firms, and they indicated that some of these firms did not maintain high standards in these critical areas. Further, they pointed out that the design specifications and acceptance criteria of most states would not catch the resulting defects.

In view of the foregoing difficulties, the manufacturers questioned the wisdom of relying on strictly competitive bids for the purchase of bearings. They believe that track record of the firm and manufacturing processes used should be considered in the bidding process. One firm indicated that mandatory proof loading of each bearing to a compressive load of twice the service load would eliminate many defective bearings. Irregularities in the bulging pattern (Fig. 6(a)) would show up many problems, such as inadequate bond or missing laminates (Fig. 6(b)). They noted that this testing procedure might eliminate 1 to 2 percent of the bearings that are still acceptable, but felt it was worth the price. Two manufacturers indicated that they use the process on all bearings, but others were strongly opposed to such a requirement.

A number of different fabrication methods were reported, several of which appear to be worthy of mention. It was noted that there are different methods for locating the laminates in reinforced bearings and each has its problems. In the United States, keeper blocks are frequently used to maintain minimum edge cover as shown in Figure 7(a). The laminating plates are slightly smaller than the space between the keepers, but the edge cover at the resulting groove is then smaller than required by specifications. The manufacturers believe that this cover at the groove is adequate to protect the bearing without filling it in, but they felt that the present specifications are not clear on the issue. It was also noted that dowels were sometimes used to ensure plate alignment, as shown in Figure 7(b). The dowels require holes in the steel plate in the region of high tensile stress, which would indicate that a thicker plate should be used; however, because this is not specified in present codes, the manufacturers generally used standard thickness plates, despite the holes. Vertical alignment of the laminates is normally assured by controlling the thickness of the elastomer. In extreme cases, fine horizontal wires are inserted in the bearings to control spacing. One manufacturer indicated that AASHTO tolerances ((*2*) para. 3.5) are generally reasonable for bearings of small or moderate shape factor. However, bearings with large shape factor and thin elastomer layers were sometimes difficult to construct because of the relative tolerance measure. It was suggested that an absolute tolerance value be applied in this case.

One respondent was concerned about the installation of his firm's bearings. He pointed out that the bearing was usually installed by a contractor who has little knowledge of rubber. The installation and connection details were believed to be



(a)

(b)

Figure 6. Photograph of the benefit of proof loading (photos by Tobi Engineering).

important, because a poor detail could render the bearing ineffective even though the bearing was of good quality. He noted that improper welding on or around the bearing could break the bond between rubber and reinforcement and cause premature problems.

PRACTICE IN OTHER COUNTRIES

The discussions with engineers in the United States suggested that there were considerable variations in practice between the United States and other countries. A thorough review of the codes and specifications indicates that wide variations exist, which are discussed in detail in Chapter 6. However, a feeling for the difference can be obtained from Table 1, which contains a summary of the predicted capacities of 13 typical reinforced bearings made of rubber of approximately 55 hardness and 575 percent elongation at break. It shows the maximum permissible compressive loads with no shear or rotation and with maximum permissible shear and rotation. The calculations are made using AASHTO Specifications and present U.S. practice, the British Specification BE 1/76 (*4*), and the International Railway Specification UIC 772R (*5*). These 3 specifications were chosen because they represent very different design philosophies and cover the wide range of variation.

The comparison suggests that the AASHTO specifications are very conservative compared to other codes for reinforced bearings with large shape factor and little or no applied shear or rotation. They are less conservative for bearings of moderate shape factor, and may be unconservative for very low shape factor bearings made of rubber with an elongation at break as low as the minimum permitted by AASHTO. UIC 772R permits the largest permissible loads for bearings of large shape factor with no shear or rotational deformation. BE 1/76 typically produces the largest permissible compressive load for bearings with large shape factor when shear and rotational deformation are present.

A more complex discussion of these differences and the reasons for them is presented in Chapter Six. However, because of the observed differences, comments from researchers, practicing engineers, and bearing manufacturers in Europe and Great Britain were solicited. The discussions indicate that rubber bearings are performing well in other countries. It appears that BE 1/76 has been used more widely, but both it and UIC 772R have been used with few major problems. However, modest amounts of rollover at the corners, edge delamination in bearings that are not hot molded, and surface cracking were reported. A number of bearings that are loaded with average compressive stress in the range of 2,000 psi were noted, and isolated examples of bearings loaded in the range of 4,000 psi were found. It should be noted that these highly stressed bearings were intended to be loaded primarily in compression with minimal shear or rotation. The practicing engineers generally believed their specifications were conservative, and few serious problems were noted. However, engineers are seldom willing to discuss their failures, and there were hints of problems at very high stress levels. Further, some engineers noted that they can use these larger allowable stresses only because they have much tighter control over the material and manufacturing process than presently employed in the United States.

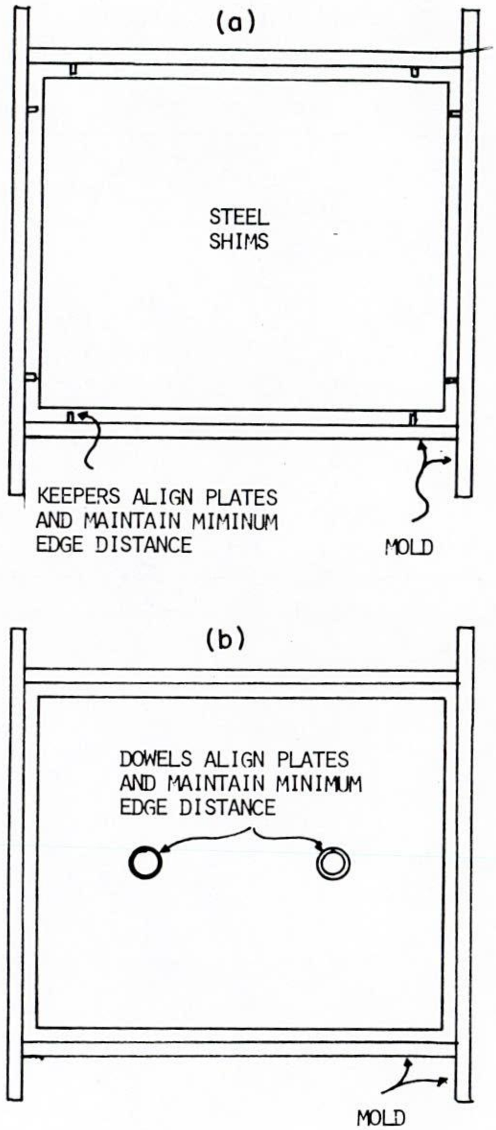


Figure 7. Two typical methods for aligning shims.

The actual design of bearings varies from country to country. For example, some countries use thin laminating plates, while others employ relatively thick ones. There appears to be little difference in the observed performance or behavior, except that bearings with very thin plates sometimes tend to curl excessively at the corners because of large shear deformations (this phenomenon will be discussed later). Local cases of separation of the laminate from the rubber or bending of the thin plate were reported. Although CR and NR are both used, some countries show a preference for one material. Germany has a strong preference for CR, while Great Britain and Australia prefer NR. Germany requires an arduous testing procedure before any change in material or manufacturing process is made. Construction of the bearings may also vary. For example, bearings are frequently manufactured in large sheets and cut to size in France. This elim-

Table 1. Sample design capacities in kips.

L(in)	W(in)	t_i (in)	Number of Layers	S_i	P_{all} AASHTO	P_{all} BE 1/76 $\Delta_s = 0$	P_{all} BE 1/76 $\Delta_s = 0.5T$	P_{all} UIC 772R $\Delta_s = 0$	P_{all} UIC 772R $\Delta_s = 0.7T$
22.85	22.85	.59	10	9.67	418	950	603	682	468
16.94	22.85	.47	3	10.29	316	749	522	599	473
12.21	22.85	.35	7	11.22	223	587	383	473	335
16.94	22.85	.71	3	6.86	310	504	344	425	322
16.94	22.85	.71	9	6.86	*	504	297	403	245
10.24	13.00	.47	4	6.06	106	153	101	123	88
6.23	13.32	.24	6	8.97	66	140	90	123	83
10.24	13.00	.59	6	9.85	*	124	74	101	63
6.23	13.32	.35	2	5.98	66	95	63	96	71
5.44	8.59	.24	2	7.04	37	63	43	53	39
7.41	8.59	.35	6	5.61	*	66	43	55	35
5.44	8.59	.24	8	7.04	*	63	36	52	30
5.44	8.59	.35	6	4.69	*	43	25	34	21

UIC 772R LOAD CAPACITIES COMPUTED BY ASSUMING MAX ROTATION

* EXCEEDS AASHTO STABILITY LIMIT

inates the edge cover and raises questions of corrosion, bond breakdown, and edge delamination that are less frequently encountered in other countries. However, this sandwich bearing is very economical.

Some of the differences in design relate to manufacturing tolerances, deflection, strain limitations, and other serviceability conditions. They are influenced by the type of bridge construction practices and connection details, which vary greatly between the United States and other countries. For example, the expansion joints that are used in the deck vary widely between countries and appear to have considerable influence on some of the limits. Further, steel bridges frequently require of the bearing larger rotations and greater shear movement than do prestressed girders of similar span. Long-span bridges place different demands on the bearing than shorter spans, and again the use of these different configurations varies from country to country because of labor costs and local engineering practice.

The manufacturing process is similar in the United States and abroad, and the problems and concerns are much the same. However, there are a few differences. In European practice the bearing manufacturer is frequently closely associated with the bridge contractor. This presumably permits better communication and better quality control during the design, construction, and installation of the bearing. The firms are generally larger and very experienced, and so there are fewer firms with limited or questionable capabilities. There appears to be little or no use of fiberglass or fabric reinforcement. One of the major firms in Great Britain and all firms in Germany make each bearing with a unique serial number identification that can be used to identify the bearing design and its properties. Thus, the European bearing manufacturer appears to assume a higher level of responsibility, and exerts greater control over the total process.

Foreign experience and specifications thus merit close scrutiny. Most foreign specifications are generally less conservative than AASHTO's, yet experience has shown few major problems in their use, and even they are believed by most engineers to be conservative. However, their direct application in the United States is likely to be inappropriate because of the many differences in engineering practice and contractual procedures.

CHAPTER THREE

MATERIALS

GENERAL

The performance of the elastomer has a significant impact on the behavior of a bearing. The bridge engineer is concerned primarily with the behavior of the finished bearing. He wants a bearing that can accommodate the necessary movements and support the required loads without developing problems within the bearing or the structure. This normally means that the bearing should be stiff and strong with respect to vertical loads and flexible with respect to shear deformation and rotation. These properties are partly controlled by geometry, but the mechanical properties of the rubber are also important. In addition, the designer wants to avoid premature failure due to fatigue, deterioration due to environmental conditions, or serviceability problems due to excess deflection or creep.

The producer of the elastomer and the bearing manufacturer have expertise and interests in rubber very different from those of bridge engineers. They recognize that elastomers are complex polymer compounds that can have wide variations in behavior depending on chemical composition and manufacturing process. Therefore, they are primarily concerned with the specification of the elastomer, the recipe used in the compounding, and the vulcanization process. The properties that interest them most are rubber hardness, elongation at break, ozone resistance, Mooney scorch time, etc. They are skilled in manipulating the rubber to obtain desired material properties such as those classified in ASTM Standard D2000-80 (*6*), and they regard this skill as proprietary information not to be shared with others. However, although they are clearly able to manufacture a bearing that satisfies the needs of the bridge engineer, their success in doing so is sometimes hampered because of failure to understand each other's needs and because the two speak different technical languages.

Because of these differences in perspective, the specification of the elastomer in a bridge bearing remains something of a black art. It is likely that many polymers could be used in elastomeric bearings, but only CR and NR are widely used, with Butyl finding limited use in very cold climates. A limited number of other materials, such as EPDM, Chlorobutyl and Hypalon (*7, 8, 9*), have been investigated for use in elastomeric bearings. However, these other materials have experienced only limited use because both CR and NR are economical and have a long history of good performance, and engineers are understandably reluctant to risk using new materials when there is little to gain and a lot to lose. In view of this pattern of use, this chapter will present a brief review of the material properties of elastomers. It will focus primarily on CR and NR with limited reference to Butyl, but the concerns raised in this chapter could also apply to many other polymers.

MECHANICAL PROPERTIES OF RUBBER

Rubber exhibits highly nonlinear, viscoelastic thixotropic constitutive properties (*10*). That is, its stress–strain relationship is nonlinear and time dependent, and it may not retrace the same paths when unloaded and reloaded. In addition it is temperature dependent. If rubber is tested under pure uniaxial stress with no transverse restraint, it shows a stress–strain relationship such as that shown in Figure 8 (*10*) where the usual engineering definitions for stress and strain are employed. That is,

$$\sigma = \frac{P}{A_o} \tag{1}$$

and

$$\epsilon = \frac{\Delta}{\ell_o} \tag{2}$$

where σ and ϵ are stress and strain, Δ is the elongation, P is the tensile force, and A_0 and ℓ_0 are the cross-sectional area and length of the undeformed specimen. Rubber can attain tensile strains of several hundred percent when these definitions are employed. The slope of the curve and the stress at rupture may vary considerably depending on the strain rate, temperature, and rubber compound.

This nonlinear material behavior is frequently explained by the long chain molecule model (*10*). Rubber-like materials have a long chain molecular structure. The chains are initially crooked and randomly oriented, but cross links between chains are introduced during the vulcanization process. At very small strains, rubber is essentially linear elastic. An apparent softening occurs at moderate strains. However, as larger longitudinal strains are applied, some cross-links break and the molecular chains tend to straighten and orient themselves in the direction of the loading. Thus, a distinct stiffening effect may be noted at large strains as shown in Figure 8. NR and CR are both strain crystallizing materials. That is, their molecular structure (*10*) changes with time at high strains, so the material becomes stronger and tougher. When an initially cracked piece of such rubber is stretched, crack growth at the tip is inhibited by the local strain crystallization, and the rubber consequently displays good fatigue resistance. It is often suggested that this is a major reason why NR and CR are more suitable for bridge bearings than other polymers that do not strain crystallize. It should be noted that many engineers associate crystallization with embrittlement. However, this is not the case in rubber for which descriptors such as tough and leathery would be more appropriate.

Cross-links in the rubber restrict the straightening of the initially crooked molecular chains, and further, some longitudinal chains will be more highly stressed than others. Thus, some chains and cross-links may break before others. These links are thermally active and depend on temperature, and new links may form with time. Thus, the elastomer may have different strength and stiffness characteristics at lower or elevated temperatures.

Further nonlinearities may be introduced by geometric changes that take place with large strains. The stress and strain of Figure 8 are related to the undeformed geometry, and give the same values as do more sophisticated, finite strain, definitions provided ϵ is less than approximately 0.1 and the shear angle is less than approximately 10 deg. Elastomers can sustain strains many times larger than these values, and geometric nonlinearities may necessitate a more sophisticated finite stress and strain definition (*10, 11*). However, despite the numerous sources of nonlinearity in rubber, nearly all analyses of elastomeric bearings have been performed assuming infinitesimal strain, isotropic, linear elastic material behavior. There are several reasons for this. First, the mathematical solution of mechanics problems with finite strain models and nonlinear constitutive relations are very difficult even for simple problems and the results generally cannot be expressed in forms readily usable by designers and practitioners. Further, the viscoelastic properties of the rubber introduce additional computational difficulties. In practice the viscous properties provide useful damping for many seismic and vibration isolation applications, but they are not so well defined as to justify their inclusion in the analysis of bridge bearings. Thus linear elastic analysis, modified by appropriate empirical factors, appears to provide acceptable accuracy for most bridge bearing applications.

Homogeneous, isotropic, linear elastic material behavior is defined by the elastic modulus, E; shear modulus, G; bulk modulus, K; and Poisson's ratio, ν. Only two of these moduli are independent, and the other two are related by the equations:

$$G = \frac{E}{2(1+\nu)} \tag{3}$$

$$K = \frac{E}{3(1-2\nu)} \tag{4}$$

Actual values of these moduli are difficult to determine from an individual rubber sample. However, it is well known that the bulk modulus of rubber is very large compared to E and G, and it is frequently assumed that rubber maintains constant volume during all deformation (i.e., K is infinitely large, ν is 0.5, and $G = E/3$). This assumption has been used in many analytical studies of elastomeric bearings, but recent studies (*12*) indicate that ν should lie in the vicinity of 0.4985 to 0.4999. At first glance, this slight variation in ν would appear to be trivial, but other recent studies (*13*) suggest that even slight changes in ν can have a significant impact on the stress distribution when ν is in the neighborhood of 0.5, since very small changes in ν in this range represent large changes in bulk modulus. Because of this sensitivity, the bulk modulus is probably a more useful constant than ν.

A number of difficulties are associated with the definition, measurement, and use of these elastic constants. Firstly, the equations of linear elasticity (including Eqs. 3 and 4) are only valid for infinitesimal strains, and will have inconsistencies if they are used with large strains. So, for example, if an E modulus is obtained from a standard test specimen by measuring force and deformation at large strains, and if a ν value is also obtained from measurement of lateral contraction, a value for G can be calculated using Eq. 3.

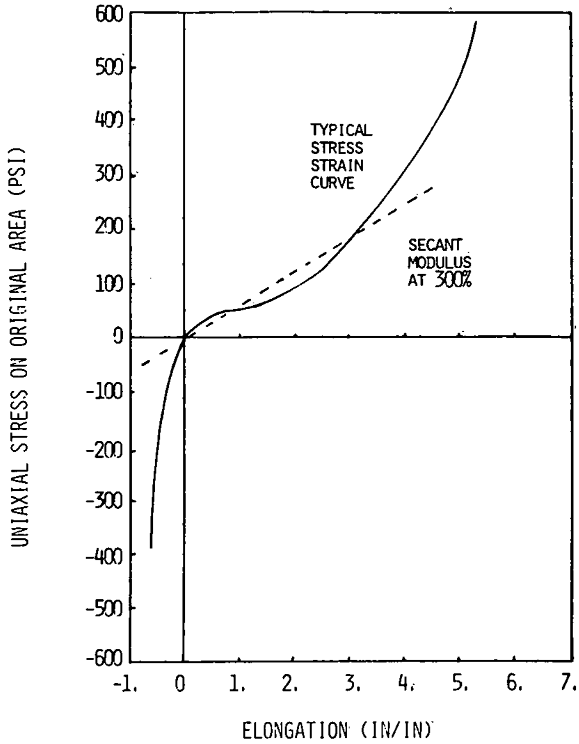


Figure 8. Typical uniaxial stress–strain curve for rubber.

However, if other tests at large strains, such as shear or torsion, are used to find the shear modulus, they will give different values for G. Another manifestation of the same problem is that E cannot exceed $3G$ (i.e., ν cannot exceed 0.5) for infinitesimal strains. If it did, it would describe a material that swells under external pressure and violates the laws of thermodynamics by requiring creation of energy in a closed system. However, if values are obtained from large deformation uniaxial and shear tests, the same limitation will no longer apply, and Ref. 4 recommends values for E that are 4 to 5 times G.

Other difficulties are caused by the details of the test arrangement. The shear stiffness of a material specimen or a full-sized bearing is really a function both of geometry and material shear modulus. While the simple assumption is usually made that the shear stress is constant across the specimen, this cannot in fact be true because equilibrium dictates that the shear is zero at the corners, so the shear distribution must be nonuniform. Thus the state of strain even in a standard material shear test (*4, 5, 15, 16, 17*) is not one of simple shear, and the shear stiffness of the specimen will differ somewhat from the simple value:

$$\text{Shear stiffness} = \frac{\text{Shear modulus} \times \text{area}}{\text{Thickness}}$$

Measurements on full size bearings are further complicated by the compression which has to be applied in order to provide enough friction to hold the bearings (Fig. 9). The compression causes three geometric effects that influence the shear stiffness. The bulging of the rubber increases the area available to resist shear (*18*), the reduced bearing thickness alters the value of shear strain for a given shear deformation, and, particularly for thick specimens, the effects of instability (see Ch. 4, section under "Stability") might become significant. The possibility of a pure material effect has also been hypothesized whereby the pure material modulus changes under compression, in a manner similar to granular soils. However, there is no hard evidence for such an effect and so it is currently presumed not to exist.

Obtaining true material properties is thus seen to be difficult, although a recent study (*14*) has shown that moduli measured at finite strains can be corrected to give reasonable estimates of the true elastic moduli. In view of the difficulties, confirmatory tests on finished bearings are clearly desirable.

It is well known that rubber hardness is related to the elastic modulus, and a number of studies have investigated this relationship. Hardness is measured by a durometer, and it is related to the depth of elastic indentation under a given load. Several different hardness scales exist, and they each use an indentor of different geometry. Gent (*19*) investigated this phenomenon and derived an expression for E as a function of International Rubber Hardness (I.R.H.) degrees by formulating the problem in terms of small deformation theory of elasticity. Experiments on specimens of NR were performed to verify this relationship, and the results are shown in Figure 10. Other hardness measures were also investigated. Only an approximate relationship could be obtained for the Shore A hardness test, but this measure is nearly identical to I.R.H. over the range of hardness of interest (i.e., approximately 50 to 70).

Other factors (*14, 12, 20*) in the compounding and vulcanization process may affect the material properties of rubber, and these factors will be discussed in greater detail later in this chapter. However, hardness is the most common measure used to describe the material properties in the design of bearings because it is readily measured. There may be considerable scatter with this method, but it is generally agreed that rubbers which are used in bridge bearings produce a shear modulus of approximately 90 to 120 psi at 50 hardness and 180 to 240 psi at a hardness of 70, when loaded quasistatically. The elastic modulus will be approximately 3 times the value of G if the true elastic properties of the material are measured. The bulk modulus is much larger than E or G with values of approximately 300,000 psi typically proposed, and it varies with different compounds. Accurate measurement of bulk modulus is very difficult, calling for much more rigorous experimental techniques than are needed to establish other properties.

MODES OF FAILURE

Failure of the rubber is an important aspect to be considered in the design of elastomeric bearings. It is generally recognized that elastomers fail with the development of tensile stresses in the body. A rubber specimen tested under uniaxial tensile stress will fail only after sustaining large elongations. The extent of this elongation at rupture depends on the compounding and stiffness of the rubber. NR compounds usually produce larger elongations than CR of the same hardness; however, minimum values of 300 percent to 400 percent are common for both materials, and elongations approaching 600 percent are not unusual. Bridge bearings are

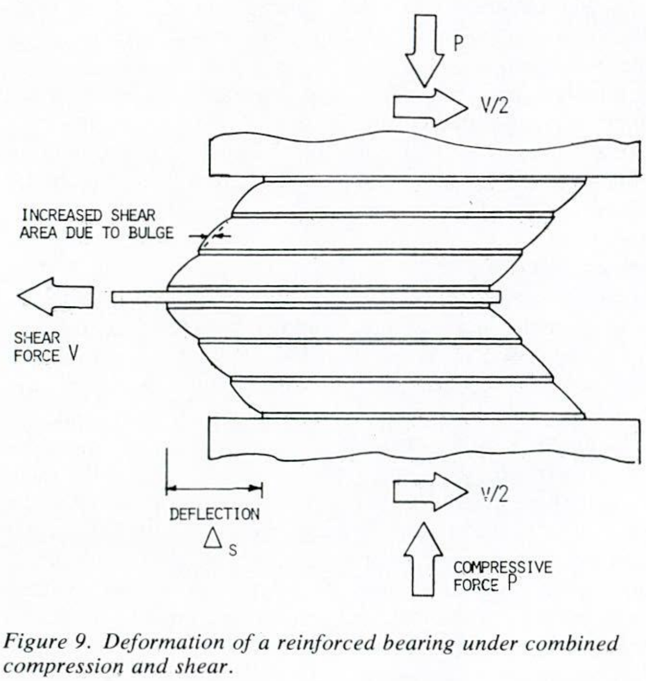


Figure 9. Deformation of a reinforced bearing under combined compression and shear.

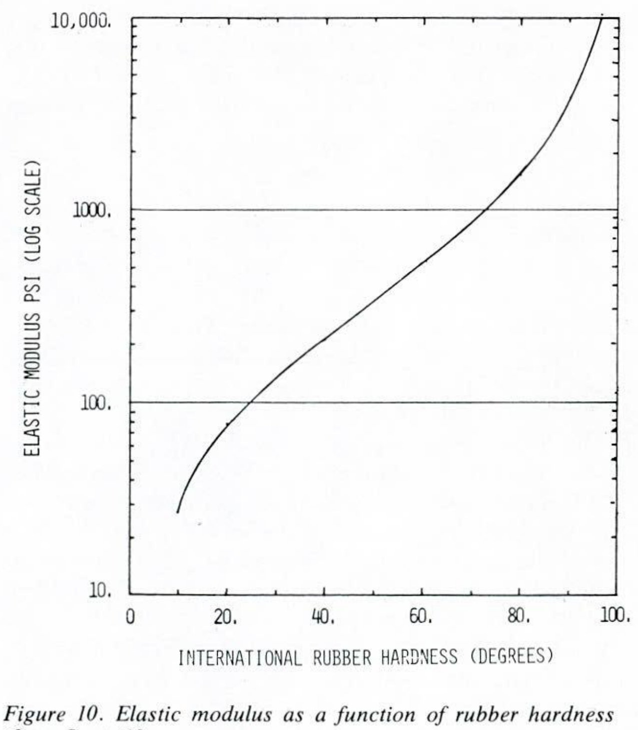


Figure 10. Elastic modulus as a function of rubber hardness (from Gent (19).

not loaded in uniaxial tension, so the results of the elongation test are not directly applicable; but elongation at break is almost universally specified as a quality control test for the material, because the test is easily performed and it permits easy identification of many inappropriate elastomer compounds.

Distress due to internal rupture may occur at stress levels well below the failure stress for uniaxial tension (*21*). This mode of failure is known to be essentially independnet of the tensile strength and elongation at break of rubber. It occurs because of hydrostatic tensile stresses that may develop in the bearing. The mode of failure has been verified experimentally, but it should be noted that internal rupture does not necessarily cause a failure of the bearing. Rupture consists of a break within the rubber body, and it is very possible for the bearing to sustain much larger applied loads before complete failure occurs. However, the rupture introduces the potential for a shortened service life and is generally avoided. Tensile hydrostatic stress can occur in rubber bearings only under specific conditions. Laminated bearings with high shape factors will develop large hydrostatic tensile stress with relatively small tensile strains (*13*), as indicated in Figure 11. Bearings that are subjected to severe rotations may also develop these hydrostatic stresses. Tensile loads on bearings are inadvisable, and rotations are usually restricted to help control the development of tensile stress in these areas.

Failure may also occur within a bearing because of initiation and growth of a crack or flaw in the rubber. Griffith and Irwin first investigated the crack growth theory, and it has received wide acceptance in studying the fatigue behavior and brittle fracture of metals. Rivlin and Thomas extended the theory to apply to rubber. Other recent studies (*22, 23, 24*) have considered the phenomenon as a potential failure mechanism for elastomeric bearings. Calculations of required energy levels have been made and have been compared with experimental measurements. These comparisons are frequently based on the results of tear tests, which are closely associated with crack growth. The results of that work indicate that crack growth is a viable method for predicting the fatigue life, tearing and tensile failure of rubber.

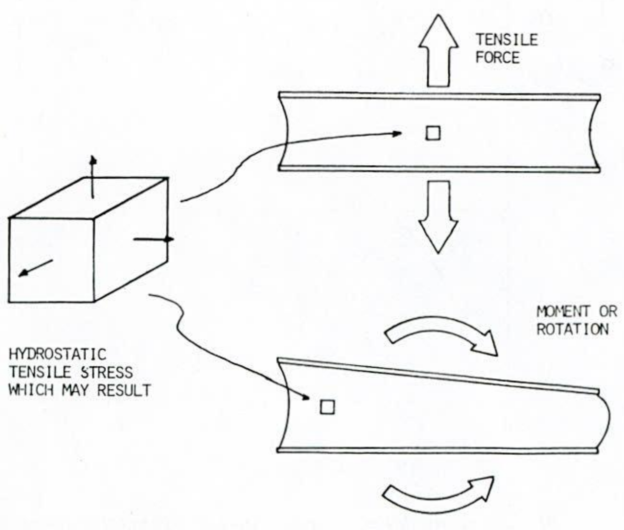


Figure 11. Hydrostatic tensile stresses in bearings.

However, this must also be regarded as an area where many questions remain to be answered. Crack growth theory was developed by Irwin on the assumption of a linear elastic isotropic continuum. Elastomers are typically long chain polymers, which may be influenced greatly by temperature, and strain rate. They have a very different molecular structure than most metals, and some authorities (*24*) question to what extent crack growth theory applies to rubber. The theory has considerable potential as a method for studying the failure of rubber, but it cannot be regarded as the complete answer at this time.

Fatigue is the last and possibly most critical failure mode of the elastomer. Many studies have investigated the fatigue problem, including tests on small specimens, tests on actual bearings, and crack propagation studies. One of the earliest studies (*25*) of fatigue in rubber consisted of repeated compression or shear loading on bonded rubber blocks. Fatigue failures were noted even when the bearing was loaded in cyclic compression, and it was found that significant increases in fatigue life were noted for strain crystallizing rubbers if complete relaxation was not permitted after each load cycle and if strain cycles did not pass through zero strain. That is, if a minimum strain is maintained on the bearing at all times, severe strain cycles can be applied with relatively long fatigue life. The study clearly shows a strong correlation between applied strain and fatigue life, but the reasoning has been extrapolated by some authorities (*4*) to suggest that fatigue life is related to elongation at break. The maximum shear strain allowed by the code (*4*) is then limited to a percentage of elongation at break.

Other studies (*26, 27, 28*) have investigated crack propagation theory as it relates to fatigue life. These studies indicate that fatigue of bearings under compressive load is initiated at locations of tensile stress concentrations at edges of the bond of rubber to laminated plates. These cracks may be initiated through mechanical means or ozone cracking (to which unprotected NR is particularly sensitive). A number of fabrication and bearing shape details (*28, 29, 30*) have been proposed to limit these stress concentrations and cracking. Tests have also been performed to substantiate many of these conclusions.

Finally, a number of full size bearings have been tested under repeated loadings (*9, 31, 32, 33, 34*). It is widely recognized that the fatigue life of sull-scale structural components (e.g., joints) may be very different from the fatigue life of small material specimens because of their more complex state of stress which often contains concentrations not present in the material specimens. Therefore, fatigue testing of larger components generally enjoys greater credibility among practicing engineers, and thus these tests should clarify the question of fatigue life. However, it is questionable whether this high credibility exists for full-size elastomeric bearing tests because the results are frequently confusing and contradictory. Some of the tests have been performed under very restricted stress and strain levels, and it is difficult to extend these results to present or future practice. Others have been performed on bearings that are of unrealistic design or the tests are poorly documented. For example, a number of tests have been performed on bearings with a hardness greater than 70, and several major testing programs have produced results so scattered that only the most general conclusions can be drawn. Further, there are several apparent contradictions in the results. A general examination of the results appears to indicate that both NR and CR produce similar fatigue lives under comparable loadings if the elastomer hardness and elongation at break are within reasonable limits. Thus, very hard rubbers (approximately 70 or greater) may produce widely varying results, and material with very low elongation at break (approximately 300 percent or less) may produce significant shortening of fatigue life. This is clearly an area where further experimental work is needed to resolve differences and to fill the many gaps in existing literature.

DETERIORATION AND SERVICE LIFE OF THE ELASTOMER

Bridge bearings are generally designed to serve for the required life of the bridge. Replacement of the bearing is sometimes possible but not desirable. Despite constant exposure to extreme climate conditions and occasional chemicals, such as deicing salts, elastomeric bridge bearings have served for many years with no significant deterioration. Bridge engineers almost universally agree that elastomeric bearings offer a longer service life with less maintenance than other types of bearings. However, the influence of the environment and aging remains a question of some concern, and several material tests (*1*) are frequently required in design specifications to ensure good resistance to deterioration.

Ozone cracking is one concern that is raised with respect to rubber materials. While it occurs in CR, the extent is small and it generally is not considered a problem. NR is more susceptible to it, but it can be readily controlled or eliminated by the addition of an antiozonant during the compounding process. However, an excess of such additives may endanger other physical properties, such as the bond. The cracks form because of a slight chemical change in the rubber at the edge of the bearing where the rubber is exposed to the environment. Cracks only form when the rubber exposed to ozone is in tension, and even then they form slowly. Measurements (*35, 36, 37*) have been made of the rate of crack growth and energy level required for propagation of the crack. The studies indicate that ozone cracking is a self-limiting phenomenon in most cases, because the cracked layer provides a stable layer that resists deeper penetration of the cracks.

Because the ozone cracks occur in a region in which existing cracks may propagate under repeated load, several studies (*26, 27, 28*) have considered the mechanical growth of these ozone cracks into a fatigue failure. The rates of crack growth and energy levels required for growth were computed and measured for both ozone cracking and mechanical cracking. It was found that the normal mechanical fatigue crack growth mechanism rapidly surpasses ozone cracking in virtually all practical situations. The energy requirements for ozone crack growth are low, but the growth rate is very slow. Energy requirements for mechanical crack growth are much higher, but the growth rate is several orders of magnitude greater than for ozone cracking. Therefore, fatigue induced by ozone cracking is possible only after a long period of low stress, cyclic loading. In view of the existing evidence, ozone cracking of the side cover must be regarded as an aesthetic problem rather than a serious structural problem. However, the bond may be more sensitive to ozone attack than the rubber (*38*). This is a major reason for requiring side cover on laminated bearings.

Aging of the bearing is also suggested as a possible deterioration problem for elastomeric bearings. It is believed that the properties of the bearing may change with time because of chemical changes induced by temperature and general weathering. These changes manifest themselves through increased hardness of the rubber, reduced elongation at break, and increased shear modulus. Accelerated aging tests (*39*) have been performed to measure these phenomena and are sometimes required in design specifications. Questions have been raised as to whether accelerated testing is a valid measure of true aging because no clear interaction has been defined between time and temperature. It is regarded by some people as simply a test for presence of the antioxidant. Only a very few elastomeric bearings have been in service for more than 15 to 20 years, and few of the early bearings were tested to determine their true mechanical properties. A research program is in progress at the Malaysian Rubber Producers Research Association and the Transport and Road Research Laboratory in England to study aging. Bearings with known properties at installation are being removed from service and tested to determine their present properties. This study should provide valuable insight into aging and deterioration of actual bearings.

Resistance to oil and to chemical attack has also been proposed as a possible consideration in the design of rubber bearings. CR appears to enjoy a consistent advantage over NR in these areas because it has a higher resistance to oils, some acids, and chemicals. NR may break down if exposed to certain chemicals (i.e. sulfuric acid, some oils, etc.) for long periods of time. However, few elastomeric bearings are subjected to such conditions for a significant period of time, and typically only the surface of the elastomer is exposed to the harsh environment. Therefore, it is questionable that this deterioration is of significant concern for most bridge bearings. It should also be noted that automobile engine mounts are usually made of NR, and they have served for many years under continuous exposure to oils without significant damage.

TIME AND TEMPERATURE EFFECTS

The mechanical properties of elastomers are time and temperature dependent. Loading rate (or strain rate) has a significant influence on the elastic modulus of rubber. Dynamic tests produce a higher G and E than quasi-static tests, and some design specifications (*5, 40*) require that the shear modulus be doubled when computing stress and strain due to dynamic loads. This dynamic effect introduces additional complications when measuring the elastic properties of rubber, and it complicates the interpretation of test data such as the results of fatigue tests.

Loads of very long duration may introduce time dependent effects (*41, 42, 43, 44*) through creep and stress relaxation. Creep is an increase in deformation due to a constant load, and relaxation consists of a reduction in stress due to constant deformation. Both phenomena are considered closely related. They are typically expected to increase as a logarithmic function of time, but the maximum magnitude and rate of creep or relaxation are sensitive to temperature, stress history, and the elastomer formulation. Stress relaxation under shear deformation is generally not viewed as a serious problem because it reduces the forces on bridge piers and other subassemblages. Creep or relaxation under compressive loads is usually regarded as an undesirable effect in bridge bearings, and creep tests (*9, 45*) have been performed on full-size bearings. These test results suggest that maximum creep values are in the order of 20 percent to 40 percent of the instantaneous deformation for elastomers typically used in bridge bearings. However, it should be noted that creep measurements may be complicated by the rate of load application and variation in the definition of creep deflection. If the load is applied slowly, some creep deformation may be included in the instantaneous deflection measurements. Most of the creep has taken place by the time the final finish surfaces are added to the bridge (*9*), and the small added compressive deformation seldom constitutes a serviceability problem. Thus, within the above limits, creep does not appear to be a serious problem for bridge bearings made from conventional materials. It may be a serious problem with new materials and new compounds; therefore creep tests are needed before considering new rubber compounds.

Low temperatures (*9, 33, 46, 47*) induce crystallization and thermal stiffening. Both effects cause a significant increase in stiffness (i.e., an increase in E and G) of the elastomer, an increase in hardness, and a reduction in the elongation at break; and they are sometimes (*47*) associated with first and second order transition temperatures. NR and Butyl appear to have an advantage over CR in this area. The rate of crystallization depends on temperature and is most rapid at approximately 10 F for CR. This is sometimes referred to as the first order transition temperature. This rate dependent phenomenon does not appear to be a serious problem for NR. However, all elastomers exhibit thermal stiffening at lower temperatures (second order or "glass transition") and this stiffening is believed to be an instantaneous effect. CR stiffens at higher temperatures (*9, 33, 47*) (typically −25 F as opposed to −40 F), and its stiffness increases more than NR's. This stiffening increases the stresses in the bearings, and therefore it increases the forces on bridge piers and other components. Thus, it is regarded as an undesirable effect, and NR or Butyl is frequently required for bridges in cold climates. It should be noted that full-size bridge bearings have significant mass, and it takes a sustained period of time for the low temperature to fully penetrate the bearing (or leave the bearing with increased atmospheric temperatures); and this complicates the assessment of low temperature effects. However, it has been suggested (*89*) that the low temperature problem is overstated because the increased stiffness applies to incremental forces and deflections. Further, there is relaxation of shear stresses under applied shear deformation even at low temperatures. This question is further complicated because the bridge girders and bearings may change temperature at different rates due to the mass and conductance of the bearing as noted above. Thus, consideration of low temperature effects requires careful thermal analysis. Further, the present AASHTO (*1*) low temperature test may be unrealistic because very few locations in the continental United States attain the low temperatures required by the test, while Alaska and a few other locations may attain much lower temperatures. A more realistic requirement may be to require a low temperature test that is consistent with the local environment or a range of standard tests (*17*) from which the appropriate one may be chosen.

EFFECT OF VULCANIZATION AND COMPOUNDING

A rubber mix consists of a raw elastomer, fillers, additives to aid the manufacturing process, a protective system, and a vulcanization system. Vulcanization consists of a cross-linking agent such as sulfur, peroxide or urethane, and the application of heat and pressure for a period of time. Sulfur is the agent typically used in elastomeric bearings, and it forms many cross-links between the long polymer chains by chemical reaction during the heating. The duration and temperature of the heating is also very important because it must be sufficient to penetrate the entire depth of the bearing, but the bearing must not be overheated because this will cause partial reversal of the vulcanization process.

Carbon Black is the filler typically used in elastomeric bearings. It is added to modify the hardness of the rubber and adjust the stiffness. However, it also affects the tensile strength, elongation at break, creep and stress relaxation, and fatigue behavior. Different types of carbon black have different particle sizes and internal structure, and so they all have different impacts on the final properties. Processing oils are also added to improve the processing of the rubber and sometimes to reduce the shear modulus, but they may also increase the creep or stress relaxation. Excessive oil may reduce the quality of the bond in the bearing. Antioxidants and antiozonants are also frequently added to inhibit ozone cracking and deterioration noted earlier.

BONDING OF ELASTOMERS

The bonding of the elastomer is also an important aspect of bearing design. A secure bond between the elastomer and the reinforcing laminate must be obtained if the bearing is to perform properly. This usually requires the application of an adhesive to the clean, sandblasted steel shims as well as the layer of elastomer. In some cases, the adhesive is a 2 part agent where one part is applied to the laminate and the other to the rubber layer. Bond is then achieved when the layers are brought in contact and cured under heat and pressure during the vulcanization process. Bonding has been done cold after the rubber has been vulcanized, but these bonds have generally been less reliable and are not encouraged for use in elastomeric bearings. Epoxy resins, polyurethane, and epoxy-polysulfide systems have been used as bonding agents. A number of patented bonding systems are available on the market. The brass plating method has also been used for bonding rubber to metal. It requires that the metal be coated with a layer of brass before bonding, but it does not appear to be widely used today.

Rubber to rubber band is sometimes required. Manufacturers who cannot make thick layers accurately build them up from a number of thinner ones. Fabric bearings are often built up by this method with the fabric embedded in the appropriate places. The layers are usually joined without adhesives by the heat and pressure of the vulcanization process. If a thick fabric bearing is required, it is sometimes constructed by cold-bonding several pieces of standard thickness bearings placed on top of each other.

Adhesives are sometimes used to attach elastomeric bearings to their seatings and prevent slip. This is an economical method; however, discussions with practitioners suggest that it is relatively unreliable and it should be discouraged.

CONCLUDING REMARKS

This discussion of material properties has been oriented toward the needs of the bridge engineer, and the focus has been on CR and NR because these materials make up the vast majority of elastomeric bearings. Both materials appear to make very good bearings if they are compounded to have a hardness in the vicinity of 50 to 60, low creep and stress relaxation properties, and long fatigue life. CR would appear to be more desirable than NR for bridges exposed to extreme chemical environments because of its better resistance to ozone, oils, acids, and chemical attack. NR would appear advantageous for bridges exposed to very low temperatures (i.e., minimum temperatures well below 0 F) because of its more desirable low temperature and crystallization behavior. Clearly, many other polymers and compounds may be suitable for bearings. These rubbers have not been discussed, but their acceptance criteria must be the same as that applied to Neoprene and Natural Rubber and discussed in this chapter. However, it should be noted that the best available evidence suggests that new polymers must be strain crystallizing materials because of their longer fatigue life.

CHAPTER FOUR

THEORETICAL MECHANICS

GENERAL BEHAVIOR

The principles of mechanics are used to provide theoretical relations between the forces and deformations of a body. They combine the boundary conditions with the constitutive equations and the equilibrium and compatibility conditions to obtain a solution. The mechanical behavior of rubber is very complex as noted in Chapter Three, but even if it is treated as linearly elastic, its behavior is unlike that of conventional construction materials because it is very flexible in shear, yet virtually incompressible. These properties require the use of special analytical techniques, which are needed for the analysis of incompressible material.

A plain rubber pad compressed between two perfectly lubricated surfaces will deform as shown in Figures 12(a) and 12(b). Considerable lateral expansion of the rubber occurs, but it is uniform with depth. If the contact surfaces are rough or the rubber is bonded to them, the bulged shape shown in Figure 12(c) will result. Because the bulging is the source of compressive deformation, the addition of internal laminates to restrain it, as shown in Figure 12(d), will significantly increase the compressive stiffness. However, the addition of internal reinforcing layers has no appreciable impact on the shear stiffness if the total rubber thickness is unchanged. Thus, the horizontal and vertical stiffness may be controlled independently within wide limits by a suitable choice of reinforcement.

The shape factor of a rubber layer is a non-dimensional parameter which gives a good indication of the compressive stiffness of the layer. It is defined by

$$S = \frac{\text{Area of one loaded surface}}{\text{Area free to bulge}}$$

$$= \frac{ab}{(a+b)\,t} \text{ for a rectangular bearing of sides } 2a \text{ and } 2b$$

$$= \frac{D}{4t} \text{ for a circular bearing}$$

Notation for dimensions is shown in Figures 13(a) and 13(b). High shape factors produce bearings that are stiffer in compression, and most reinforced bridge bearings fall in the range 4 to 12.

Vertical compressive stresses exist within a reinforced bearing, and theory (*48, 49*) and experiments (*50, 51*) suggest that the distribution is approximately parabolic as shown in Figure 14. The bulged shape of the rubber indicates that shear stresses and strains exist throughout the material, and their distribution appears to be approximately linear across the width and through the depth. They cause tension in the reinforcement, with a maximum value near the middle.

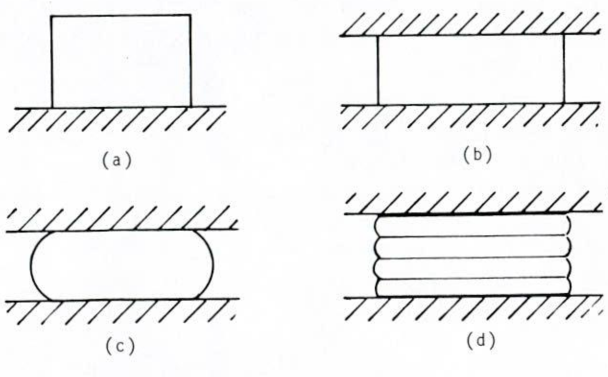


Figure 12. Compressive behavior of bearings.

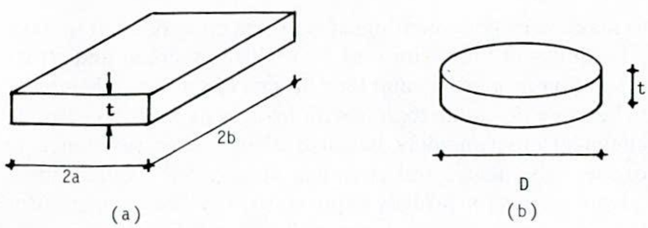


Figure 13. Notation for dimensions.

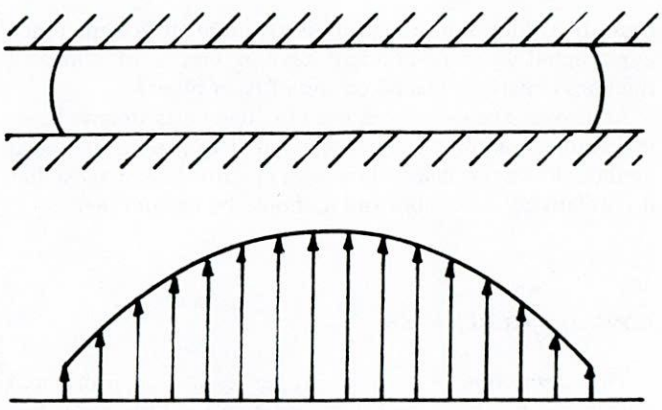


Figure 14. Compressive stress distribution.

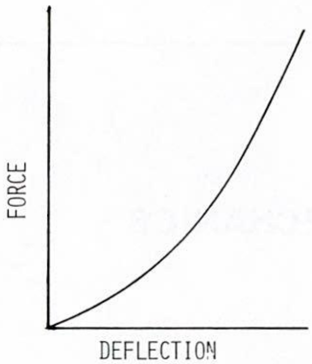


Figure 15. Force-deflection curve in compression.

BEHAVIOR UNDER DIRECT COMPRESSION

Linear Elastic Analysis

Bearings tested to high compressive strains display a nonlinear, stiffening, force-deformation curve as shown in Figure 15. To describe this behavior accurately requires a nonlinear theory, but in the interests of simplicity most bearings under compression are analyzed assuming linear elasticity and infinitesimal strains. Special assumptions are needed to avoid the numerical difficulties that arise when ν, Poisson's ratio, approaches 0.5. Conversy *(49)* appears to have been the first to present a small deflection analysis for nearly incompressible materials. Others *(52, 53)* have subsequently published work along very similar lines. The basic assumptions are:

1. Points lying on a normal (vertical line) in the unstrained rubber lie on a parabola after deformation.
2. At any point the direct stresses are all equal to each other ($\sigma_{xx} = \sigma_{yy} = \sigma_{zz} = p(x,y)$).
3. And p is zero along the lateral faces.

Rectangular coordinates are used for rectangular bearings, the hydrostatic pressure, $p[x,y]$, is taken as the basic variable, and displacements in the x,y,z directions are u,v,w. From assumption 1,

$$u(x) = u_o(x)\left(1 - \left(\frac{2z}{t}\right)^2\right) \quad (5)$$

where u_o is the lateral displacement at mid-depth.

Using the sign convention that compression is positive, the change in volume due to bulging in the x-direction of a typical element is (see Fig. 16):

$$\Delta V = \bar{\epsilon}_c \, t dx dy = \frac{\partial}{\partial x}\left(\frac{2}{3} t u_o(x)\right) dx dy - \frac{\partial}{\partial y}\left(\frac{2}{3} t v_o(x)\right) dx dy \quad (6)$$

where $\bar{\epsilon}_c$ is the compressive strain. However,

$$\Delta V = t \, dx dy \frac{p}{K} \quad (7)$$

where K is the bulk modulus of rubber, and so

$$\frac{tp}{K} = \bar{\epsilon}_c t - \frac{2t}{3}\left(\frac{\partial u_o}{\partial x} + \frac{\partial v_o}{\partial y}\right) \quad (8)$$

The shear stress at the steel–rubber interface is

$$\bar{\tau}_{zx} = G \left.\frac{\partial u}{\partial z}\right|_{z = \frac{t}{2}} = \frac{-4Gu_o}{t} \quad (9a)$$

but to satisfy equilibrium it must also be

$$\bar{\tau}_{zx} = +t \frac{\partial p}{\partial x} \quad (9b)$$

so

$$u_o = -\frac{1}{8}\frac{t^2}{G}\frac{\partial p}{\partial x} \quad (10)$$

Using similar relations in the y-direction one obtains

$$\frac{\partial^2 p}{\partial x^2} + \frac{\partial^2 p}{\partial y^2} - \frac{12p}{t^2}\frac{G}{K} = -\frac{12G\bar{\epsilon}_c}{t^2} \quad (11)$$

This is the fundamental equation, which can then be solved as a given function of the imposed vertical strain $\bar{\epsilon}_c$. For circular bearings, the Laplacian operator

$$\left(\frac{\partial^2}{\partial_x^2}+\frac{\partial^2}{\partial y^2}\right)$$

becomes

$$\left(\frac{\partial^2}{\partial r^2}+\frac{1}{r}\frac{\partial}{\partial r}+\frac{1}{r^2}\frac{\partial^2}{\partial \theta^2}\right)$$

For an incompressible material, Eq. 11 reduces to Poisson's equation. This same equation governs the torsion of solid shafts, except that the meaning of the variable, p, is different in the two cases; therefore, mathematical solutions for the two problems are interchangeable and existing solutions for one may be used for the other. If the distributions of p in the x and y directions are assumed to be independent, the equation can be solved by separation of variables and all other quantities are then calculable from (Eq. 5) − (Eq. 10). The quantities of greatest interest are

$$\text{Compressive strain} = \bar{\epsilon}_c = \frac{\Delta_c}{t} = \frac{\overline{\sigma}_c}{f_c E} = \frac{\overline{\sigma}_c}{E_c} \tag{12}$$

and the maximum shear stress in rubber (at the mid-point of the long side)

$$\bar{\tau}_{zx} = g_c\,\overline{\sigma}_c \tag{13}$$

where $\overline{\sigma}_c = \dfrac{P}{A}$ = average compressive stress and E_c is the apparent modulus which varies with geometry. E, Young's modulus, has been substituted for $3G$ in Eq. 11. This is strictly valid only for incompressible materials, but introduces negligible error for nearly incompressible materials, such as rubber. For rectangular bearings with dimensions as shown in Figure 13, the dimensionless coefficients, f_c and g_c, are defined by:

$$f_c = \frac{32}{\pi^4}\left(\frac{2a}{t}\right)^2 \sum_{1,3,5}^{\infty} \frac{1}{R_n^2 n^4}\left(1-\frac{\tanh\theta_n}{\theta_n}\right) \tag{14}$$

and

$$g_c = \frac{\pi^2}{8}\frac{t}{a}\,\frac{\left[\sum_{1,3,5}^{\infty}\frac{1}{R_n^2 n^4}(1-\operatorname{sech}\theta_n)\right]}{\left[\sum_{1,3,5}^{\infty}\frac{1}{R_n^2 n^4}\left(1-\frac{\tanh\theta_n}{\theta_n}\right)\right]} \tag{15}$$

where

$$R_n^2 = 1 + \frac{4E}{K}\left(\frac{2a}{n\pi t}\right)^2 \tag{16}$$

and

$$\theta_n = \frac{nR_n\pi b}{2a} \tag{17}$$

Numerical values can be seen in Figures 17 and 18 for typical cases. Rejcha (*52*) presented the same analysis but for the incompressible case only. Topaloff (*53*) did the same, but solved Eq. 11 (without the third term) without assuming separation of variables. He presented different series solutions for f_c and g_c, but Gent and Meinecke (*54*) demonstrate that the series have the same sums as Conversy's.

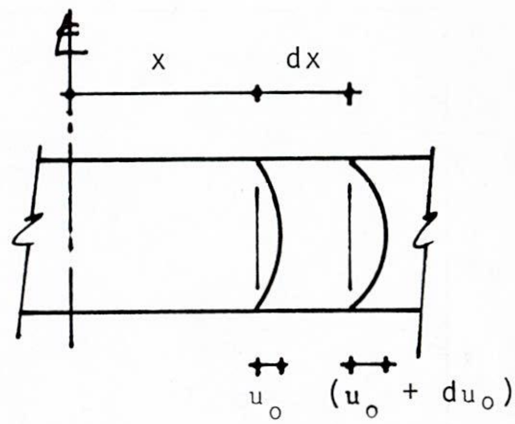


Figure 16. Dimensions of deformed elastomer layer.

Gent and Lindley (*45*) addressed the problem for circular blocks and infinitely long strips, and later Gent and Meinecke extended the solution to a variety of shapes. However, they approached the problem differently. They view the deformed state as the superposition of:

1. The vertical load applied to a system where the reinforcement offers no lateral restraint (i.e., the plates are perfectly greased), resulting in pure homogeneous compression with unrestrained lateral expansion.
2. An applied stress distribution (direct and shear), which causes the rubber elements on the interface to deform laterally in an exactly equal and opposite way to load case 1. Vertical displacement at the interface is suppressed and the free faces bulge.

Thus,

$$P = P_1 + P_2 = EA\bar{\epsilon}_c\ (f_{c1} + f_{c2})$$

Under a given vertical strain $\bar{\epsilon}_c$, the vertical load in case 1 is given as

$$P_1 = f_{c1}AE\bar{\epsilon}_c = AE\bar{\epsilon}_c\left[\frac{4}{3}-\frac{2}{3}\,\frac{(ab+t^2)}{(a^2+b^2+2t^2)}\right] \tag{18}$$

This single expression was chosen for any rectangular cross section because it gives the true solution for the cases $b/a = \infty$ and 1.0 and good approximations for intermediate conditions. For the second load case, Eq. 11 is used with $E/K = 0$, and thus Gent's solution is identical to the others except that it has an extra term. The two solutions are compared for rectangular bearings in Figure 17. The solutions are essentially the same except at small shape factors. Gent's solution converges on the true uniaxial stress condition with a zero shape factor, while the other solutions predict a zero stress at this extreme condition. For circular bearings and

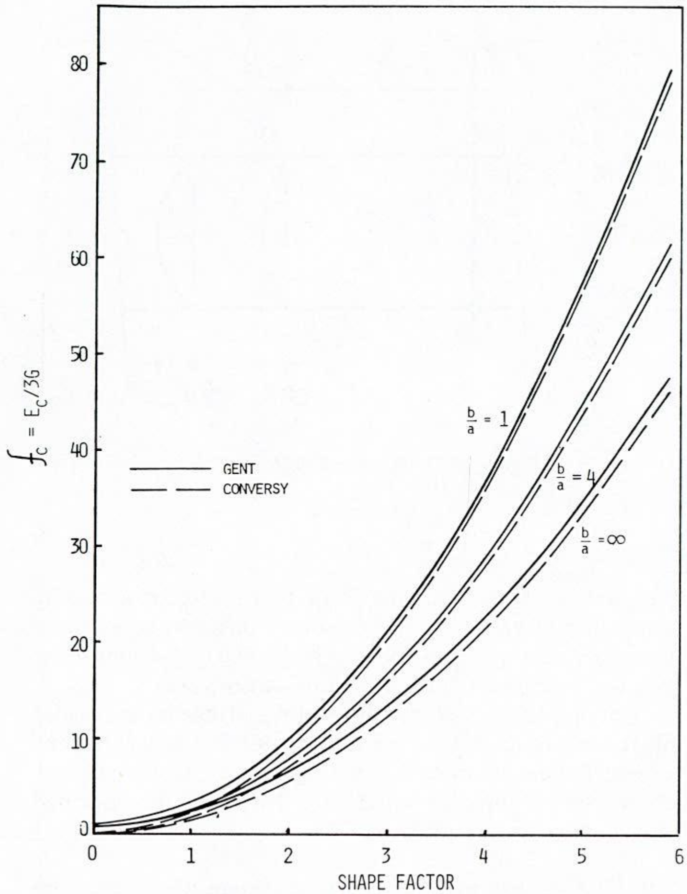


Figure 17. Comparison of effective compression moduli calculated by Gent's and Conversy's theories—incompressible material.

long rectangular strips, Gent's solution simplifies to $E_c/E = (1 + 2S^2)$ and $4/3\ (1 + S^2)$, respectively.

Conversy's analysis accounts explicitly for the deformation due to bulk compression of the rubber. Other derivations present only an incompressible analysis. Gent and Lindley (*48*) suggest that the bulk modulus may be accounted for approximately by the expression

$$\frac{1}{E_c'} = \frac{1}{E_c} + \frac{1}{K} \tag{19}$$

where E_c is the apparent compression modulus for an incompressible material. E_c' is the apparent modulus accounting for compressibility. This formulation has the virtue of simplicity, but it implies that the bulk compression is uniform throughout the rubber. Figure 18 shows a comparison of the two approaches for a variety of shape factors and material properties, and demonstrates that bulk compression is important in bearings with large shape factors and high values of $3G/K$. The approximate method (Eq. 19) gives reasonable values in most cases, and predicts stiffness values that are, if anything, too high. $3G/K$ is generally assumed to be about 0.001 for bridge bearing rubber.

Nonlinear Formulations of the Compression Problem

The foregoing solutions all use linear elastic material behavior with infinitesimal strains. These theories are most commonly used in practice, but they do not recognize the full complexity of bearing behavior. Holownia (*55*) proposed a solution which considers the consequences of finite strain. He rewrote Eq. 11 as

$$\frac{\partial^2 p}{\partial x^2} + \frac{\partial^2 p}{\partial y^2} - \frac{12p}{t^2}\frac{G}{K} = -\frac{12G}{t^2}\frac{\bar{\epsilon}_c}{(1-\bar{\epsilon}_c)^3} \tag{20}$$

This accounts for the fact that the deformed bearing has a height $t(1 - \bar{\epsilon}_c)$ rather than t. The force deformation relation for the bearing thus becomes

$$P = AE_c \frac{\bar{\epsilon}_c}{(1-\bar{\epsilon}_c)^3} \tag{21}$$

where

$$\bar{\epsilon}_c = \frac{\Delta c}{T} \tag{22}$$

This represents a bearing which stiffens under increasing strain as noted in Figure 15.

The kinetic theory of rubber-like solids (*10*) is another simple theory that models nonlinear uniaxial behavior. In the absence of lateral restraint

$$\sigma = -G\,(\lambda - \lambda^{-2}) \tag{23}$$

where

$$\lambda = \frac{\text{Strained length}}{\text{Unstrained length}} = \text{Extension ratio}$$

and compression stress is positive. This results in uniaxial behavior which softens in tension and stiffens in compression as shown in Figure 19. Gent (*56*) suggests that Eq. 23 can be combined with the apparent modulus E_c to give

$$\overline{\sigma}_c = -\frac{E_c}{3}(\lambda - \lambda^{-2}) \tag{24}$$

for a complete bearing.

A similar relation can be derived by assuming that the elastic modulus, E, is a constant which relates incremental uniaxial stress and strain. That is,

$$\frac{dP}{A} = E\frac{d\ell}{\ell} \tag{25}$$

where the cross-sectional area, A, varies with extension to maintain constant volume and ℓ is the deformed length of the element. The result for uniaxial stress with no lateral restraint is

$$\sigma = -E\,(1 - 1/\lambda) \tag{26}$$

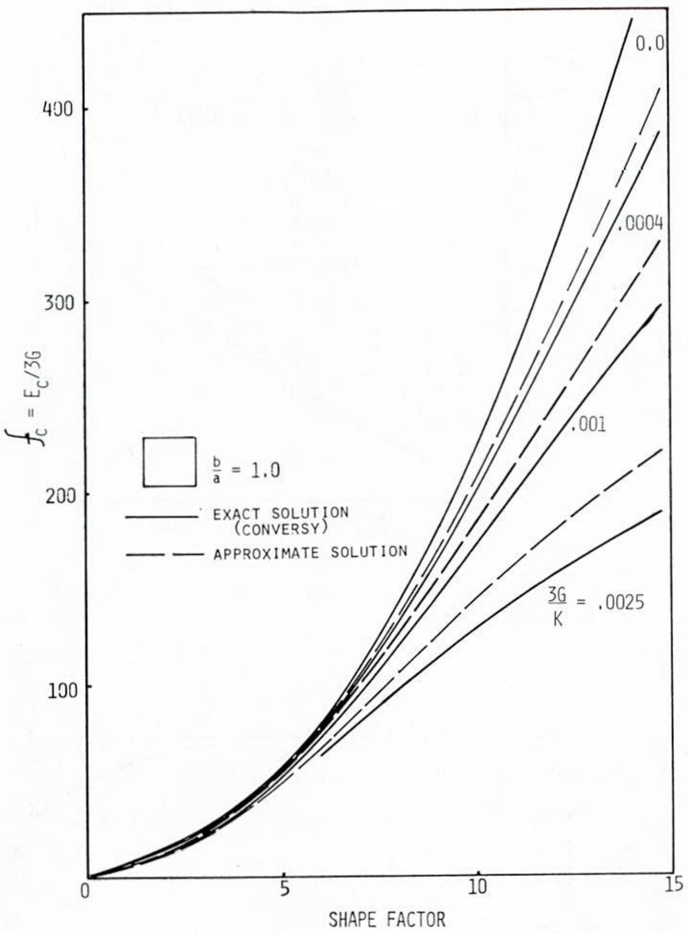


Figure 18(a). Effect of bulk compression, comparison of approximate and exact solutions—square bearing.

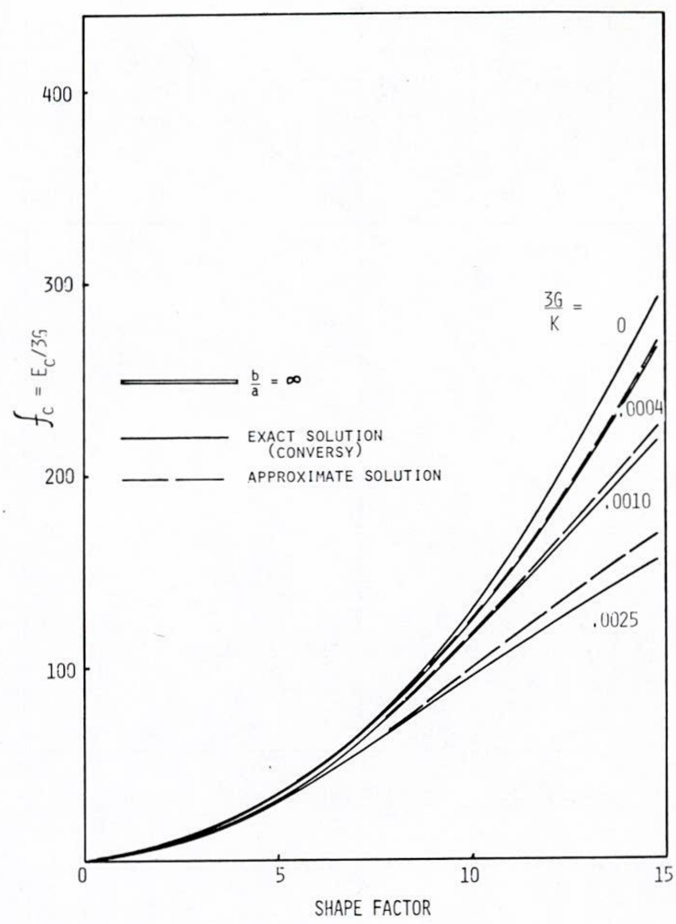


Figure 18(b). Effect of bulk compression, comparison of approximate and exact solutions—infinite strip.

Note that $\lambda = 1 - \bar{\epsilon}_c$, and for small strains Eqs. 23 and 26 both reduce to

$$\sigma = E\epsilon \tag{27}$$

The three nonlinear theories are compared in Figure 20.

Lindley (*57*) developed a large deformation plane stress finite element computer program, which was later modified to include bulk compression and plane strain. This program (*59*) was then used to compute effective compression moduli (at small strains) for infinitely long strips with different shape factors and material properties. These apparent moduli are compared with other approaches in Figure 21 and agreement is seen to be good in the range of practical interest. This technique appears to be a useful tool for research, but, because computing times are long for near-incompressibility, it is unlikely to find application in everyday bridge bearing design. Lindley (*60*) has also presented a closed-form solution for long strips of compressible material, which is also plotted in Figure 21.

Other Stresses in the Compressed Bearing

The bulging due to compression sets up shear stresses that are greatest at the steel–rubber interface. The value of these stresses can be predicted by linear elastic theory by combining Eqs. 12 and 13. Thus,

$$\tilde{\tau}_{zx} = g_c f_c \, E\bar{\epsilon}_c \tag{28}$$

All the linear solutions predict the same shear stresses and strains with incompressible rubber, since the first component of Gent's solution produces only uniaxial stress and causes no shear stress in the plane of interest. The inclusion of bulk compression reduces the shear stresses and strains for a given vertical strain. The shear strain, γ_c, caused by a unit compressive strain varies with the aspect ratio as shown in Figure 22. This shows that the commonly accepted value of shear strain

$$\gamma_c = 6S\bar{\epsilon}_c \tag{29}$$

is valid only for infinite strips and circular bearings, and underestimates the shear strain for other shapes (*4*).

The reinforcement in the bearing restrains bulging, and the resulting shear stresses between the rubber and the reinforcement surface cause tensile stresses in the reinforcement. No general solutions for these tensile stresses have been derived, but they can be computed for an incompressible strip as shown in Figure 23. Note that this analysis

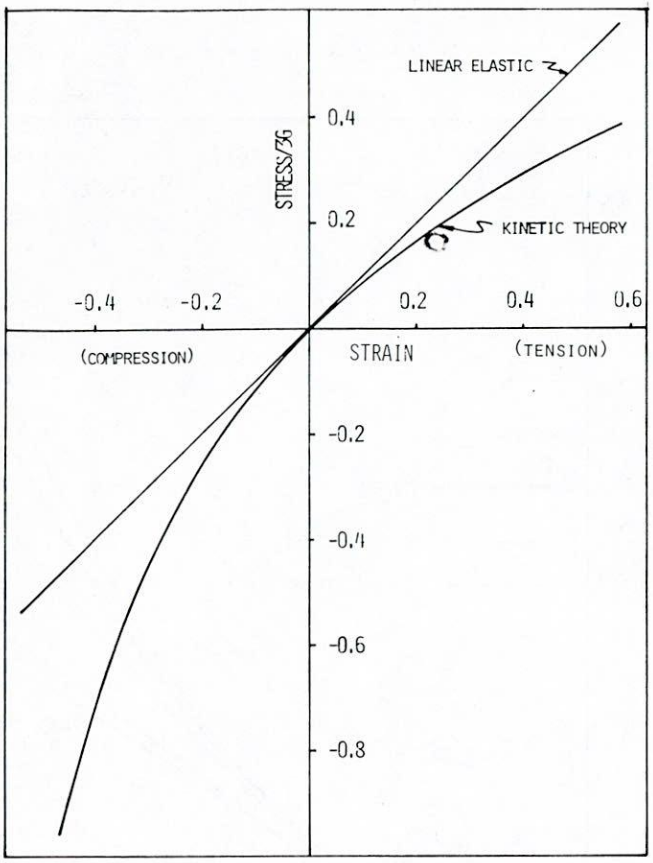


Figure 19. Kinetic theory nonlinear uniaxial stress–strain curve.

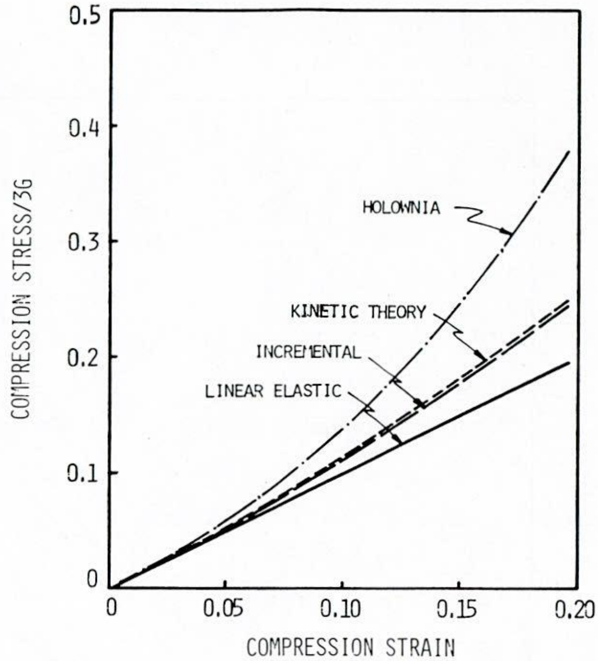


Figure 20. Nonlinear stress–strain corrections for large deformations.

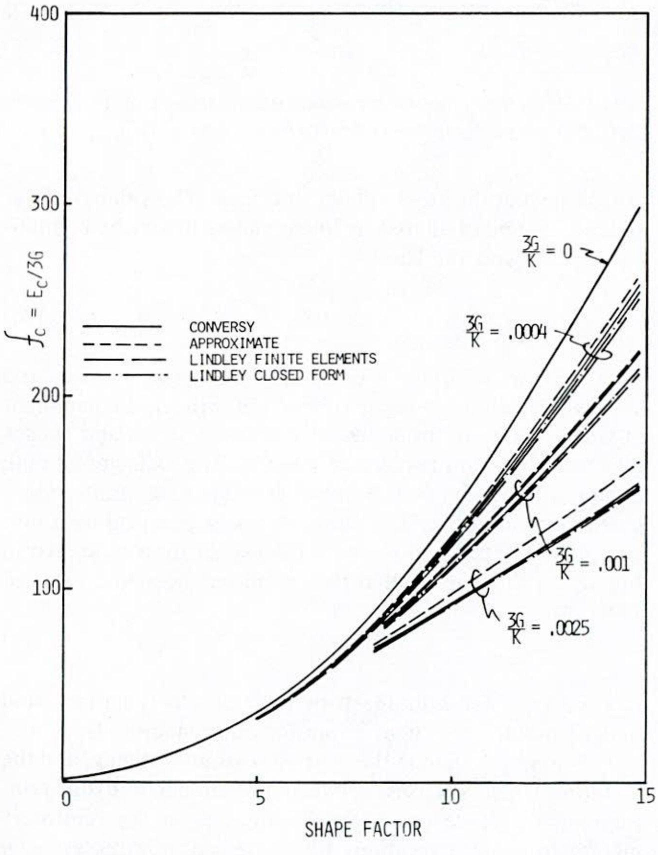


Figure 21. Effective compression modulus for an infinite strip calculated by different methods.

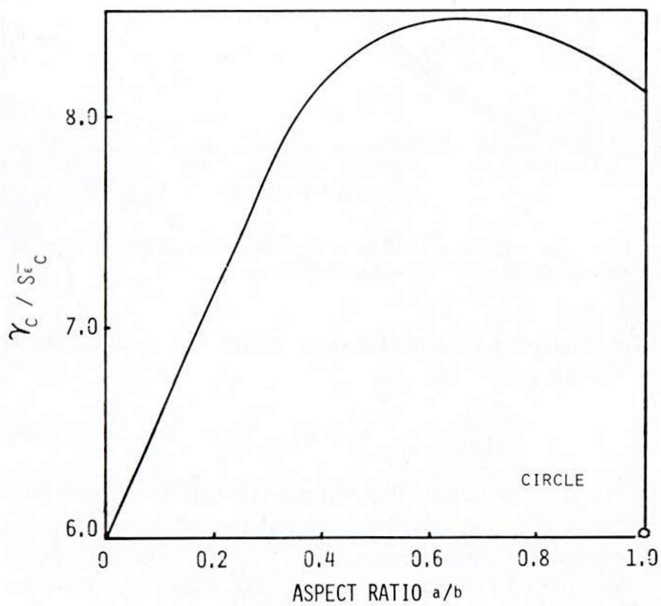


Figure 22. Nondimensional shear strain caused by compression as a function of aspect ratio.

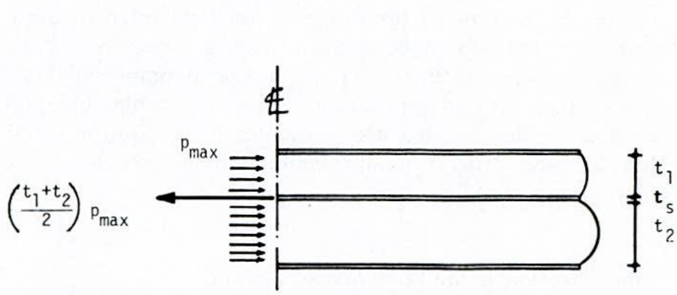


Figure 23. Internal stresses on reinforcement.

assumes the reinforcement is inextensible compared to the rubber, so it may not be realistic for fiberglass reinforcement. Static equilibrium requires that

$$\sigma_{s,\max} = \left(\frac{t_1 + t_2}{2t_s}\right) p_{\max} \tag{30}$$

where $p_{\max}$ is the maximum hydrostatic compression in the rubber. This can be related to the applied load by using Gent's theory (*54*).

$$p_{\max} = f_{s2} E \bar{\epsilon}_c \tag{31}$$

where

$$f_{s2} = \frac{16}{\Pi^3}\left(\frac{2a}{t}\right)^2 \sum_{1,3,5}^{\infty} \frac{1}{n^3}\left(-1\right)^{\frac{n-1}{2}} \left(1 - \operatorname{sech}\frac{n\pi b}{2a}\right) \tag{32}$$

However,

$$\bar{\sigma}_c = \frac{P}{A} = (f_{c1} + f_{c2})\, E \bar{\epsilon}_c \tag{33}$$

Thus,

$$\frac{\sigma_{s,\max}}{\bar{\sigma}_c} = \left(\frac{f_{s2}}{f_{c1} + f_{c2}}\right)\left(\frac{t_1 + t_2}{2t_s}\right) \tag{34}$$

so

$$\sigma_{s,\max} = \left(\frac{f_{s2}}{f_{c1} + f_{c2}}\right)\left(\frac{t_1 + t_2}{2t_s}\right)\bar{\sigma}_c \tag{35}$$

For a long strip, $f_{c1} = 4/3, f_{c2} = 4/3\ S^2$, and $f_{s2} = 2\ S^2$. Thus,

$$\sigma_{s,\max} = \left(\frac{t_1 + t_2}{2t_s}\right)\left(\frac{1.5}{1 + 1/S^2}\right)\bar{\sigma}_c \tag{36}$$

and this can be approximated as

$$\sigma_{s,\max} \doteq 1.5\left(\frac{t_1 + t_2}{2t_s}\right)\bar{\sigma}_c \tag{37}$$

for reasonably large shape factors.

Discussion of Compression Theories

The preceding sections have described several theories that predict the behavior of elastomeric bearings under compression. The theories are somewhat unusual because they must account for bulging, finite strains, near incompressibility, and material nonlinearity. A number of points are worth noting. First, none of the theories satisfies equilibrium for all bearings. That is, the equilibrium condition

$$\tau_{ij,i} = 0 \tag{38}$$

is not satisfied exactly, reflecting the approximations implicit in the assumptions. Secondly, none of the theories predicts tensile stresses at the peak of the bulge. However, there is considerable evidence that these stresses exist because horizontal splits sometimes occur at the edge of bearings when they are tested under large compressive loads.

The constitutive theories do not describe the full complexity of the material behavior. For this reason and others, Gent and Lindley (*48*) found that the best fit with the experimental results was obtained by introducing an empirical constant k to give

$$\bar{\sigma}_c = E\,(1 + 2kS^2)\,\bar{\epsilon}_c \tag{39}$$

For a truly elastic material, k has the value 1.0. Figure 24 shows the variation of k with hardness required to give good correlation with experiments. Its value reflects the influence of carbon black on the mechanical properties of the compound.

For design purposes, the behavior of a bearing is often defined only in terms of its shape factor, disregarding other aspects of its geometry. Bearings of different shape (i.e., square, circular, rectangular, etc.), but of the same shape factor, will not behave in precisely the same way, as can be seen from Figures 17 and 18. However, if they could be treated as doing so, the calculations would be simplified. The question then is whether the approximation is acceptable in the light of other uncertainties, such as true mechanical properties, precision of loads, construction tolerances on the bridge, manufacturing tolerances on the rubber quality and bearing dimensions, etc. A comparison by Spitz (*62*) suggests that variations in rubber quality and manufacturing tolerances will cause variations greater than those discussed here, thus lending support to the practice of using the shape factor as the sole geometric parameter to be considered.

SHEAR DEFORMATION

Lateral motion of one loaded surface relative to the other is accommodated by both bending and shear (as shown in Fig. 25). The flexural component of the deformation is generally small, even when interaction with compressive load (the $P - \Delta$ effect) is taken into account. Lateral bulging under compression and details of the boundary conditions also have a modest effect on the shear component. However,

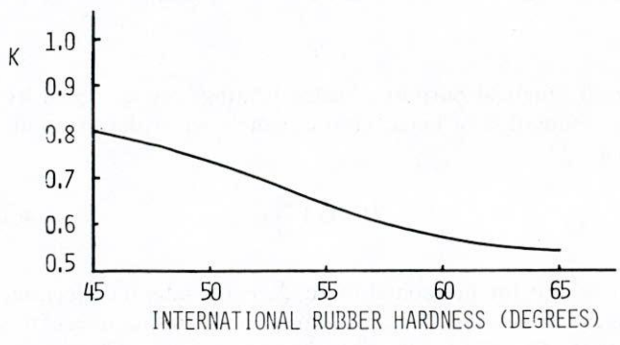


Figure 24. Material constant k *as a function of hardness.*

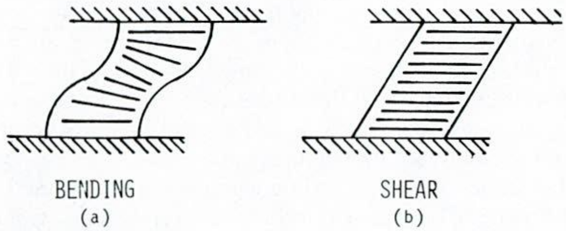


Figure 25. Lateral deflection of bearing.

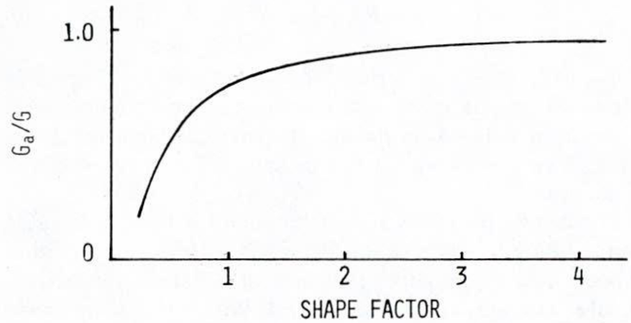


Figure 26. Effective shear modulus as a function of shape factor for infinite strips.

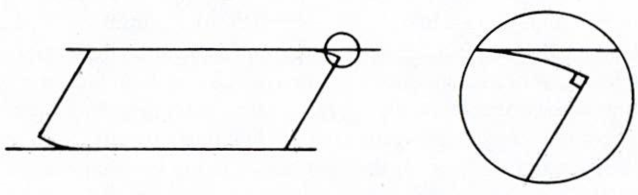


Figure 27. Rollover at edge of bearing.

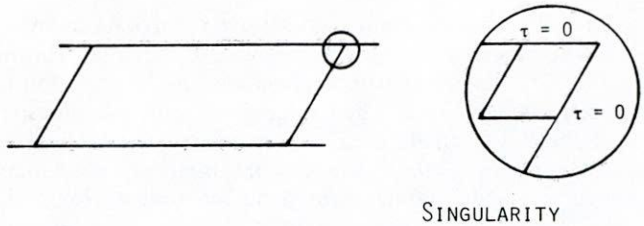


Figure 28. Shear stress singularity at edge of bearing.

for all practical purposes bridge bearings are designed by the assumption of linear elastic, simple shear deformation. That is,

$$H = GA\,\frac{\Delta_s}{T} \tag{40}$$

where H is the horizontal force, Δ_s is the lateral deflection, T is the total rubber thickness, and A is the surface area of the bearing. The effect of the boundary conditions can be accommodated by substituting for G an apparent shear modulus G_a for which values are plotted against shape factor in Figure 26. The bearing may stiffen at low temperatures, but the higher G value need be used only for the incremental deformation, so that

$$H = \frac{A}{T}\sum_i G_i\,\Delta_{si} \tag{41}$$

While the above equations are widely accepted for the theoretical prediction of shear deformation, variations in this behavior have been noted. Typical deformation of a bearing under shear load with no axial compression is as shown in Figure 27. The deformation is almost identical to simple shear except that there is a rollover of the bearing corner. The shear stresses are nearly uniform over the width of the bearing, but there is a sharp discontinuity in the shear stress at the edge. This discontinuity is necessary to satisfy statics as shown in Figure 28. Any compressive stresses will also vary as shown in Figure 29, because they have to resist the overturning moment caused by the shear force. These observations have been confirmed in experiments (*51*) and theoretical calculations (*50*). The rollover and shear stress discontinuity are both local effects, which have only a small effect on the bearing stiffness. However, it has been suggested that the rollover can contribute to local debonding of the laminate at the edge of the bearing or cause flexure in the plates if they are thin.

ROTATION OF THE BEARING

The analysis of rotation follows closely that of bonded compression. The rotational stiffness of the bearing is of interest both to ensure that moments introduced into the bridge supports are not excessive and to assure that the bearing is flexible enough to sustain the required rotation without lift-off of the girders (see Fig. 30). Rotation may also cause significant stresses in the rubber. Conversy (*49*) analyzed compressible rectangular bearings at small deformations. The fundamental equation (analogous to Eq. 11) is

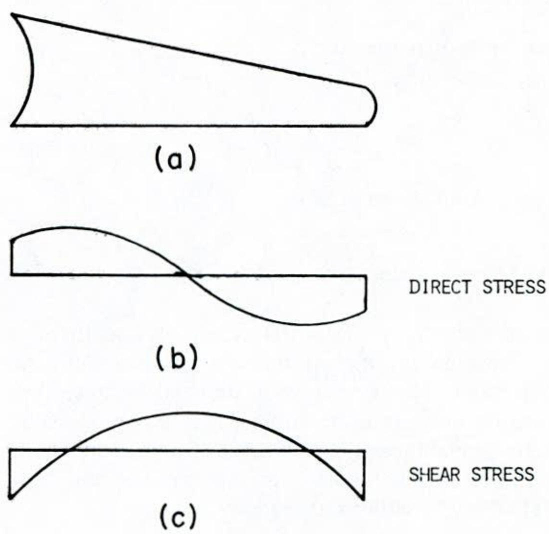


Figure 29. Stresses caused by rotation.

$$\Delta_p^2 - \frac{4pE}{t^2K} = -\frac{4E}{t^3}\alpha x \qquad (42)$$

in which α is the angle of rotation about the Y-axis. The important solutions are:

$$M = E_r \frac{I\alpha}{t} = f_r \frac{E(2a)^3\,(2b)}{12}\frac{\alpha}{t} \qquad (43)$$

where

$$f_r = \frac{E_r}{E} = 3\left(\frac{a}{\pi t}\right)^2 \sum_1^\infty \frac{1}{n^4 Q_n^2}\left(1 - \frac{1}{\phi_n}\tanh\phi_n\right) \qquad (44)$$

$$\phi_n = Q_n\left(\frac{n\pi b}{2a}\right) \qquad (45)$$

and

$$Q_n = 1 + \frac{E}{K}\left(\frac{2a}{n\pi t}\right)^2 \qquad (46)$$

Note that the apparent modulus in bending (E_r) is different from that in bonded compression (E_c).

The distribution of stresses due to rotation is shown in Figure 29. For bearings with $b/a > 1/2$, the maximum shear stress occurs at $y = 0$, $x = \pm a$, in the center of the most heavily loaded edge. Then,

$$\tau_{zx,\,\max} = g_r E\alpha \qquad (47)$$

where

$$g_r = \left(\frac{2a}{\pi t}\right)^2 \sum_1^\infty \frac{1}{n^2 Q_n^2}\,(1 - \operatorname{sech}\phi_n) \qquad (48)$$

For bearings with $b/a < 1/2$ (unlikely in practice), the maximum shear stresses may occur on the axis of rotation at the bearing edge $y = \pm b$. Values of g_r are plotted in Figures 31(a) and 31(b) for the incompressible case.

Gent and Meinecke (*54*) provide an incompressible analysis for a variety of shapes. They again employed a super-

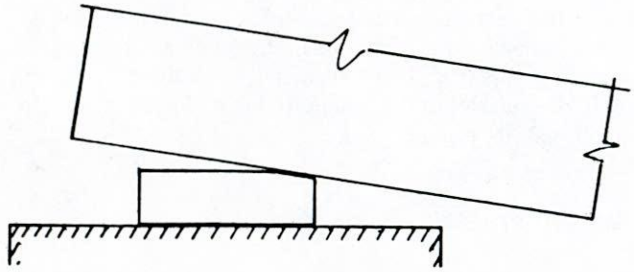


Figure 30. Girder lift off caused by excessive rotation.

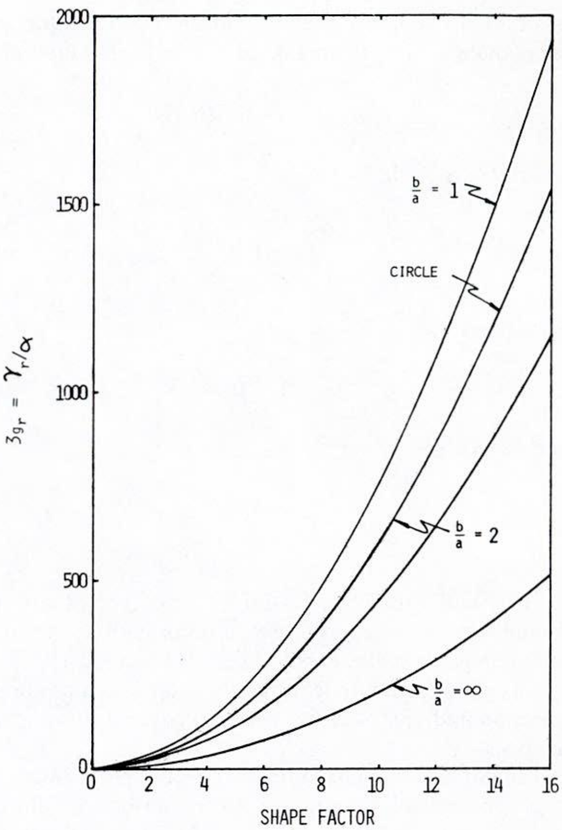


Figure 31(a). Maximum shear strain per unit rotation vs. shape factor—large shape factors.

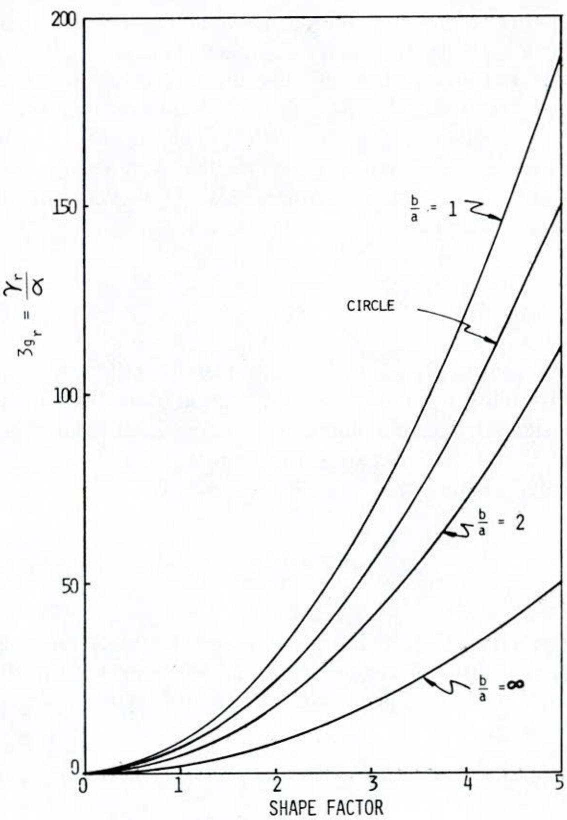


Figure 31(b). Maximum shear strain per unit rotation vs. shape factor—small shape factors.

position of two linear elastic solutions and thus they added a term analogous to P_1 in Eq. 18. Their formulation gives

$$M = f_r EI\alpha \tag{49}$$

where for a circle,

$$f_r = 1 + \frac{2}{3} S^2 \tag{50}$$

for a square,

$$f_r = 1 + 0.742\, S^2 \tag{51}$$

and for a strip,

$$f_r = \frac{4}{3} + \frac{1}{4} S^2 \tag{52}$$

Lindley and Teo (*59*) used finite elements to obtain rotation moduli for strips based on small deformations but including bulk compressibility. Lindley (*63*) also derived a plane strain, small deflection analytical solution which includes bulk compression and gives results within 6 percent of his computer analysis.

Tamhankar and Chhauda (*64*) recommend obtaining the appropriate vaue of E in Eq. 43 by performing direct compression tests and then using Eq. 12. Various simplifications of Eq. 43 have also been suggested (*64*) for design purposes.

If the girder end rotates excessively, it will lift off the bearing on one side, increasing significantly the pressure on those areas that remain in contact. This is clearly undesirable, but no efforts have been made to establish the critical angle at which it will occur. Preliminary analysis by the authors suggests that the critical angle increases with compressive stress, but will not be the dominating factor for bearings of common dimensions. However, the subject merits more detailed investigation.

STABILITY

A bearing that is too tall and slender will fail by buckling (instability). Conventional calculation of the buckling load of a slender elastic column with ends fixed against rotation takes into account only the bending stiffness, EI, to give Euler's equation,

$$P_{cr} = P_E = 4\frac{\pi^2 EI}{\ell^2} \tag{53}$$

However, if the column has shear flexibility (such as in a laced column), Timoshenko and Gere (*66*) show that the buckling load will be reduced to the equation derived by Haryngx (*67*).

$$P_{cr} = \frac{\sqrt{1 + 4\,P_E/P_G} - 1}{2/P_G} \tag{54}$$

where $P_G = GA_s$ and A_s is the effective shear area. If P_E/P_G approaches infinity, the critical load approaches $\sqrt{P_E P_G}$. If P_E/P_G approaches zero, the critical load approaches the Euler load, P_E. Rubber bearings are flexible in shear but relatively stiff in flexure and compression, and it is approximately correct to say

$$P_{cr} = \sqrt{P_E P_G} = \frac{2\pi aAG}{T}\sqrt{f_r} \tag{55}$$

where T is the total bearing thickness. For practical cases the term f_r can be conservatively estimated by using Eq. 52, and the critical stress can be approximately defined as

$$\sigma_{cr} = \frac{\pi GaS}{T} \tag{56}$$

This equation is used in several design codes (*5, 15*) with a suitable factor of safety (e.g., $3\pi/4$) to convert critical stress to allowable stress.

The foregoing formulation of the buckling problem is reasonable, but the transition from Eq. 54 to Eq. 55 is unconservative if P_E/P_G is large but not infinity. For example, if $P_E/P_G = 100$, Eq. 55 overestimates P_{cr} by approximately 5 percent. Further, the estimate of the shear stiffness, GA_s may be unrealistically large, and this may also produce an overestimate of P_{cr}. Gent (*68*) corrected for these difficulties. He started with an equation similar to Eq. 52, and used an apparent shear modulus, G_a,

$$G_a = \frac{G}{1 + \dfrac{\ell^2 GA_s}{48E_rI}} \tag{57}$$

and apparent bending stiffness, E_rI. However, the use of Eq. 52 to develop Eq. 56 introduces a conservative error, which may be on the order of 2 to 3 for practical bearings. This theory was further developed by Schapery and Skala (*69*) to account for bearings with curved steel plates (with applications in helicopter bearings). Both theories assume that the plates are rigid in flexure. If they are not, an overestimate of the critical stress may result, particularly for bearings of large shape factor.

Care should be taken to use the correct expression for P_E in Eq. 53. If one point on the bridge is restrained against lateral movement,

$$P_E = \frac{4\pi^2 f_r EI}{T^2} \tag{58}$$

because the bearing resembles a column fixed against rotation and translation at both ends. However, if the bridge is free at all points (i.e., it is "elastically restrained"), P_E will be only one quarter of that value, reducing P_{cr} (from Eq. 54) to about half its previous value.

COMBINED STRESS

Elastomeric bearings generally experience a combination of loadings. Superposition may be used when considering very small deformations with linear elastic behavior, but elastomeric bearings frequently exceed these condition. Despite these limitations superposition is used for nearly all practical applications because no better alternatives are available. Thus, shear stresses or strains due to compression, rotation, and shear deformation are added directly even when the summation exceeds strains of several hundred percent. The problem is particularly severe when combining compression and shear deformation. Geometric effects enter into this load combination as noted in Chapter Three. This is believed to be one of the major unanswered questions in the behavior of elastomeric bearings.

FRICTION AND SLIP OF BEARINGS

A bearing may slip if subjected to excessive horizontal force. Common causes are insufficient allowance for shrinkage and creep of prestressed concrete girders, girder placement at extreme temperatures, construction misalignment, etc. Although a single slip is unlikely to do damage, cyclic slip deteriorates the outside of the rubber and may precipitate cracking earlier than would otherwise be the case. In extreme cases the bearing might slip out of position. Field experience suggests that plain pads may be more susceptible to slipping than reinforced bearings. Part of the problem is that the coefficient of friction of rubber varies with load and other conditions. Slipping may be prevented by introducing lips, stops, or dowels or by using external steel plates that have a reliable coefficient of friction against concrete and can be directly connected to steel girders. Other positive connections can be made with sole plates that are bonded to the bearings and positively attached to the bridge piers or girders.

One solution to the slip problem is to use a PTFE slider plate in combination with the elastomeric bearing (*72*), but design is complicated by the fact that neither material obeys the simple laws of friction widely accepted for most materials. However, the principle of such a bearing is that large, one-time movements such as creep and shrinkage are absorbed by slip of the PTFE, while smaller cyclic movements such as temperature effects are taken by elastic shear deformation of the rubber.

In general the coefficient of friction of both materials reduces with load (PTFE from about 0.25 to about 0.04) (*73*). That for PTFE also varies with lubrication and slide path. That for rubber may vary with compounding and humidity. (Antiozonants include waxes which migrate to the surface and may influence the friction coefficient.) Bonding PTFE to rubber is extremely difficult (*74*), and the usual arrangement is to interpose a steel plate between them. The PTFE then slides on a stainless steel plate as shown in Figure 3. The PTFE is apt to creep out laterally under vertical load, but this may be prevented by recessing it into the lower steel plate.

THEORIES OF FAILURE

Rubber subjected to hydrostatic compression is almost indestructible. Cases have been reported (*71*) of bearings being tested up to 18,000 psi with no apparent damage at all. However, many other failure modes are possible. These include:

1. Fatigue of the rubber.
2. Rupture or yield of the steel plate due to tensile stress.
3. Bond failure or separation of the reinforcement from the rubber.
4. Buckling or instability which may readily occur even for reasonably short bearings.
5. Excessive forces in the bearings overstressing bridge piers or subassemblages. This may be a particular problem when the bearings are cold because of crystallization and/or thermal stiffening.
6. Internal rupture caused by hydrostatic tensile stresses in the rubber.
7. Creep, giving excessive deflection and serviceability problems.
8. Material deterioration with age.
9. Finally, the limitations of the theories, calculations, and design procedures may so limit the accuracy of the design procedures that problems occur.

These failure modes have been discussed in Chapters Three and Four. In most cases the questions are not fully answered, but some guidelines are available in all cases. These chapters have attempted to present these guidelines and describe the limitations.

CHAPTER FIVE

EXPERIMENTAL WORK

GENERAL

A very large number of experimental programs have been conducted, ranging from tests on a single bearing to comprehensive studies of a range of aspects of behavior. The results are scattered and only the very fundamental questions are free from conflicting evidence. This is in part caused by inevitable variations in material properties, but failure to control unwanted variables and record all relevant information often exacerbates the difficulty. Differences in procedures (number of load cycles before taking readings, rate of loading, etc.) also make the drawing of conclusions difficult.

The following sections describe some of the major comprehensive studies and address the correlation between theory and experiment.

COMPREHENSIVE TEST PROGRAMS

du Pont Test *(3, 75)*

The du Pont Company has performed two extensive test programs. The first formed the basis for the company's catalog (*3*), which outlines a simple design methodology and provides typical compression stress–strain plots for CR bearings. Since its publication in 1959, it has been one of the most used sources of design information. It fails to address many important questions in elastomeric bearing design such as fatigue, rotational capacity, and the stress in the laminate, but few bearings designed according to this document have suffered distress.

A second study (*75*) has just been completed. Shear and compression tests were performed on CR bearings reinforced with glass-fiber or steel laminates and on unreinforced pads. Fatigue, serviceability limits, rotational effects, and combined loadings were not considered, and the main effort was concentrated towards providing improved deflection data under shear and compression at more realistic stress–strain levels. Previous data included results up to 45 percent compressive strain, which exceeds limits used in practice. The full test results are not yet available but are expected to provide a useful contribution to the experimental knowledge. Some of the reinforced bearings were tested to failure under monotonically applied load. Nearly all of these bearings failed at high compressive stresses by rupture of the reinforcement. Fiberglass reinforcement typically failed when compressive stresses on the bearing reached approximately 2,000 psi, while the steel laminated bearings sustained considerably higher stresses. The steel laminates were 14 gage regardless of bearing size or shape factor. The greater strength of the steel reinforced bearings emphasizes the importance of the strength of the laminate. However, it should again be emphasized that this test program neglected several factors that are very important to the design of bearings.

ORE Study 1964 *(33)*

This study is important because it forms the basis of the widely used UIC 772R Specifications. The test series was ambitious, covering many combinations of shear and axial load (both static and cyclic), eccentric load, low temperature behavior, and tests to destruction. The tests were conducted on CR and NR commercial bearings representing a variety of design and manufacturing techniques.

The results verified many accepted trends, such as the dependence of compression stiffness on shape factor. The authors of the study also concluded that satisfactory fatigue behavior can be obtained by limiting the total shear stress to 5G and that use of the hardest possible rubber is advantageous. This latter conclusion is noteworthy because it appears to contradict the results of some other studies. The compression force-deflection curves for reinforced bearings and plain pads were also found to be nonlinear, and empirical formulas were proposed to describe them.

The report is one of the few to attempt to assess fatigue behavior, the importance of which may be judged from the large difference between static and fatigue failure loads. However, the scatter in the results, the small number of specimens tested, and the differences in their materials, design, construction and test conditions, render some of the conclusions open to question.

Battelle Memorial Institute 1970 (Minor and Egen Report *(9)*)

The researchers report four series of tests. Three concerned laboratory samples (shear modulus and stress relaxation, influence of geometry on compression, reversed cyclic shear), and commercial bearings were tested in the fourth. Most tests were performed on CR and EPDM, with a few NR specimens. Low temperature performance was investigated only for the synthetic rubbers.

The study found that shear modulus is related to hardness, but numerical values differed from those suggested by others. Compressive stress–strain relationships were dependent on shape factor, and were generally nonlinear. They were compared with two theoretical predictions and found to match neither precisely. Creep was found to be larger for harder rubbers, but again values differed somewhat from the results of other studies. The properties of bearings with holes were found to be calculable using the concept of a shape factor, provided the dimensions of the hole were included. However, the researchers did not examine the effect of the hole on the strength or fatigue life of the steel reinforcement in the bearing. Cold-bonded bearings delaminated rather frequently compared to fully vulcanized ones.

The results of this report are useful as far as they go, but many important questions, such as development of analytical expressions to predict behavior, criteria for failure, etc., were left unaddressed. Inconsistencies are also apparent, which complicate the interpretation of the published results.

Italian Code Study 1972 *(76)*

Sanpaolesi and Angotti presented theoretical and experimental results with the intention of recommending changes in the Italian Code. The theory was largely a restatement of Conversy's work, although for some design relationships the theory is abandoned in favor of the empirical form recommended by the ORE Committee D-60 (*33*).

The tests were carried out on 176 specimens. They were obtained by cutting up large sheets of bearing material obtained from four commercial sources. Sizes ranged from 4 in. × 4 in. to 16 in. × 16 in., and hardness, from 55 to 71. Quality control appears to have been moderate at best, because the highest EB (elongation at break) was 370 percent (55 and 61 durometer), hardnesses (averaged over many readings and samples) differed by at least 5 degrees from the nominal value, and more than half the bearings had at least one dimension which lay outside the stated tolerances.

In most tests the bearings were loaded only to working levels. In compression they were found to exhibit nonlinear stress–strain relations that were described by an equation similar to the ORE one but with different coefficients. By comparing bearings with different numbers of identical rubber layers, but the same outer cover, an effective thickness of 1.2 t was obtained for the outer cover layer. (Experimental values lay between 1.0 and 1.4.)

In tests to destruction, a kink was observed in the σ-ϵ plot which is believed to correspond to yielding of the reinforcement. The plates were thin (1 or 2 mm) and the steel stress derived from the applied stress at the change in stiffness (using Eq. 36) fell in the range of material yield stresses given by the manufacturer. No fatigue loads were used, and, apart from the tests to destruction, the loading did not tax the bearings severely. However, the Italian Code (*40*) developed from the study contains a limit on shear stress due to combined compression and rotation of 3G (compared with the UIC 772R code which allows 5G for combined compression, bending, and shear). It also requires provision for a minimum of 0.03 radian rotation to account for poor placing. The way in which these rather stringent design criteria were derived from the experimental results is not clear.

Caltrans Tests (*45*)

California is currently making extensive use of fabric reinforced bearings and thus they performed a study to investigate their behavior. The report describes tests on CR bearings reinforced with steel, fiberglass or polyester. Loadings were elastic compression, creep, compression failure, compression plus shear, and compression plus rotation.

The researchers concluded that polyester was unsuitable for reinforcement because it is too flexible and creeps too much. At service loads, steel and fiberglass bearings behaved similarly, and a single set of compression stress–strain curves was developed. Creep is estimated at about 25 percent of elastic deformation if 800 to 1,000-psi compression is maintained for 25 years. It is recoverable on removing the load. As the load was increased, the steel reinforced bearings showed a change in stiffness when the plates yielded, but carried 50 percent to 100 percent more load before failing. In all cases the bearing failure was caused by rupture of the reinforcement, and the weakest (reinforced with fiberglass) failed at an applied stress of 1,600 psi.

Shear behavior was found to be almost linear with an apparent modulus between 90 psi and 109 psi. (All specimens were 53-54 hardness CR.) Some softening was noticed with increased shear deflection, but increased vertical load increased G_a slightly. Rollover of the corners was noticed at 25 percent and 50 percent for fiberglass and steel-reinforced bearings, respectively. The vertical deflection was monitored during shear tests, but the largest change (0.31 percent of pad thickness) was considered insignificant. However, the vertical and horizontal stiffnesses are so different that the work done by the forces in the two directions is of the same order of magnitude, so the conclusion of insignificance may not be valid.

CORRELATION BETWEEN THEORY AND PRACTICE

General

Besides the comprehensive test series, such as those described previously, many others have been performed which concentrated on investigating one particular aspect of bearing behavior. This section considers both types of studies in an attempt to compare theoretical and experimental findings. Frequent conflicts exist in the evidence, even on rather fundamental points, which sometimes makes it hard to draw clear conclusions. Such apparent conflicts are partly attributable to variations in material properties.

Manufacturers tend to think of material properties as being totally defined by hardness; therefore, bearings are generally ordered that way. However, hardness values vary from one operator to another, the values measured on many of the test specimens differed by at least 5 points from the nominal value. This is exacerbated by the fact that many operators do not follow the standard ASTM procedure. Further, two specimens with the same hardness (a surface measurement) may display significantly different mechanical properties (such as shear modulus), which depend on the nature of the material throughout its bulk. In the light of these variations, scatter in the test results is hardly surprising, but, since similar scatter must be expected in the field, design formulas of excessive precision are not warranted. For similar reasons precise conclusions cannot be drawn from the tests, and the task is made more difficult by variations in test procedure and by incomplete documentation in the reports. However, comparisons were made as well as possible and are presented below according to the type of behavior.

Compression Under Concentric Loading

Compression stiffness is important because excessive bearing deflection may contribute to serviceability failure of the bridge. This is a particular problem with some types of expansion joint and deck systems that are sensitive to relative displacement across the joint. A number of research programs (*3, 9, 33, 45, 75, 76*) have studied the compressive stiffness of bearings. Both du Pont studies, the Caltrans study, and the Minor and Egen report all appeared to focus on the development of compressive load deflection curves for design aids. Their curves show significant stiffening at higher stress levels. The curves by Minor and Egen showed the same general tendency, but to a lesser extent than might otherwise be the case because "settling-in" effects were eliminated by aligning experimental and theoretical curves at 100 psi. These experimental deflections were then compared with the predictions of approximate theory (*48*) used in many design codes (*4*). The comparison was not very favorable because the experimental bearings were generally about twice as stiff as the linear theory. Application of Holownia's nonlinear theory (*55*) gives results that generally fall closer to the experimental ones. In addition, the nonlinear approximation proposed by Gent (*56*) (see section under "Nonlinear Formulations of the Compression Problem in Ch. 4) improves the comparison between the theory and experimental results.

Two other research programs (*33, 76*) used a different approach. They developed empirical equations to predict compressive stiffness based on the experimental results. Both use equations of the same form, but the coefficients are very different. This indicates that the experimental results were not very repeatable and that the studies did not consider all the necessary parameters.

All the experimental programs and design codes investigated use the shape factor and material properties to correlate compressive force with deflection, overlooking the effects of aspect ratio and size. Test results suggest that this simplification is acceptable, but this may be in part because the majority of specimens have similar sizes and aspect ratios. Theory predicts significant differences in behavior. For example, Figure 17 indicates that a long strip with a shape factor of 5 should have a deflection 70 percent larger than that of a square bearing with the same shape factor, material properties, and compressive stress. Neglect of the aspect ratio may contribute some of the scatter observed in experimental results.

Further, there is a definite size effect. Edge cover, the thickness of which is generally independent of bearing size, is likely to affect the stiffness of small bearings but may be negligible in big bearings. The material properties are also likely to vary through a big bearing because of nonuniform heat penetration during vulcanization. Further, the adhesive at the rubber-to-steel bond penetrates the rubber to a fixed depth regardless of rubber layer thickness, and so the influence of the adhesive on mechanical properties will be greater for a thin layer. For the same shape factor, small bearings have thinner layers than big bearings. For these reasons, tests on small, high-shape factor bearings are likely to predict poorly the performance of big ones, and thus the need for proof testing of commercial bearings is greatest for bigger bearings. If the size and shape were considered in the evaluation of experimental results, it is possible that the scatter would be reduced.

Creep

Some of the earliest information on creep was published by du Pont (*3*). Their curves imply that the ratio of creep strain to elastic strain is independent of stress, depending only on time and material hardness. The ratio is higher for harder rubbers, which is attributed to their greater percentage of fillers (*70*). The Battelle results (*9*) show the same tendency, although they also display a lot of scatter. The Caltrans (*45*) tests lasted up to 30 hours, and an extrapolated log plot to 25 years suggests values of about 20 percent to 25 percent for 55-durometer bearings under 600 psi dead load. Their results are in agreement with du Pont's, except that the Caltrans results show the majority of creep happening earlier than do the du Pont tests.

Compressive deflection also appears to increase with time when the load is accompanied by cyclic shear deformation. No theoretical work is known that can predict the magnitude of the effect; but, because the effect is much more severe in plain pads, it seems likely to be related to the lateral restraint of the elastomer (although the opinion has been expressed that a purely material effect, analogus to the behavior of a granular soil, also exists). The Battelle researchers (*9*) found increases in vertical deflection between 14 percent and 73 percent after only a few cycles of shear deformation. Once again higher percentages corresponded to harder material. Ozell and Diniz's results (*34*) showed a similar phenomenon under the guise of stress relaxation. Two pads compressed between plates maintained at a constant distance apart showed final stresses between 40 percent and 80 percent of the original stress after being subjected to cyclic shear load.

The Caltrans study (*45*) investigated the effect of cyclic compressive "live load" (superimposed on constant "dead load"). At the end of the cyclic test, some residual deflection was found, but recovery from it was rapid and, for practical purposes, complete.

Shear Stiffness

Previous discussions pointed out the importance of distinguishing between material stress–strain behavior and bearing-force-deformation behavior when examining shear stiffness. The shear modulus is best determined from large thin sheets, because geometric effects are small. Kelly, in unpublished work, found a linear constitutive relationship up to 100 percent shear strain for 1/4 in. thick sheets with a shape factor of about 8. Data published by MRPRA (*70*) show curves for a particular rubber with no filler, and then 30 and 85 pph of SRF black. The gum rubber curve is linear, while the other two soften initially and later stiffen. The nonlinearity is more pronounced for the higher filler content, but is always small. Others (*77*) show curves with similar tendencies, but, although the diagrams are called stress–strain curves, the specimens' dimensions are not known, so the influence of geometry in such tests remains uncertain. The nonlinearity of the material is thus modest in magnitude and hard to detect experimentally, and so the material shear modulus G is generally taken to be a constant whose value shows reasonable correlation with hardness. As noted earlier, the shear modulus and shear stiffness depend on temperature and strain rate. Shear moduli at low temperatures have been reported for the rubbers used in bridge bearings which are 2 to 3 times that at normal temperatures. However, increases of several hundred-fold have been recorded for CR at −40 F (*33*), presumably caused by second order (glass) transition effects. It should again be emphasized that this increase in stiffness is applied only to incremental deformations.

Force deflection curves in shear have been obtained by many researchers. Typical results (*8, 76, 83*) show curves that soften initially and then become linear, but show considerable hysteresis on unloading (*84*). However, as noted earlier, geometric effects of vertical load have a significant influence on this behavior. Gent (*68*) showed that instability effects reduce the shear stiffness of tall slender bearings, and that the magnitude of the change can be predicted. Also the reduction in bearing height due to vertical load increases $\tan\gamma$ for a given displacement, and bulging increases the effective plan area of the bearing, both of which will increase its apparent shear stiffness. Porter and Meinecke (*18*) incorporated both these effects as corrections and included a horizontal component of the compressive load in the shear force. They found that stress–strain curves so derived were close to linear although the force deflection curves were more nonlinear. However, their results were for circular Butyl rubber specimens with a shape factor of 0.6 and a maximum shear strain of 12 percent, conditions not representative of bridge bearings.

Further difficulties are caused by rolling over at the edges of the bearing as shown in Figure 27. Fiber reinforced pads and those with thin steel shims roll over more readily than those with relatively rigid plates, but the effect starts between 25 percent and 65 percent shear strain in most bearings.

Brielemaier and Hoblitzell (*31*) found an effect that appears to be purely material. They used 60-durometer CR pads and laminated bearings 5 in. × 30 in. × 1/2 in. and found that the material crept after imposition of shear displacement, but their curves (after creep had taken place) show that at 120 F, G is constant and independent of compression. However at 0 F, G reduces with increasing strain and is increased by compression. If the material property G depends on the compressive stress, further complications arise because the state of stress and strain varies widely within the bearing and the strain may be very large.

While there are good theoretical reasons for expecting shear force-deformation behavior which is nonlinear and compression-dependent, correlation between theory and experiment depends on the availability of consistent data reported in full detail. Because this is not available, little choice exists but to assume that shear deformation is linear and independent of compression. Clearly, much useful work could be done in this field.

Rotation

Two researchers have studied effects of rotation or eccentric load. Theory predicts that rotation induces large shear stresses and strains in bearings with medium-to-large shape factors (see Fig. 31), so response to eccentric load is important if the total shear stress or strain is used as the failure criterion. Price (*85*) loaded bearings between platens that were inclined relative to each other, up to a maximum of 5 deg. He concluded that under typical conditions, a 1-in. taper caused a maximum compression stress that was 50 percent greater than the average and that bearings can be subjected to normal loads and 2-deg taper without permanent damage. However his tests were static and did not consider the effect of fatigue.

In the ORE study (*33*) cyclic compression loads were applied at a given eccentricity and in combination with a fixed shear deformation. The values of the stress limits, eccentricity, and shear angle differed for each bearing tested and so meaningful conclusions are once again elusive. However, the hardest (70-durometer bearings) underwent at least 2 million cycles up to at least 1,400 psi with no damage. All the others suffered damage. The plain pads started to break up after about 50,000 cycles to 850 psi. The laminated bearings with no cover suffered extensive debonding during 2 million cycles to the same stress. The NR bearings gave a mixed performance. One suffered only minor abrasion, and the other suffered serious delamination and plate fracture (starting at the dowel holes). It should be noted that this is the only known fatigue test of a bearing with holes in the reinforcement, and it failed early because of plate fracture.

Meaningful correlation with theory is virtually impossible with so little data and such large scatter. Rotation appears to cause some of the most important stresses and yet very few data are available.

Distribution of Stresses in the Rubber and Reinforcement

Experimental determination of localized stress in any material is difficult. No studies are known that have measured internal stresses in rubber bearings, but some writers (*50, 51*) have studied the distribution of surface strains across the loaded face. They used similar approaches, but the latter Ref. 51 was more comprehensive. Circular and square rubber discs bonded to steel face plates were used, and compressive and shear stresses at the interface were measured when the bearing was subject to shear or compressive load. Measurements were made by cutting small holes in the steel plate and inserting an instrumented piston that was in contact with the rubber.

Gent's (*51*) specimens had shape factors between 1.1 and 2.6. In compression they were loaded to about 225 psi and agreement with theory is generally good. The theory outlined in Chapter Four predicts a maximum shear stress at the edge, but the absence of shear stresses on the unloaded faces indicates that there can be no shear stress at the corner of the rubber, indicating a singularity at this point. The singularity was too sudden to detect experimentally except in one specimen which debonded slightly at the edge, causing the shear stress in the region to drop off less sharply and over a longer distance.

A shear displacement applied to a square block caused a shear stress distribution that was roughly uniform. It peaked in the middle and reduced at the edges, maximum and minimum stresses (away from the edge discontinuity) each differing from the average by about 15 percent. Again the edge singularity was not detected. The load caused an average shear strain of about 0.065, and was accompanied by a compressive stress of about 110 psi.

Explicit tests for the distribution of steel stresses have not been performed, but correlation can be achieved in two ways. For an infinite strip, Eq. 30 relates rubber shear stress to steel tension using equilibrium alone, so it must be valid. Thus, for that shape confirmation (*50, 51*) of shear in the rubber also confirms the stresses in the reinforcement. Steel plates in a rectangular bearing of finite length will exhibit shear lag, and the tension across the bearing will be less than that in a strip of the same width. Thus, Eq. 30 can be used safely for rectangular bearings.

Indirect correlation of the stress in the reinforcement may also be obtained from observations of compression stiffness. Sanpaolesi and Angotti (*76*) tested to destruction a number of rectangular bearings with thin plates. Most failed when the reinforcement ruptured, but before that, a marked change in compression stiffness occurred that was attributed to yielding of the plates. Using their range of values for the tested yield strength of the steel, Eq. 35 provides a value of applied stress necessary to cause yield of the steel which correlates well with the kink in the experimental compression curve. Thus, Eq. 35 may be considered useful in the service-load range of typical bearings.

Orthogonal mats of fiberglass have no strength or stiffness in shear and so they can sustain no shear stress. Hence, Eq. 35, which was derived for an infinite strip in which shear lag is absent by virtue of geometry, should also apply to fiber-reinforced rectangular bearings. Applying this equation to typical results of the Caltrans (*45*) tests (in which the compressive stress at failure was always greater than 1,600 psi) results in a calculated strength for the fiberglass of about 1,300 lb/in. per layer. This should be compared with a guaranteed minimum of 800-lb/in. layer.

These results confirm that the approximate theories give reasonably accurate values for strains in the rubber and reinforcement provided that deformations are small. No direct verification is available for stresses in bearings loaded to levels commonly found in practice.

Yield and Rupture of Reinforcement

The post-yield behavior of steel plate reinforcement has not received explicit attention, but some evidence is available. Tests to destruction by ORE (*33*), Sanpaolesi and Angotti (*76*), and others show failure by plate fracture. However, if Eq. 35 is used, the calculated steel stress is roughly 4 times the steel ultimate strength. The cause for the discrepancy is not known but is probably due to effects of large deformation in the rubber, inelastic stress redistribution, strain hardening in the reinforcement, and the effect on the rubber of deformations in the steel (assumed zero under elastic conditions). The large post-yield strength has little practical use in every day design because service loads should not yield the reinforcing. However, occasional heavy loads that stress the plates nearly up to yield might be permitted in the knowledge that a large reserve of static strength exists beyond this point.

In contrast to this apparent reserve of strength is the adverse effect of dowel holes in the plates. They are often present even in bearings that have no dowels, because temporary dowels are inserted to support the plates during vulcanization. However, the available experimental evidence suggests strongly that they should be avoided if possible or the plate should be thickened significantly if holes are present.

In Price's tests (*85*), even quite thick plates suffered permanent bending, apparently initiated by the holes, when loaded heavily in concentric monotonic compression.

In the ORE (*33*) study, the NR bearings each had two 1-3/4-in. diameter holes. Under static load the rubber separated from the plate around the hole and left a residual slip after the load was removed so that part of the surface of the plate could be seen through the hole in the rubber. Under fatigue loading, the stress concentrations around the holes initiated cracks which propagated to cause early failure. The stress distribution in the rubber is also affected. If the hole is plugged using an adhesive that has a shear strength as high as that of the rubber itself, bulging will be restricted at the hole, but the shear stresses cannot transfer to the plate at the hole and the stress distribution will still be different.

It is understood that many highway bridges in New Zealand are being built on rubber bearings with holes containing lead plugs. The intention is that lead creeps enough not to impair the day-to-night expansion and contraction of the bridge, but it dissipates a lot of energy by hysteresis during an earthquake. Further information is being sought but is not yet available.

Instability

The two most important aspects of instability are the elastic buckling load P_{cr} and the reduction in shear stiffness caused by compression. Gent (*68*) outlines a theoretical calculation resulting in Eq. 54 and describes an experimental program. Theory and test results agree closely. The worst agreement was for short columns, and this was attributed to neglecting the axial deformation of the column (which changes the dimensions to be used in the buckling calculations). The taller columns buckle at a load low enough that the axial deformation is genuinely negligible. Experimental buckling loads slightly exceeded predictions in all cases.

Gent also compared theoretical and experimental values of reduced shear stiffness for bearings under combined shear and compression. Again, agreement was good, with the greatest errors at high values of compression (i.e., as P approaches P_{cr}), in which case the test results showed the greater stiffness (because the theory slightly underestimates the true buckling load). The rubber layers were circles and 3:1 and 6:1 rectangles with shape factors between 1.17 and 2.31.

In Minor and Egen's tests, two bonded bearings buckled under compression. This was apparently not anticipated, but Gent's theory predicts the critical load within 10 percent or 15 percent in both cases. Schapery and Skala (*69*) extended the analysis to include curved plates (as used in helicopter bearings), and again found that test values exceeded predicted buckling loads.

Equation 54 requires a knowledge of the bending stiffness of the bearings, which, in turn, relies on an apparent Young's modulus in rotation. Values have been predicted theoretically (Eq. 43), but not confirmed experimentally, even for small displacements. Experiments show that in compression, stiffness varies with load and the same is probably true in rotation. Thus, practical bridge bearings that have higher shape factors than those tested for instability may buckle at loads which are less well predicted by theory.

Perhaps the most serious difficulty associated with instability of rubber bearings is convincing engineers that buckling is possible. An engineer unfamiliar with the reduction in buckling load caused by flexibility in shear is apt to look only at the outside dimensions of the bearing. The bearing is so much stockier than the Euler columns with which he is familiar that he concludes (wrongly) that buckling need not be considered.

Friction

Friction is important for location of bearings and lateral restraint of plain pads. The friction coefficients of greatest interest are those of rubber against steel or concrete, and values have been obtained in many studies. Some researchers (*75, 85*) have concluded that the value of the coefficient depends on the applied stress. While Bakirzis and Lindley (*86*) quote constant values for rubber to steel between 1.0 and 1.75 depending on surface finish, Price (*85*) found about 0.85 at low load, dropping to about 0.4 at about ⅔ design load, extrapolated to 0.33 at full design load. Values against concrete showed a similar tendency with an extrapolated full-load value of about 0.4. The ORE study (*33*) suggested that the coefficient depends on stress but not on the material in contact with the rubber and could be conservatively estimated by

$$\mu \geq 0.1 + \frac{29}{\bar{\sigma}_c}$$

where $\bar{\sigma}_c$ is the compression stress in psi.

Interest in the behavior of automobile tires has prompted a considerable body of research, but it concentrates on sliding friction which is quite different from the static friction of interest here. During the Caltrans tests (*45*), a slip test was made with two bearings (rubber to rubber) on top of each other. The reported data imply that slip occurred at a μ of about 0.06 under a compressive stress of 400 psi.

The scatter among test results suggests that compounding is important. Bakirzis and Lindley found no dependence on hardness for natural rubber specimens 47-70 IRHD, but it seems likely that some of the compounded waxes and oils will affect the surface condition of the rubber. The effect may change over time with their migration to the bearing surface.

Some authorities also believe that friction is less under dynamic loads, so that temperature forces on the substructure are alleviated by the bearing slipping as a result of dynamic traffic loads. No experimental work is known to have confirmed this hypothesis.

Friction is thus seen to be imperfectly understood and the possibility of slipping merits conservative consideration. Laminated bearings need only be fixed to the support if slipping due to shear loads is possible. Plain pads, however, risk additional vertical deflection under compressive load alone if lateral restraint due to friction is inadequate. Many codes allow for the probable extra deflection by using an effective thickness of the pad greater than the true thickness, thereby reducing the shape factor. If a shear load is applied simultaneously, it will increase the shear stress on one side of the bearing and decrease it on the other. Cyclic shear loads thus provide the possibility of increasing vertical deflections as discussed earlier under "Creep" and of allowing the bearings to "walk" out of place.

Bakirzis and Lindley (*86*) show that plain pads subjected to compression will not slip at their edges if $S \leq \mu/2$, in which μ is the coefficient of friction. Their experimental results confirmed this. However, the lower values of μ found by others, combined with the effects of dynamic live load which further reduce them, make some slip almost inevitable for pads of practical shape factors. Additional work is needed in this area. An in-depth study of friction was recently completed in Germany (*87*), but the results could not be included in this report.

Failure Theories

Rubber bearings do not fail in a manner which is as simply defined as, say, the collapse of a beam. Even if the reinforcement ruptures, the material is still present to stop the girder from falling onto its support. Thus, serviceability limits are of greater interest than a collapse load and they are more difficult to pin down.

Material failures, such as yield (or fracture) of reinforcement, significant delamination at the interface, or shearing of the rubber, will lead to permanent deformation or unacceptable changes in stiffness. Instability is clearly to be avoided. Fatigue conditions dominate material failure, but, as yet, no research program has specifically addressed the question of what limits should be set on stress (or strain) to prevent failure. The problem is difficult, involving as it does material fatigue and the precise geometry of the specimen, both of which vary widely. Because of the lack of understanding of fatigue, there is wide variation in practice. Some codes (*5, 15, 40*) suggest that fatigue may be controlled by limiting the total shear stress to numbers such s 5G or 3G. Others (*4, 88*) suggest that shear strains must be limited to a percentage of elongation at break, EB. These are very different conclusions and have considerable impact on the use of NR and CR in bearings. The fatigue problem is further complicated by holes in bearings (*33*), unusual shapes and stress concentrations, since these factors may cause fatigue of the reinforcement rather than fatigue of the rubber. Thus, fatigue must be regarded as one of the major unanswered questions in bearing design.

Even under static loads, conditions for failure in the rubber are not clearly defined. For example, although ultimate tensile strength and elongation at break define conditions at rupture in a standard uniaxial test specimen of given dimensions, it is not clear how these values relate to a rubber layer in a bearing subjected to a multiaxial state of stress. A major difficulty is caused by the fact that the deformations in rubber are large, so the failure theories (e.g., Tresca and von Mises) used with usual civil engineering materials are not applicable.

Bond failure is also a failure mode of some importance. It can be avoided by high quality, clean manufacturing techniques. The consensus of existing opinion is that careful workmanship and good materials produce a bond strength that is higher than the shear strength of the rubber. Instability is also an important mode of failure, but it appears to be reasonably predictable with existing theory. However, dynamic creep reduces stiffness, and it will also reduce the buckling load. This effect is also worthy of consideration, but no existing tests have considered this phenomenon.

Theories of failure and serviceability criteria are important to the design of bearings, but only minimal guidance is available at the present time. The assessment of these criteria is particularly important for load factor and limit state design. Considerable research is needed in this area.

CHAPTER SIX

COMPARISON OF CODES AND SPECIFICATIONS

GENERAL

There are significant differences in the design methods used in different countries. They become evident through discussions with practicing engineers and from sample calculations such as those shown in Chapter Two. This chapter presents the detailed provisions of the major specifications and a discussion of the differences between them, and then compares their various design approaches with theoretical and experimental research results.

PROVISIONS OF INDIVIDUAL SPECIFICATIONS

AASHTO Specifications

The AASHTO Specifications (*1*, *2*) serve as the basis of nearly all the state specifications, the American Railway Bridge Specification (AREA), and other design procedures in the North American continent. It was appently developed in conjunction with some developmental research for Neoprene bearings by the du Pont Company (*3*), and its major provisions are summarized in Tables 2 and 3. It limits the average compressive stress on the bearing to 800 psi due to dead plus live load, and 500 psi due to dead load only. The average compressive strain, $\bar{\epsilon}_c$, must be less than 0.07, but the specification provides no method for computing or estimating it. The usual practice in the United States is to employ design aid curves (*3*) for estimating $\bar{\epsilon}_c$. These curves were derived experimentally for unrestrained CR pads and show average stress–strain curves for different hardnesses and shape factors. The 800-psi limit controls the design in many cases. The reasons for selecting this stress limit (*3*) appear to be related to the permissible bearing stress for the concretes that were used in the 1950's, rather than to the strength of the bearing. The compressive strain limit controls the design only for low shape factors (i.e., $S \leq 5$ for 50 hardness, and $\leq$ 4.0 for 60 hardness).

The shape factor of the *i* th layer, S, of a reinforced bearing is defined in the same way by all specifications. That is,

$$S = \frac{D}{4t} \tag{59}$$

for circular bearings, and

$$S = \frac{LW}{2t(L + W)} = \frac{ab}{t(a + b)} \tag{60}$$

for rectangular bearings (see Fig. 32). The shear strain due to shear deformation is limited to less than 50 percent. That is,

$$\frac{\Delta_s}{T} \leq 0.5 \tag{61}$$

where

$$T = \Sigma t_i \tag{62}$$

in which t_i is the thickness of each rubber layer and T is the effective rubber thickness. However, the wording of the specification is such that Δ_s refers to the total deformation—that is, the change in girder length caused by going from one extreme environmental condition to the other. The specification does not provide a method for estimating the shear forces, but the usual practice is to estimate shear force, H, by

$$H = \frac{G\Delta_s}{T} LW \tag{63}$$

where the shear modulus, G, is taken as a function of hardness from a design aid (*3*). The rotational capacity of the bearing is not considered by the AASHTO Specifications; however, standard practice is to limit the rotation, α, to

$$\alpha \leq \frac{2\Delta_c}{L} \tag{64}$$

which is intended to prevent the development of tensile stresses in the bearing. It does so conservatively, but the

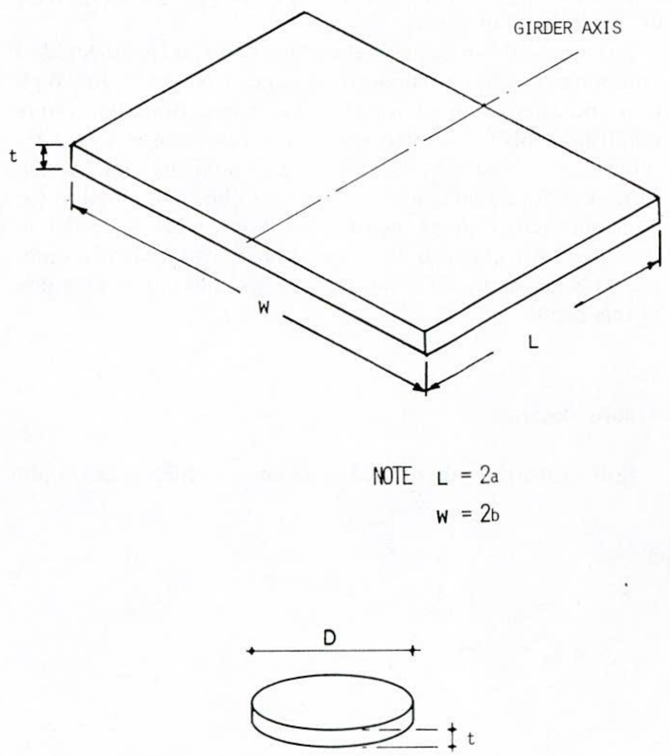


Figure 32. Important dimensions in bearing design.

Table 2. Code comparison for reinforced elastomeric bearings.

	UIC 772R	BS 5400	BE 1/76	AASHTO
ALLOWABLE SHEAR DISPLACEMENT	$\Delta_s \leq 0.7\ T$	$\Delta_s \leq 0.7\ T$	$\Delta_s \leq 0.5\ T$	$\lvert\Delta_s^+\rvert+\lvert\Delta_s^-\rvert \leq 0.5\ T$
ALLOWABLE VERTICAL DISPLACEMENT	$\Delta_c \leq 0.15\ T$	To be specified by engineer	$\Delta_c \leq 0.1\ T$	$\Delta_c \leq 0.07\ T$
VERTICAL DEFLECTION EQUATION	$\Delta_c = \sum \frac{t_i \bar{\sigma}_c}{4GS_i^2 + 3\bar{\sigma}_c}$ $\bar{\sigma}_c = \frac{P}{A_{re}}$	$\Delta_c = \sum \frac{Pt_i}{A_{re}} \left\lvert \frac{1}{5GS_i^2} + \frac{1}{K} \right\rvert$	$\Delta_c = \sum \frac{Pt_i}{A_{re}} \left\lvert \frac{1}{E_c} + \frac{1}{K} \right\rvert$ $E_c = E(1 + 2kS_i^2)$	None
ALLOWABLE ROTATIONAL CAPACITY	$\alpha \leq \frac{6\Delta_c}{L}$	$\alpha \leq \frac{3\Delta_c}{L}$	$\alpha \leq \frac{2\Delta_c}{L}$	No restriction
LIMITING CRITERIA FOR ALLOWABLE LOAD P	TOTAL SHEAR STRESS $\tau_t = \tau_c + \tau_s + \tau_r \leq 5G$ $\tau_c = \frac{1.5P}{S_i\ A_{re}}$; $\tau_s = \frac{\Delta_s}{T} G$ $\tau_r = \frac{GL^2\alpha}{2t_iT}$	TOTAL SHEAR STRAIN $\gamma_t = \gamma_s + \gamma_c + \gamma_r \leq 5$ $\gamma_c = \frac{1.5P}{GA_{re}S_i}$; $\gamma_s = \frac{\Delta_s}{T}$ $\gamma_r = \frac{L^2\alpha}{2t_iT}$	TOTAL SHEAR STRAIN $\gamma_t = \gamma_s + \gamma_c \leq \frac{EB}{3}$ or $\frac{EB}{2}$ and $\gamma_c \leq \frac{EB}{4}$ or $\frac{EB}{3}$ $\gamma_c = 6S\bar{\varepsilon}_c$, $\gamma_s = \Delta_s/T$ $\bar{\varepsilon}_c = \frac{P}{A_{re}E(1+2kS_i^2)}$	AVERAGE COMPRESSIVE STRESS $\bar{\sigma}_c = \frac{P}{A} \leq 800$ psi provided $\Delta_c \leq 0.07\ T$ and $\frac{P_p}{A} \leq 500$ psi
REINFORCEMENT STRENGTH	$(t_s)net \geq \frac{2(t_i+t_{i+1})(P_p+1.5P_d)}{A_{re}F_s}$	$t_s \geq \frac{1.3(t_i+t_{i+1})}{A_{re}F_y}$ $\geq .08"$	$t_s \geq \frac{2(t_i+t_{i+1})P}{A_{re}F_s}$ $\geq \frac{1}{16}"$ (internal plates)	None
STABILITY REQUIREMENTS	$\bar{\sigma}_c \leq \frac{2LGS_i}{3T}$	same as UIC 772	$t_i \leq L/4$ and $T \leq L$	$T \leq L/3$ and $T \leq W/2$
SLIP REQUIREMENTS	$\bar{\sigma}_c \geq 290$ psi $\bar{\sigma}_c \geq \frac{10H}{A_{re}} - 290$ psi	$\bar{\sigma}_c \geq \frac{10H}{A_{re}} - 290$ psi	$\bar{\sigma}_c \geq \frac{3H}{A}$ (concrete) $\geq \frac{4H}{A}$ (steel)	$\bar{\sigma}_c > 200$ psi due to D.L.
NOTES	$A = LW$, $A_{re} = (L - \Delta_s)\ W$ for rectangular bearings without holes			

conservatism may be justified because prevention of tensile stress is essential if good fatigue behavior is to be obtained.

The stability of the bearing is assured by restricting the effective rubber thickness.

$$T < L/3 \tag{65}$$

$$T < W/2 \tag{66}$$

This stability criterion is more generous than most code provisions for a single layer, but it is more restrictive than the usual provisions for the total bearing. The bearing must be secured against horizontal movement if the minimum compressive stress is less than 200 psi. Unsecured bearings stressed to this level need a friction coefficient of at least ¼ to prevent slip under maximum shear strain conditions, and presumbaly this rationale underlies the provision. The specifications permit the use of both plain pads and bearings rein-

Table 3. Code comparisons for plain pads.

	UIC 772R	BS 5400	BE 1/76	AASHTO
ALLOWABLE SHEAR DISPLACEMENT	$\Delta_s \leq 0.7\ T$	$\Delta_s \leq 0.7\ T$	$\Delta_s \leq 0.5\ T$	$\lvert\Delta_s^+\rvert + \lvert\Delta_s^-\rvert \leq 0.5\ T$
ALLOWABLE VERTICAL DISPLACEMENT	$\Delta_c \leq 0.15\ T$	To be specified by engineer	$\Delta_c \leq 0.1\ T$	$\Delta_c \leq 0.07\ T$
VERTICAL DEFLECTION EQUATION	$\Delta_c = \frac{T\bar{\sigma}_c}{10GS + 2\bar{\sigma}_c}$ $\bar{\sigma}_c = \frac{P}{A_{re}}$	$\Delta_c = \frac{PT}{A_{re}} \left\lvert \frac{1}{5GS^2} + \frac{1}{K} \right\rvert$	$\Delta_c = \frac{1.8PT}{A_{re}} \left\lvert \frac{1}{E_c} + \frac{1}{1.8K} \right\rvert$ $E_c = E(1 + 2kS^2)$	None
SHAPE FACTOR	$S = \frac{LW}{2T(L + W)}$	$S = \frac{LW}{3.6T(L + W)}$	$S = \frac{LW}{3.6T(L + W)}$	$S = \frac{LW}{2T(L + W)}$
LIMITING CRITERIA FOR ALLOWABLE LOAD F	$\bar{\sigma}_c < 2GS$	$\bar{\sigma}_c < 2GS$	$\gamma_t = \gamma_s + \gamma_c \leq \frac{EB}{3}$ $\gamma_s = \frac{\Delta_s}{T},\ \gamma_c = 6S\bar{\varepsilon}_c$ $\bar{\varepsilon}_c = \frac{1.8P}{E(1 + 2kS^2)A}$	$\bar{\sigma}_c = \frac{P}{A} \leq 800$ psi provided $\Delta_c \leq 0.7\ T$ and $\frac{P_p}{A} \leq 500$ psi
STABILITY REQUIREMENTS	$T \leq L/5$ $\leq W/5$	$T \leq L$ $\leq W/5$	$T \leq L/4$ $\leq W/4$	$T \leq L/5$ $\leq W/5$
SLIP REQUIREMENT	$\bar{\sigma}_c \geq 145\ (1 + \frac{L}{W})$ psi	$\bar{\sigma}_c \geq 145\ (1 + \frac{L}{W})$ psi	$\bar{\sigma}_c > \frac{3H}{A}$ (concrete) $> \frac{4H}{A}$ (steel)	$\bar{\sigma}_c > 200$ psi due to D.L.
NOTES	$A = LW$, $A_{re} = (L - \Delta_s)\ W$ for rectangular bearings without holes.			

forced with steel shims or fabric, but the allowable stress is the same for all types of bearings. Further, the strength of the reinforcement is not related to the dimensions of the bearing.

Both NR and CR compounds with hardness in the range 50 to 70 are permitted by AASHTO, but 70 hardness is not permitted for laminated bearings. In addition, the material must satisfy a minimum elongation requirement, a low temperature test, a heat resistance test, a compression set requirement, an ozone test, and an adhesion test between the rubber and the laminate. No material properties (e.g., E, G, ν, and K) are provided in the specifications. A minimum ⅛-in. edge cover is required to protect bearings reinforced with steel shims, and typical manufacturing tolerances are summarized in Table 4.

Plain pads are designed in the same way as reinforced bearings. The allowable stresses, deformations, computed deflections, and the shape factor definition are identical to that for a layer of a laminated bearing of the same thickness. This does not recognize the slip (*5*) and changed bulging patterns that must occur at the interface of unreinforced bearings with bridge piers or girders. The stability check for unreinforced bearings is

$$T \leq L/5 \tag{67}$$

and

$$T \leq W/5 \tag{68}$$

This check is somewhat more severe than the check for reinforced bearings, and is typical of that used in other codes.

The advantages of the AASHTO Specifications are clear. They are easy to use and generally conservative, and bearings which satisfy them have seldom suffered distress. How-

Table 4. Minima and tolerances from different specifications.

ITEM/CODE	UIC 772 R	BS 5400	BE 1/76	EXISTING AASHTO	UNITS
Plan dimensions					
L, W, ≤ 36" L, W, ≥ 36"	-0 + 0.20	-0.12 + 0.24	N/A	-0 + 0.25 -0 + 0.5	in in
Total bearing thickness					
0.39 ≤ h < 1.18 in 1.18 ≤ h < 1.97 in 1.97 ≤ h < 3.15 in 3.15 ≤ h < 4.72 in	± 0.02 ± 0.03 ± 0.035 ± 0.043	±.04 (h ≤ 0.79) ± 5% (h ≥ 0.79)	N/A	-0 + 0.125 -0 + 0.25	in
Individual layer thickness	± 0.02	N/A	N/A	± 0.125	in
Edge cover	N/A	0.18	N/A	0.125" $^{-0}_{+0.125}$	in
Out of parallel (top & bottom)	N/A	0.2%	N/A	0.125 in	% or in
Elongation at break (NR/CR)					
50 ± 5 IRHD 60 ± 5 IRHD 70 ± 5 IRHD	as laid down by individual railroad	450/400 400/350 300/300	N/A	450/400 400/350 300/300	%
Tensile strength	"	2200	N/A	2500/2500	psi
Steel plate thickness	.04 or load dependent	.08 mm or load dependent	0.12 (outer) 0.06 (inner)	N/A	in

ever, they have several major disadvantages. The code makes no statement as to the rotational capacity of the bearings or the shear force transmitted by them. It specifies reinforcement strength for fiberglass and the thickness for steel laminates without considering the geometry of the bearing. No effort is made to account for combined stress. It limits the compressive stress to 800 psi, but this limit does not appear to be dictated by the strength of the bearing. No distinction is made between plain pads and reinforced bearings for the purpose of computing the shape factor. With the same definition for both, the stiffness of a plain pad will be overestimated if it is based on a design aid generated for reinforced bearings, because the slip at the loaded interface is neglected. Further, the code makes no distinction between fiber reinforcement and steel shims, which appears to be inconsistent with existing research results. Finally, the code provides no help with attachment details or inadvertent eccentricity, and it generally lacks the versatility displayed by some other codes.

British Specification BE 1/76

The British Specification BE 1/76 (*4*) is probably the most widely used elastomeric bearing specification in the world. It has served as the basis of bearing design in Great Britain, Australia, and parts of Europe and Asia. It is based on a design method (*88*) which was developed in response to the early research in reinforced bearings. It recognizes that shear strains, γ_s and γ_c, are induced in reinforced bearings by compression and shear deformation as shown in Figure 33. These strains are then limited by

$$\gamma_s = \frac{\Delta_s}{T} \leq 0.5 \tag{69}$$

and

$$\gamma_c + \gamma_s \leq \eta\ EB \tag{70}$$

where EB is the elongation at break of the rubber, and η is a ratio (⅓, or ½) which is a function of loading type. Δ_s is the horizontal deflection between the bridge setting position and the extreme environmental condition. If the bearing is simultaneously sheared in two perpendicular directions, γ_s is limited by

$$\frac{\Delta_{SL}}{T} + \frac{\Delta_{SW}}{T} \leq 0.5 \tag{71}$$

The total shear strain is used in Eq. 70 because it is considered the most consistent measure of potential fatigue failure due to a combination of different loadings, and it also is a measure of the potential for delamination of the bearing. The limitation of the total shear strain to a percentage of the elongation at break is based on an early fatigue study (*25*). The major conclusion of the study was that fatigue life of strain crystallizing rubbers is dramatically improved if neither strain reversal nor complete relaxation (to zero strain) is permitted at any time during the load history. The evidence in the study to support the idea that fatigue resistance increases with EB is less conclusive. However, many engineers knowledgeable in rubber believe it is true, and while test results on full-scale bearings generally support it when EB is small (i.e., less than approximately 300 percent) contradictory evidence also exists. Thus, the limitation must be questioned, particularly at very large elongations at break.

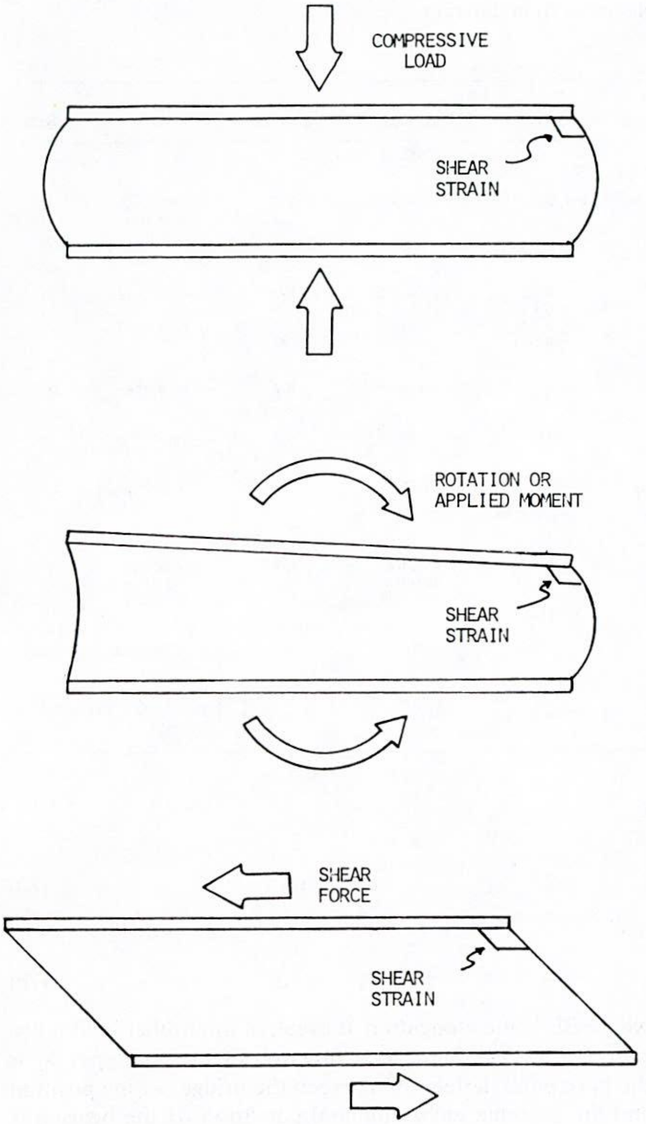


Figure 33. Shear strains in deformed bearings.

The shear strains in the bearing are illustrated in Figure 33. The shear strain due to shear deformation, γ_s, is computed as

$$\gamma_s = \frac{\Delta_s}{T} \tag{72}$$

where Δ_s is the maximum movement caused by the combined effects of creep, shrinkage, wind load, braking loads, and thermal effects. The shear strain due to compression, γ_c, is to be calculated as

$$\gamma_c = 6S\bar{\epsilon}_c \tag{73}$$

where

$$\bar{\epsilon}_c = \frac{P}{A_{re}E_c} = \frac{P}{A_{re}}\left[\frac{\beta}{E(1+2kS^2)}\right] \tag{74}$$

A_{re} is the reduced effective area of bearing (see Fig. 34 and discussion below), P is the compressive load, E is Young's modulus, $\bar{\epsilon}_c$ is the apparent compressive modulus of the bearing, k is a material constant that is defined as a function of hardness of the rubber, and β is the ratio of effective to true rubber layer thickness (to be used for cover layers of reinforced bearings or for plain pads). The specification gives values of material properties in terms of rubber hardness, to be used in the absence of better information. The shape factor, S, for reinforced bearings is defined by Eqs. 59 and 60 as for all other specifications. No strain due to rotation or load eccentricity is considered in the strain check of Eq. 70. This appears to be a definite deficiency because many engineers are of the opinion that the rotational strain is the most significant, and this would appear to be substantiated by the theoretical comparisons of Chapter Four. However, the rotations in the two principal directions, α_1 and α_2, are limited by

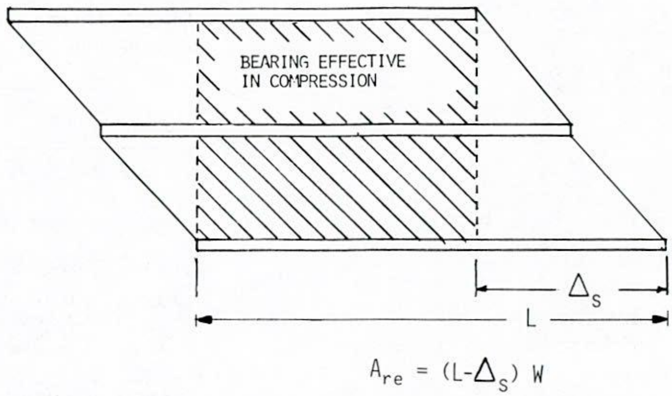


Figure 34. Reduced effective area for bearings loaded with combined shear and compression.

$$\frac{\alpha_1 L}{2} + \frac{\alpha_2 W}{2} \leq \Delta_c \tag{75}$$

This is a conservative method of preventing tensile stress in the bearing since it ensures that no point on the bearing displays a net upward movement. If rotations are restricted to one direction, it coincides with the restriction normally applied in U.S. practice.

The stability of the bearing is maintained by restricting

$$t_i \leq \frac{L}{4} \tag{76}$$

for each layer, and

$$T \leq L \tag{77}$$

for the total bearing. This specification does not explicitly permit fabric reinforcement. The thickness of the steel shims, t_s, is defined by

$$t_s \geq \frac{2(t_i + t_{i+1})}{\sigma_{s,all}} \frac{P}{A_{re}} \tag{78}$$

where A_{re} is the reduced effective area, t_i and t_{i+1} are the elastomer layer thickness on each side of the plate, and $\sigma_{s,all}$ is the allowable tensile stress in the plate. No allowance is made for holes in the plates despite the obvious introduction of a stress concentration and the removal of metal from the most highly stressed area of the plate.

The design procedure relies on approximate small deflection theory for the design and analysis of the bearings. Combined loadings are accounted for by superposition. However, the actual bearing is subjected to large finite strains which invalidate the superposition principle and introduce additional secondary bending and shear stresses due to the P-Δ effect. These are approximately accounted for by using the reduced effective area (see Fig. 34) in place of the effective area, which has the effect of increasing the computed compressive stress for a given load, and so reducing the compression capacity of the bearing under combined stress. The use of the reduced effective area shown in Figure 34 is at best an approximate analytical technique, but at least it approaches the physics of the problem in a manner that intuitively seems appropriate. BE 1/76 employs it in the calculation of γ_c, the shear strain due to compression, and the plate thickness, t_s.

The relative vertical deflection between the two sides of a joint in a bridge deck is limited to avoid damage to certain expansion joints that can tolerate only small movements. The limits are particularly tight because in Britain a continuous asphalt topping is often laid over a joint in the concrete deck. Elastomeric bearings are sometimes used to restrain bridges elastically (i.e., no point on the bridge is fixed rigidly to the substructure) and thus the specification has provisions for analyzing the bearing for applied shear loads (i.e., wind, vehicle braking, etc.) in addition to applied deformations. Attachment of the bearing to the bridge foundation and/or girder is required if the ratio of the minimum compressive load on bearing to the maximum shear force falls below a value B, which varies from 3 to 6 depending on the contact surface. Minimal guidance as to methods for attaching the bearings is provided, but welding of bearing to steel girders is expressly prohibited.

This design method was primarily developed for reinforced bearings of NR; however, it is widely used for both reinforced bearings and plain pads made from CR or NR. Plain pads are designed by the same method as reinforced bearings except that the shape factor is reduced by a factor of 1.8, i.e.,

$$S = \frac{D}{7.2t} \tag{79a}$$

for circular pads and

$$S = \frac{LW}{3.6t(L+W)} \tag{79b}$$

for rectangular ones. This results in significant increases in the strains and deflections and in reduced allowable loads for unreinforced pads. This is consistent with research results, which show that friction alone cannot be relied upon to provide full lateral restraint at the contact surfaces of the rubber.

BE 1/76 is probably the most consistent design procedure in use today. Its provisions usually are supported by research results, and sufficient information is provided to perform a complete design without resorting to other sources of information. Its treatment of plain pads is consistent with existing knowledge, and it attempts to account for combined stress. However, it has defects. The design equations are valid only for small strains. Thus, it neglects nonlinear effects and it does not accurately combine stresses and strains. It does not consider eccentric load or shear strain due to rotation. It limits the combined equivalent shear strain to a percentage of elongation at break. This is probably a reasonable limitation if EB is reasonably small (300 percent or less), but for higher values of EB it appears to be questionable. Finally, it makes no corrections for the increased stresses in the reinforcement caused by holes or unusual shapes.

International Railway (UIC 772R) Specification

The UIC 772R Specification is a railway bridge bearing specification developed by the International Union of Railways (Union Internationale des Chemins de Fer) after an extensive testing program (*5, 33*) on full-size bearings. The specification serves as the basis of a number of highway and railway bridge codes in Europe and other parts of the world. Its primary limitation is expressed in terms of shear stresses rather than strains.

$$\tau_c + \tau_s + \tau_r \le 5G \tag{80}$$

where τ_c, τ_s, and τ_r are the shear stresses due to compression, shear deformation and rotation respectively, and G is the shear modulus under static loading. The $5G$ limitation on shear stress was chosen because the testing program indicated that no fatigue failures occurred when stress levels were maintained within this limit. It should be noted that this limit differs significantly from BE 1/76, because it is independent of the elongation at break of the material and it includes the shear stresses due to rotation. The testing program also indicated that the shear modulus under dynamic loading was approximately twice the value obtained under static loading. In addition, it should be noted that some designers limit compression stress to 100 kg/cm² (1450 psi) even though this is not specified in the code.

The shear stresses for Eq. 80 are therefore computed by

$$\tau_c = \frac{1.5\,(P_p + 1.5\,P_d)}{S\,A_{re}} \tag{81}$$

$$\tau_s = 2G\frac{\Delta_{sd}}{T} + G\frac{\Delta_{sp}}{T} \tag{82}$$

$$\tau_r = \frac{GL^2}{2t_iT}(\tan\alpha_p + 1.5\tan\alpha_d) \tag{83}$$

Subscripts p and d indicate permanent (static) and dynamic loads. P is the compressive load on the bearing, Δ_s is shear deformation, and α is the rotation. The static deflection Δ_s is defined between bridge setting and the extreme environmental condition. S is the shape factor for the thickest layer of rubber and is defined by Eq. 60. The UIC 772R Specification does not recognize circular bearings and thus Eq. 59 does not apply.

There are some significant differences between the procedure outlined above and that of BE 1/76. First, the limits are expressed in terms of shear stress, not strain. In the absence of dynamic effects Eq. 80 could be divided through by G, and the total equivalent shear strain would then be limited to 500 percent (regardless of the material properties). However, the increased stiffness under dynamic load can be accommodated more readily by using the stress formulation and multiplying the dynamic components of the load by suitable factors, as is done in Eqs. 81 through 83. UIC 772R is more realistic than BE 1/76 in that it considers the shear stress and shear strain induced by rotation or eccentric loading as shown in Figure 33. This component is very significant for bearings of medium to large shape factor but it is neglected by most specifications. The maximum rotation α permitted by UIC 772R is

$$\alpha \doteq \tan\alpha_p + 1.5\tan\alpha_d \qquad (84)$$

$$\leq \frac{6\Delta_c}{L}$$

This maximum rotation is approximately 3 times as large as that specified by BE 1/76 and used in standard practice in the United States. It is based on the prevention of strain reversal at the most lightly loaded edge rather than net upward movement in the bearing. The apparent compressive strain, $\bar{\epsilon}_c$, for any layer is computed by

$$\bar{\epsilon}_c = \frac{\bar{\sigma}_c}{4GS^2 + 3\bar{\sigma}_c} \qquad (85)$$

where

$$\bar{\sigma}_c = \frac{P_p + 1.5\,P_d}{A_{re}} \qquad (86)$$

This compressive strain relationship is based on theory modified by empirical observations made during the experimental research (*33*). The shear strains due to shear deformation are limited by

$$\gamma_s \leq 0.7 \qquad (87)$$

This is 40 percent larger than that permitted by most other specifications. Vertical deflection due to compressive load, Δ_c, is limited to $0.15T$. This is approximately twice the value allowed by AASHTO, but it would not appear to be a controlling factor in the design of bearings of usual proportions.

UIC 772R assures stability by

$$\bar{\sigma}_c \leq \frac{2LGS_i}{3T} \qquad (88)$$

where S_i is the shape factor of the thickest layer. The steel plate thickness is controlled by

$$t_s \geq \frac{2(t_i + t_{i+1})\,(P_p + 1.5P_d)}{A_{re}\,\sigma_{s,\,all}} \qquad (89)$$

This equation is similar to the method used in BE 1/76 except that dynamic stresses are amplified; however, UIC 772R requires that net section be considered if the plates have holes. Slip restraint is required if the average stress produced by the dead load acting on the effective area is less than approximately 280 psi or if the horizontal force exceeds minimum friction factor requirements. The coefficient of friction is given by

$$\mu \geq 0.1 + \frac{29}{\bar{\sigma}_c} \qquad (90)$$

where $\bar{\sigma}_c$ is in psi.

Plain pads are designed differently. The shape factor calculation is identical to the single layer of a reinforced bearing, but the vertical deflection under compression load, Δ_c, is

$$\Delta_c = \frac{T\bar{\sigma}_c}{10GS + 2\bar{\sigma}_c} \qquad (91)$$

This equation produces larger deflections than Eq. 85 for nearly all bearings because of the slip between the bearing and the load surface. Because of their empirical nature both formulas should be used with caution outside the normal range of stress and bearing geometry. The average compressive stress is limited to

$$\bar{\sigma}_c \leq 2GS \qquad (92)$$

and rotations are restricted to one-third those permitted by Eq. 84. Stability requires that

$$T \leq \frac{L}{5} \qquad (93)$$

The specification permits the use of both synthetic and natural rubber in the hardness range 50 to 70. Material properties are specified as a function of hardness.

The UIC 772R Specification has several strengths when compared to other design procedures. It incorporates the shear stress and strain due to rotation, τ_r and γ_r, which are absent from many other specifications. It makes use of the reduced effective area concept to account for combined stress with finite strain, and it appears to have the most rational procedure for selecting steel plate thickness. There are also a number of uncertain features and disadvantages. The specification is based on a research program with full-size bearings. This could be an advantage, but unfortunately the research reports and the specification are not well documented and contain several errors and inconsistencies. The specification further relies on a few empirical conclusions, but one of these conclusions (Eq. 85) was apparently not repeatable in a later study (*76*). Using the same allowable stress for all materials conflicts with some research on fatigue and other specifications (*4*), which generally show that higher elongation at break implies better fatigue resistance, particularly at lower *EB* levels. (However, the relationship is not as clearly defined as BE 1/76 would suggest.) The maximum allowable shear deformations and rotations are larger than permitted by most codes, but the larger numbers are not well justified. Finally, it must be noted that UIC 772R was developed for railway bridges which have different loadings and deflection characteristics than highway bridges. Thus, the demands on the bearing will be significantly different, and a few of the provisions may not be appropriate for highway bridges.

German Design Procedures

The German design procedures (*89, 90*) follow a completely different approach from those of other countries, but apparently they are effective. They do not rely on extensive design calculation. They assure satisfactory bearings through the use of standardized shapes and sizes, tight certification procedures for bearing manufacturers, specification of a single elastomer and compounding process, and a rigorous quality control check during the manufacturing process.

Elastomeric bridge bearings may be built only of CR of a specified compound. Only specific dimensions and layer thicknesses are allowed. These dimensions produce bearings with somewhat larger shape factors than many other procedures. Although the concept of shape factor is not used, geometric restraints mean that the minimum is 5 with the majority of sizes lying between 7 and 13. The allowable compressive stress in the bearings is limited to 100, 125, or 150 kg/cm^2 (1420, 1775, and 2130 psi) depending on the dimensions of the bearing, and a 50 percent stress increase is permitted if the shear deformations are limited (i.e., max of 225 kg/cm^2 = 3250 psi). The largest bearings, for which the allowable stress is 150 kg/cm^2, all have shape factors in the range 9 to 13, which is higher than the values commonly used in the United States. Shear deformations are limited so that γ_s is less than 0.6 or 0.7, and rotations are limited to a specific numerical value per layer for each geometry. Stability is assured by limiting the rubber thickness to a proportion ($\frac{1}{5}$ or $\frac{1}{3.33}$) of the minimum bearing dimension. The basis for the selection of these limits is not known, but discussions would suggest that they were based on the research (*33*) for the International Railway Specification. No check of combined stress or strain is required.

The above design procedure permits the selection of a bearing size, but it is not the key to German practice. In fact, it is not even codified into a design specification. Individual bearing manufacturers may not produce bridge bearings until they have been certified. To become certified, they must submit a compound recipe and manufacture a series of bearings according to it for extensive testing. The recipe is regarded as proprietary information as in most other countries. Thus, it is secured by a trustee, and it is used only to check the chemical content of the manufacturer's bearings from time to time. Extensive tests lasting a year or more are performed on these bearings before the manufacturer may be certified. He must enter into a separate contract with an independent quality control supervisory agency, and only then may he begin producing bearings for the specific compound and bearing sizes listed. Periodic recertification tests may be required, and retesting will be needed if the manufacturer wants to change his elastomer compound or manufacturing procedures. At present four manufacturers are certified, but they all use exactly the same compound.

The design procedure would appear to be very easy for the bridge engineer to use, and it has apparently produced very satisfactory results for many years (*90*). It permits relatively high stress levels with large shear deformations and rotations. Further, it is rational in that it tightly controls the manufacturing process that is apparently the source of most problems in actual practice. This approach is taken because of a belief that the mechanical behavior of elastomers is too complex to be treated reliably by analysis. However, it is a very restrictive, rigid procedure, and it does not appear to recognize the very good performance attained by other codes with looser quality control standards and more explicit analytical design procedures. The certification procedure is expensive to the state and the manufacturer, and it would probably eliminate most manufacturers, particularly smaller ones, if directly applied in the United States because of the cost. It is not very versatile, and would appear to share many of the present deficiencies of the AASHTO Specifications in accommodating new technology.

Other Specifications

The previous sections have presented detailed discussions of individual design codes. The discussion has emphasized the design procedures and factors that influence size and shape of the bearings rather than the material requirements and manufacturing tolerances, because this was believed to be the problem of primary importance in this study. A summary of these design requirements for different specifications is provided in Tables 2 and 3. Table 4 includes a brief summary of manufacturing tolerances. The above specifications were chosen for discussion because they covered the wide range of design procedures used in world practice, and they formed the basis for many other codes and specifications. A number of other specifications will be discussed in this section, but they are covered in less detail because they are similar to one or more of the previous specifications.

The Australian Standard 1523-1976 (*91*) is the basis for elastomeric bearing design in the Australian continent. It is nearly identical to BE 1/76 with a few major exceptions. First, the elastomer is restricted to NR. Other materials may be used, but only after considerable testing; therefore, in practice NR enjoys almost exclusive use, providing an interesting contrast with the situation in Germany. The minimum elongation at break permitted by this specification is 575 percent, which is attainable, but it is considerably larger than the approximately 400 percent typically used in other design codes. This greater elongation produces larger allowable loads and smaller bearings because of the strain limits of Eq. 70, but, as noted earlier, it is not clear that this criterion is totally justified at these larger elongation levels. The specification also encourages the use of standard bearings and contains extensive design tables. Finally, the reduced effective area concept is used with large shear deformations (as depicted in Fig. 34) but with the additional provision that the shear must be limited so that the reduced effective area is never less than 80 percent of the original area.

British Standard BS 5400 (*15*) is also worthy of discussion. This is a complete bridge specification, now in final draft form, of which Part 9 governs elastomeric bridge bearings. When accepted it will become the new design procedure for Great Britain, and in its present form it suggests a significant departure from the BE 1/76 toward the UIC 772R design procedures. In many ways, the BS 5400 combines some of the best features of both approaches. It retains from BE 1/76 the compressive strain calculation for all bearings and the shape factor calculations for plain pads. However, it includes the shear strain due to rotation (see Fig. 33), limits the total shear strain to a constant (5.0), and increases the maximum permissible shear strain due to shear deformation to 0.7 as in UIC 772R. The distinction between dynamic and quasi-static loads is carried over to BS 5400, and the vertical deflection calculation appears to be a combination of both methods. The maximum permissible rotation is approximately 50 percent larger than Eq. 75. Several of the changes appear to be conservative and well justified, but the bases for others are less clear. However, in the majority of cases, the designs produced should fall between BE 1/76 and UIC 772R.

The Italian Specification CNR-UNI 10018-72 (*40*) also is a recent design code that incorporates most of the design philosophy of UIC 772R. It is based on the test results (*76*) of 176 commercial bearings and pads. It is generally more conservative than UIC 772R in that it limits shear stresses to

$$\tau_s \leq 0.5G \tag{94}$$

$$\tau_c + \tau_r \leq 3G \tag{95}$$

The basic equations for stress calculations are the same as those for UIC 772R except that the reduced effective area concept is not employed. The average compressive strain, $\bar{\epsilon}_c$, and the deflection under compressive load, Δ_c, have the same nonlinear form as those in UIC 772R, but the coefficients in them are different. Plain pads and external layers of reinforced bearings are also treated more conservatively than in UIC 772R because of the deflections observed in the study. Stability checks and maximum permissible rotations are essentially the same. Thus, while the CNR-UNI Specification is similar to UIC 772R, it is expected to be more conservative for plain pads and for reinforced bearings with small rotations and shear deformations, but the degree of conservatism is less with bearings fully loaded with shear or rotation.

The French highway bridge specification SETRA (*92*) is also similar to UIC 772R. The international prestress organization Freyssinet (*93*) manufactures bearings and has its own design procedure. This is not a specification in that it is legalized by a government agency, but is widely recognized because the organization is active in many countries. The design procedure is very similar to that of BE 1/76.

The Ontario Ministry of Transportation and Communication recently has rewritten its bridge design code (*94*), including a section on elastomeric bearings. It is quite similar to the AASHTO specification. Average compressive stress under service loads is limited to approximately 1,000 psi (1,400 psi at limit loads) and the limits on shear deformation, rotation, and stability are similar. The average compressive strain is limited to 0.04 when measured by a test procedure which increases the compressive stress from approximately 200 psi to 1,000 psi. This code contains different requirements for plain pads and reinforced bearings. Quality assurance is covered by a formal approval procedure, although this is less rigorous than the German requirements. The Ontario Specification differs significantly from AASHTO in that it relies totally on load factor design. This technique is being increasingly emphasized by most specifications for the design of bridges themselves, but bearings appear to represent some special problems and only the Ontario Design Code has made a significant effort in this area. Load Factor Design requires a knowledge not only of the potential modes of failure, the geometric and material properties which govern them, and the applied loads, but also of the statistical variations in these quantities. Since, in elastomeric bearings, even the critical modes of failure are ill-defined and conclusive quantitative information is still less readily available, it is not surprising that rational development of load factor design has been inhibited.

COMPARISON OF DESIGN CALCULATIONS

Considerable differences were noted between the specifications discussed in this chapter. The rationale behind some of the differences was discussed, but it is not always clear how these differences affect the resulting bearing design. The influence of individual code provisions is discussed in this section and illustrated by calculating the capacity of a spectrum of bearings according to the rules in different specifications and plotting the results in dimensionless form (Figs. 35 through 45).

The AASHTO, BE 1/76, UIC 772R, and BS 5400 Specifications are used. BS 5400 was chosen because it represents a major transition from BE 1/76 toward the international railway design procedure, and the others were selected because they typify the range of design logic. The Australian code AS 1523-1976 is nearly identical to BE 1/76 and so is not shown. The calculations were all made for a rubber of 55 hardness and shear modulus of approximately 110 psi, because this is typical of the material most widely used in bridge bearings. The calculations assume that all loads are quasi-static, and they are made for rectangular bearings with no edge cover. Dynamic loads would reduce the load capacity of the UIC 772R and BS 5400 results but have no influence on the others. Edge cover has little impact on the allowable load on the bearing. Bearings with the same shape factor but of different proportions will give different results (particularly rotational capacity), but the general conclusions noted in following discussions remain the same. Unless noted otherwise, the elongation at break of the rubber is taken as 575 percent. This value is higher than demanded by most specifications, but it is obtainable in practice with a 55 hardness and it is the minimum permitted by one major specification. Later calculations will show the significance of this factor on the design results.

Figure 35(a) compares the allowable compressive stress on the total area versus the number of layers for a square reinforced bearing of shape factor 4 with no rotation and γ_s values of 0.0, 0.5, and 0.7. The AASHTO design is controlled by the 0.07 compressive strain limitation (the stress to cause it was obtained from Ref. 9), and stability criteria prevent the use of bearings with more than 5 layers. The allowable stress level is not influenced by shear strain. BE 1/76 permits larger allowable compressive stresses with no shear deformation (i.e., $\gamma_s = 0.0$), but these are reduced to values similar to the AASHTO level when a large shear deformation is applied. Stability does not control BE 1/76 until 15 layers are stacked. UIC 772R is the most generous specification if only a few layers are used, but stability controls the design as layers are added. It reduces allowable stresses well below BE 1/76 when many layers are stacked.

Figures 35(b) and 35(c) show comparable curves for square reinforced bearings of shape factor 8 and 12, respectively. The AASHTO specification is consistently conservative, and allowable stresses are as much as 400 percent larger in UIC 772R and 150 percent larger in BE 1/76. UIC 772R and BS 5400 produce identical results, and when only a few layers are used they permit the largest allowable stresses. Allowable compressive stresses of approximately 4,500 psi can be obtained with a shape factor of 12 and no rotation or shear deformation. The stability condition reduces this allowable stress for bearings with many layers and may cause allowable stresses to drop below the BE 1/76 stress level in some cases. Note that load factors and dynamic loads would further reduce the allowable load capacity for BS 5400 and UIC 772R, but they would have no impact on AASHTO or BE 1/76.

Rotation of the girder also influences the load capacity of the bearing because it induces shear strains, γ_r, as shown in Figure 33. Its influence is demonstrated in Figures 36(a–c). These figures consider the same parameters as Figures 35(a–c), except that the bearing is subjected to its maximum permissible rotation rather than zero rotation. Note that the

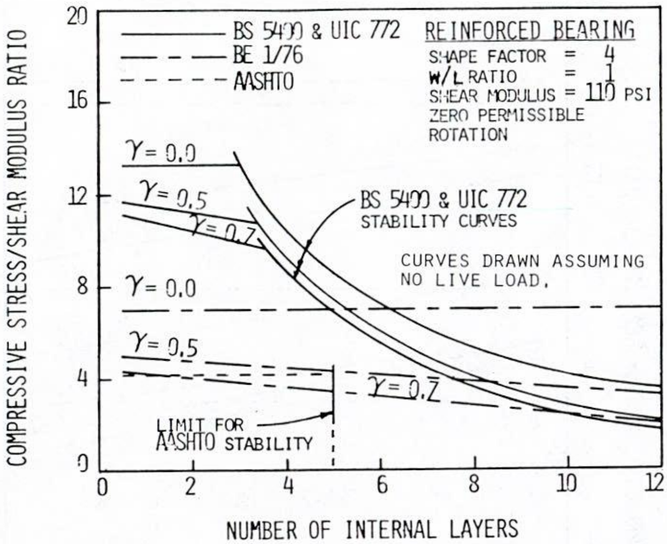


Figure 35(a). Comparison of design codes for reinforced bearings with no rotation—shape factor 4.

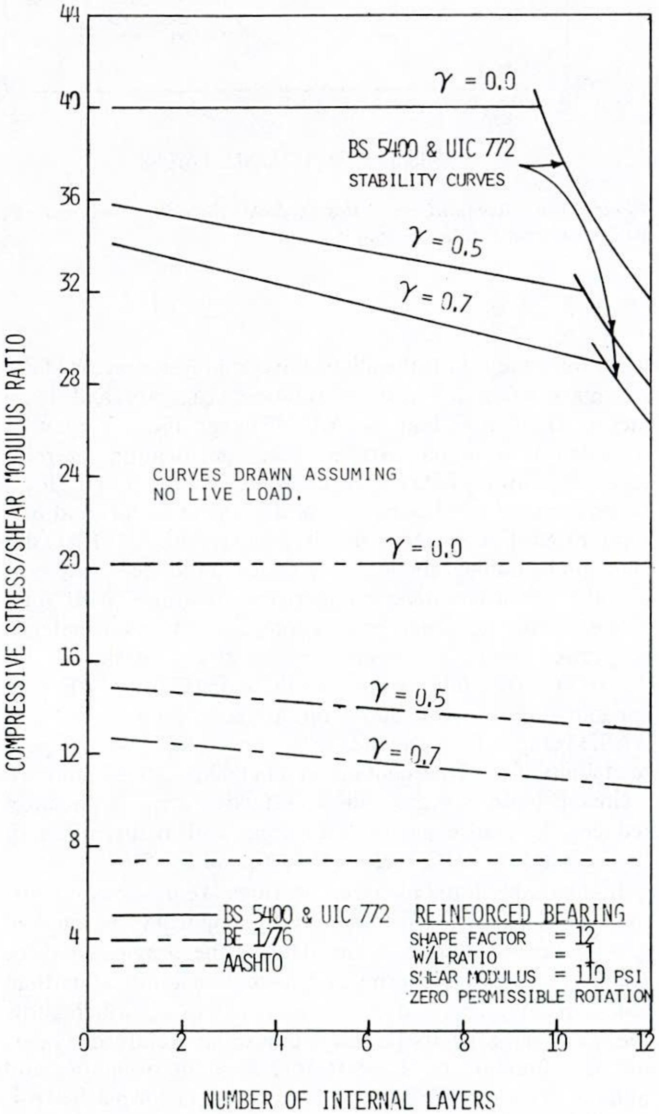


Figure 35(c). Comparison of design codes for reinforced bearings with no rotation—shape factor 12.

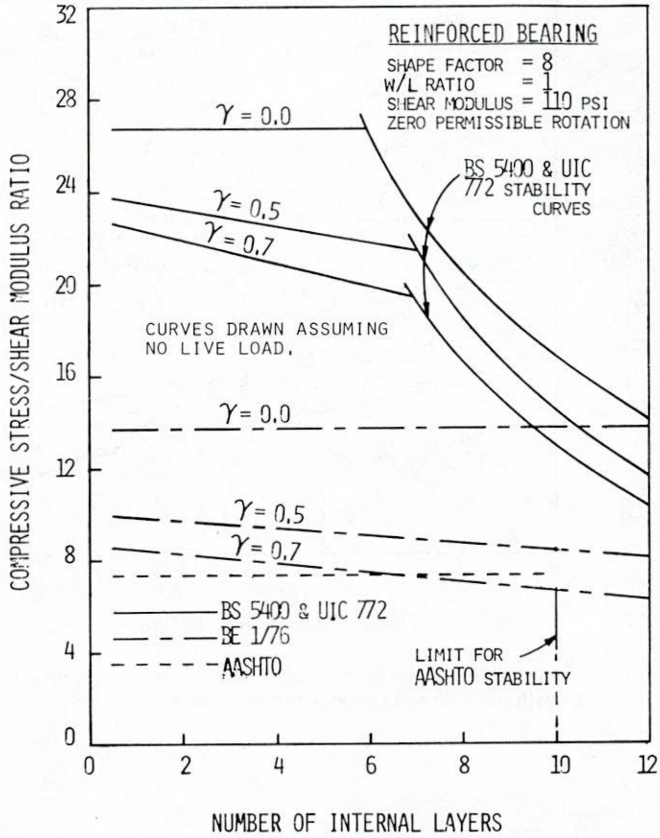


Figure 35(b). Comparison of design codes for reinforced bearings with no rotation—shape factor 8.

maximum permissible rotation is approximately 200 percent larger for UIC 772R and 50 percent larger for BS 5400 than AASHTO or BE 1/76. Further, AASHTO and BE 1/76 load capacities are independent of rotation and thus are the same in both sets or curves. However, BS 5400 and UIC 772R reduce allowable compression significantly. The reduction is greater for the railway specification because its permissible rotation is larger. Under these conditions BE 1/76 clearly is the most generous specification, and AASHTO is not always conservative because it may produce allowable compressive stresses that are 30 percent larger than UIC 772R when the shape factor is approximately 4.

Elongation at break also has a significant impact on bearing capacity for the BE 1/76 specification, and Figures 35 and 36 were generated for a large elongation capacity (575 percent). Figures 37(a–c), show the allowable compressive stress for reinforced bearings with no applied rotation and a 400 percent minimum elongation at break. This minimum value is typical of that required in most specifications. Comparison with Figures 35(a–c) indicates that the stress permitted by BE 1/76 is significantly reduced by the smaller *EB*. The AASHTO specification is not overly conservative when compared to BE 1/76 under these conditions even though BE 1/76 ignores strain due to rotational deformation. UIC 772R and BS 5400 are independent of minimum elongation.

Figures 38(a–c) serve as a summary of the previous fig-

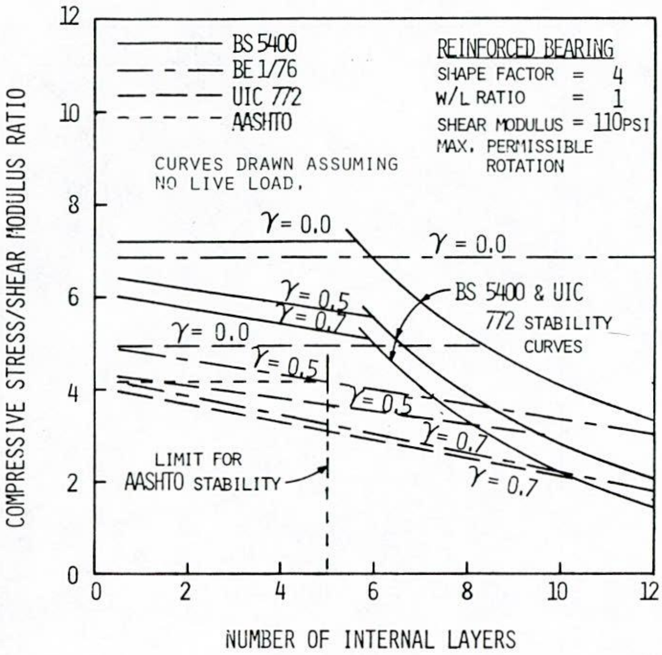


Figure 36(a). Comparison of various design codes for reinforced bearings with maximum rotation—shape factor 4.

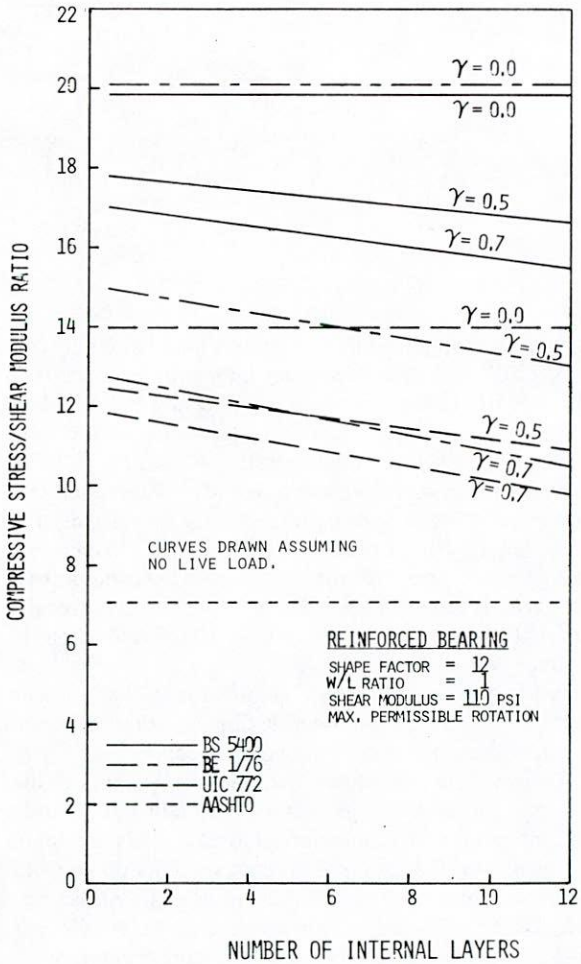


Figure 36(c). Comparison of design codes for reinforced bearings with maximum rotation—shape factor 12.

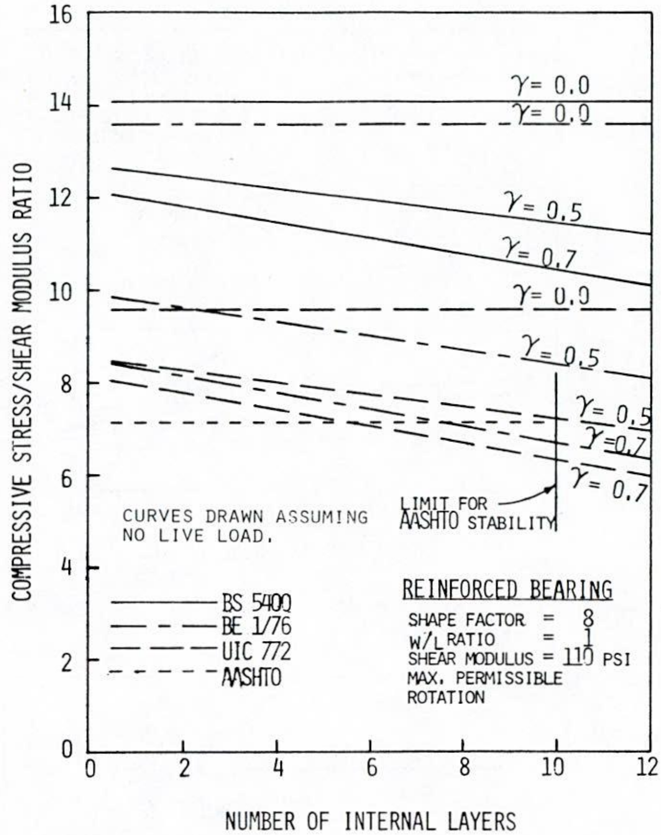


Figure 36(b). Comparison of design codes for reinforced bearings with maximum rotation—shape factor 8.

ures, since they show the allowable compressive stress of the bearing as a function of shear, rotational capacity, and shape factor. They show that the AASHTO specification may be regarded as a simple extreme case specification for reinforced bearings. That is, if the designer assumes that the least favorable load conditions, geometry, shape factor, and material properties all occur simultaneously, the AASHTO design will be comparable to other codes, although it may be a slightly unconservative comparison. Bearings with high shape factor, no shear or rotation, or very good material properties may be very conservatively designed by AASHTO. The stability limits of the AASHTO and BE 1/76 specifications do not show up in these figures, but the AASHTO specification tends to be conservative with respect to stability if the 7 percent strain and 800-psi stress limit are included. Note that the reduced effective area significantly reduces the load capacity of bearings with many layers in Figure 38(a) for UIC 772R, BE 1/76, and BS 5400.

It should be noted also that bearings are designed to sustain translations and rotation while supporting the applied gravity loads. Therefore, the data of the previous figures could be expressed in terms of deformation limits rather than load capacity. This is done in Figures 39 (a–c), which show the rotational capacity per layer of a square reinforced bearing as a function of shape factor, shear deformation, and number of layers for given applied stresses of 400 psi, 800 psi, and 1,400 psi. When shape factors are large, rotation is

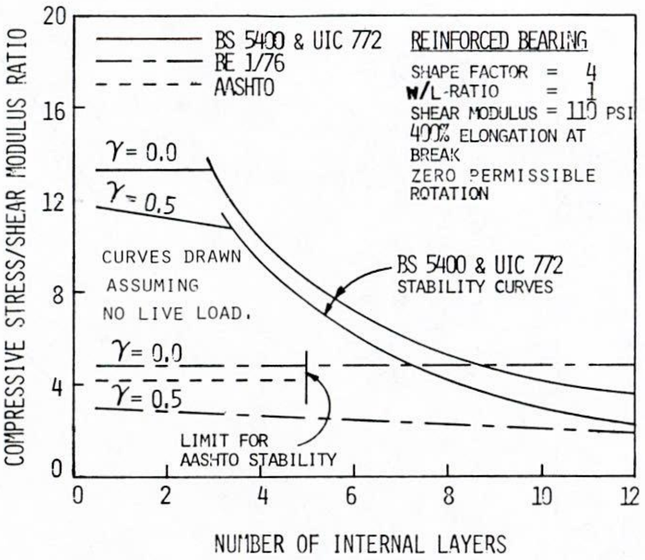


Figure 37(a). Comparison of design codes with reduced elongation at break—shape factor 4.

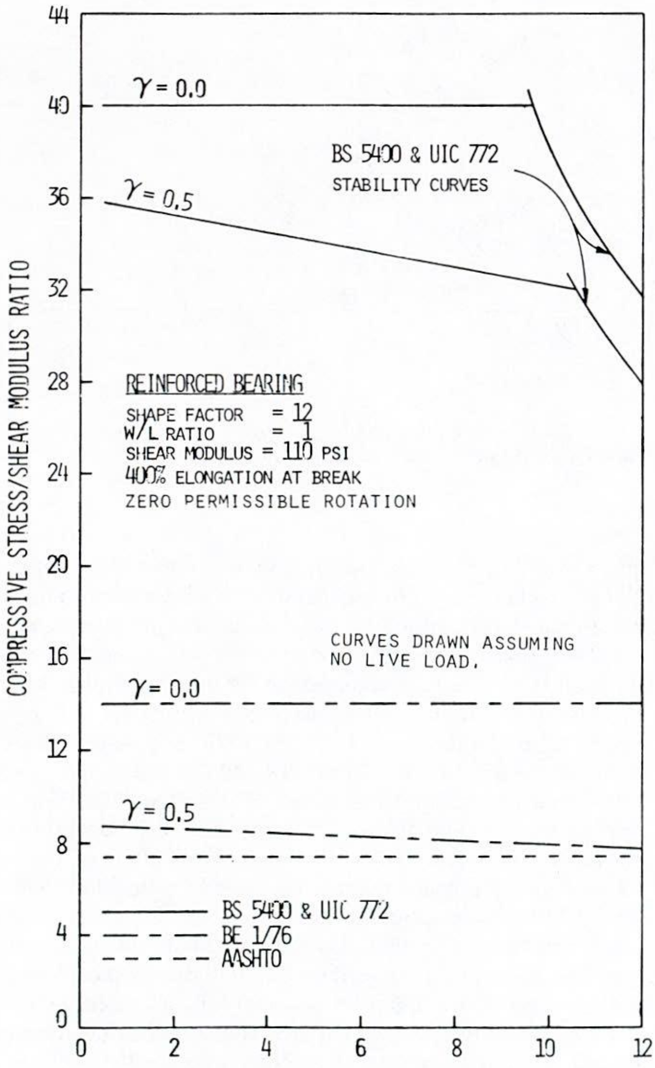


Figure 37(c). Comparison of design codes with reduced elongation at break—shape factor 12.

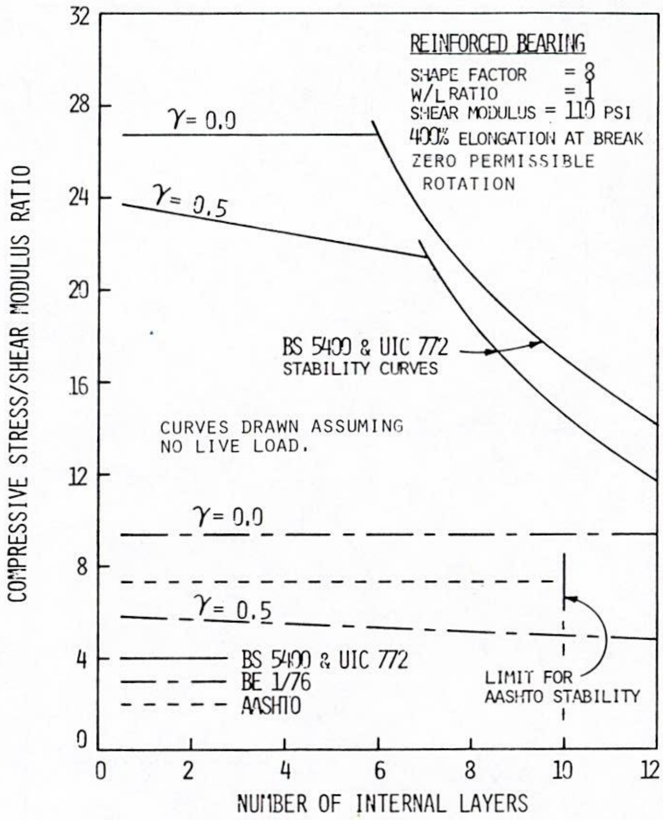


Figure 37(b). Comparison of design codes with reduced elongation at break—shape factor 8.

controlled by the tensile stress limitations (i.e., Eq. 64 for AASHTO or BE 1/76, and Eq. 84 for UIC 772R). At lower shape factors shear strain limitations, compressive strain limitations, or stability controls the design. The allowable rotation per layer decreases dramatically for high shape factors at a given compressive stress level. The rotations permitted by the international railway specifications for a given shape factor and stress level are approximately 100 percent larger than those noted for BS 5400 and approximately 200 percent larger than those for AASHTO or BE 1/76. Note that BE 1/76 and common U.S. practice (quoted on the figures as AASHTO) use the same basic equations for limiting maximum permissible rotation (Eq. 64), but they employ different Δ_c calculations and thus produce different rotational capacities. Large compressive stresses reduce the rotational capacity of bearings with small shape factors (because more of the allowable total shear strain is taken by γ_c), but increase capacity for large shape factors because Δ_c is increased. The curious appearance of the BS 5400 curves is explained by the use of the effective area A_{re} for compressive calculations. At small shape factors, total rubber thickness T is large, so Δ_s is large for a given γ_s,. Large Δ_s reduces A_{re} dramatically and so increases γ_c, which then absorbs so much of the allowable total shear strain that the rotation and its corresponding shear strain, γ_r, must be reduced with lower shape factors.

Strains due to shear deformation, γ_s, consistently reduce the rotational capacity of all bearings. The shape of the bearing becomes very important when rotations are considered. Figure 40 shows the rotational capacity of a rectangular bear-

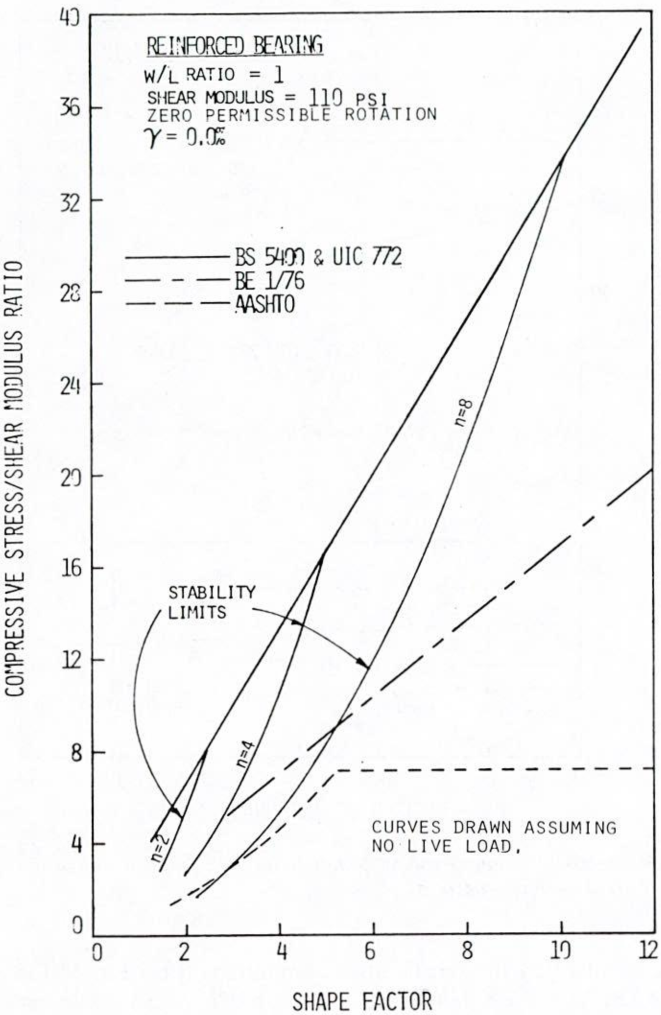


Figure 38(a). Allowable compressive stress as a function of shape factor with no shear or rotation.

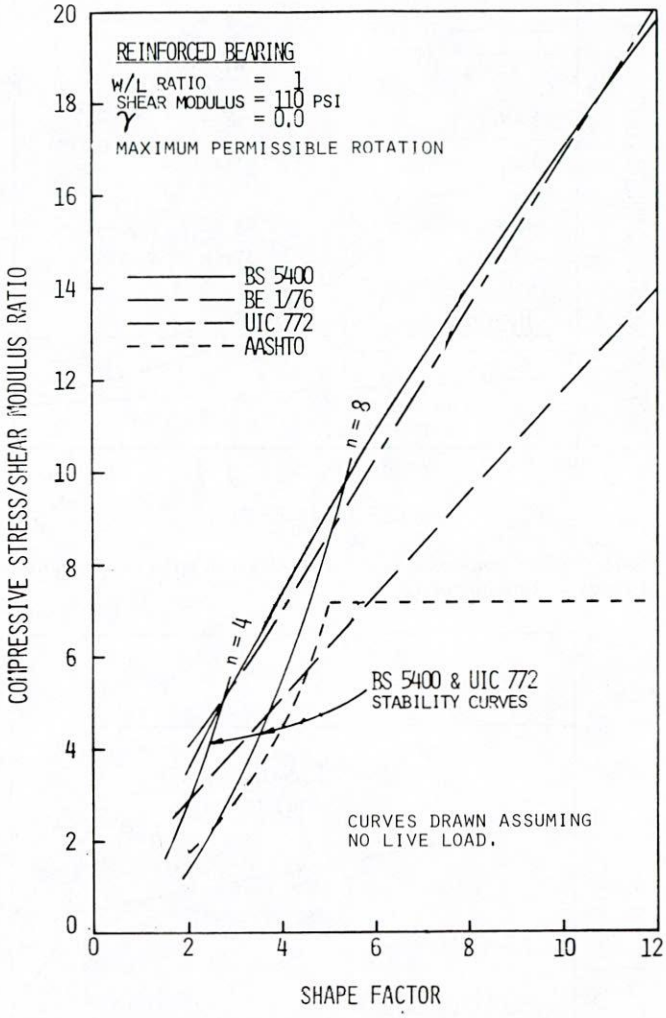


Figure 38(b). Allowable compressive stress as a function of shape factor with no shear and maximum rotation.

ing ($W/L = 4$), and this can be compared to a square bearing at the same stress level in Figure 39(b). The observed increase in rotational capacity is very significant. Rectangular strip bearings are most efficient in rotation because they are capable of sustaining large rotations while causing smaller corresponding shear strains, γ_r, and requiring smaller deflection, Δ_c.

Plain pads usually employ different design procedures, and this comparison is shown in Figure 41. This figure shows the allowable compressive stress as a function of shape factor for the major design procedures. The AASHTO specification is apparently more conservative than the international railway specifications and BS 5400, and it is similar to BE 1/76 with no shear deformation but permits stresses about double those of BE 1/76 when high shear deformations are added. However, this comparison is dependent on the way in which compression strain is calculated. The AASHTO data in Figures 41 and 42 are based on the stress–strain curves in Ref. 9, which do not distinguish between plain pads and reinforced bearings. If design curves which reflect the lower stiffness of plain pads were used, the allowable stress on them would be reduced. It should be noted that all specifications for plain pads are independent of shear strain due to rotational deformation. While the above comparison is a valid measure of the influence of shape factor on allowable loads, it is not the best comparison for individual plain pads because the definition of shape factor varies for different codes. The shape factor for UIC 772R and AASHTO is defined in Eqs. 59 and 60, but BS 5400 and BE 1/76 use a value that is smaller by a factor of 1.8. The reduction is applied because slip may occur at the load surface, and thus the bulging pattern and deformations will be different. Figure 42 shows relationships that are corrected for this difference. The AASHTO specification for unreinforced pads is much less conservative in this corrected comparison. It is consistently less conservative than BE 1/76 for all shear deformations, and it is similar to BS 5400 for low shape factors (i.e., factors 5 or less), which are typically used for unreinforced pads. It is conservative compared to BS 5400 with high shape factor (i.e., 5 or greater by AASHTO definition) and UIC 772R.

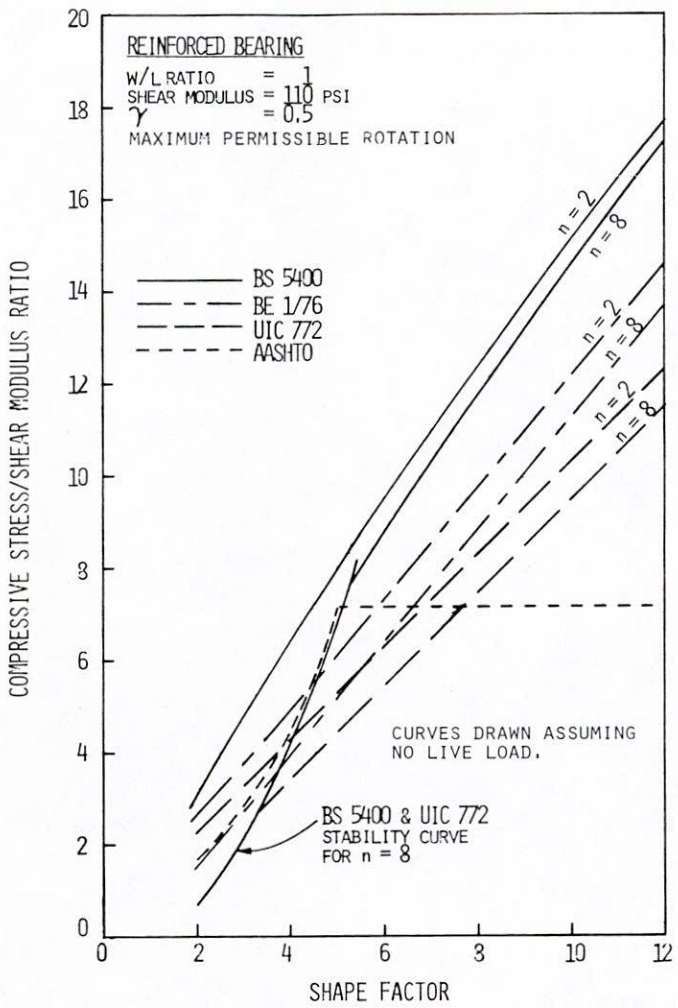


Figure 38(c). Allowable compressive stress as a function of shape factor with maximum shear and rotation.

COMPARISON OF SPECIFICATIONS WITH THEORY

The previous discussion described the design procedure of several specifications. Design is controlled by equations for shear stress, shear strain, compressive strain, plate thickness, and bearing geometry (for stability), many of which are presented without justification or verification. Many of the terms are empirical and based on observations of bearing behavior, but others have a sound basis in previously discussed research results. This section is a brief discussion of the relationship between some of these expressions and research results.

Shear stress and shear strain are the controlling factors in the design of many bearings, and a number of equations are presented for determining these stresses or strains. Gent (*48, 54*) has derived solutions for the stresses and strains in a reinforced bearing under compression. The linear infinitesimal strain theory predicts that the τ_c and γ_c are functions of the bearing geometry multiplied by the average compressive stress. Figure 43 shows a plot of the function for rectangular bearings with different aspect ratios and for circular bearings. It is in conflict with Eqs. 73 and 81, which are used by BE 1/76 and UIC 772R, respectively, for the computations of γ_c. Rectangular bearings produce similar shear stresses and strains for a given shape factor regardless of the aspect ratio, W/L, if the shape factor is greater than 2.0. If the shape factor is less than 2.0, considerable divergence occurs; however, these very small shape factors seldom occur in practice. Circular bearings theoretically produce smaller shear strains than rectangular bearings of the same shape factor for the practical range of shape factors. Both BE 1/76 and UIC 772R provide a conservative estimate of shear (i.e., they overestimate γ_c); UIC 772R is somewhat more realistic in the practical range of shape factors, but is increasingly conservative for small shape factors. It should be noted that the effective area, A_{re}, was taken as the total area in Figure 43. Therefore, the strains predicted by the specifications will be much larger when shear deformation is combined with compression.

The strain due to shear deformation, γ_s, is widely accepted as reasonably correct as defined in Eq. 72 and thus no further discussion is warranted. However, shear strain due to rotation, γ_r, is not so well understood. Gent again derived a theory for rotation and it can be shown that γ_r is directly proportional to the rotation but is also a function of the shape factor and geometry. This function is plotted in Figure 44 along with Eq. 83 of the International Railway specifications. Very good comparison results and the solution is nearly identical for strip bearings. Square or circular bearings with large shape factors produce the largest strains, γ_r, and this clearly indicates why strip bearings are preferable for bearings subjected to rotation. It should be noted that UIC 772R does not recognize the use of circular bearings, but these bearings should be expected to produce large rotational shear stress. Comparisons indicate that rotational strains may dominate the design with BS 5400 or UIC 772R specifications. Thus, circular bearings are ideal when no rotation can occur because γ_c is small and they avoid many problems associated with improper placement, but they are less advantageous when rotation must be considered.

The shear stresses and strains are generally limited either to a constant or a linear function of elongation at break. These limitations were discussed earlier. There is evidence to support both conclusions, but it is clear that neither approach is completely correct.
This must be regarded as a major unanswered question in bearing design.

Deflection limits also merit attention. Relative deflection across a deck joint needs to be limited to protect the joint and to prevent serious discontinuities in the roadway. Absolute deflection will influence the distribution of stresses in continuous bridges, but the extent will be negligible in most cases. The purpose of including a compressive strain limit is less clear. It is not a foolproof way of providing deflection control, and is only effective in preventing material overstress if limits on total shear strains are absent. Deflections and strains are frequently computed by graphical design aids, design equations that are nonlinear approximations of real behavior, or by theoretical solutions that are based on linear infinitesimal strain theory. Both the BE 1/76 and UIC 772R deflection equations (Eqs. 74 and 85, respectively) have some theoretical basis. However, actual comparisons with theory are not shown because neither theory matches experimental results exactly due to the nonlinearity of compressive deflection.

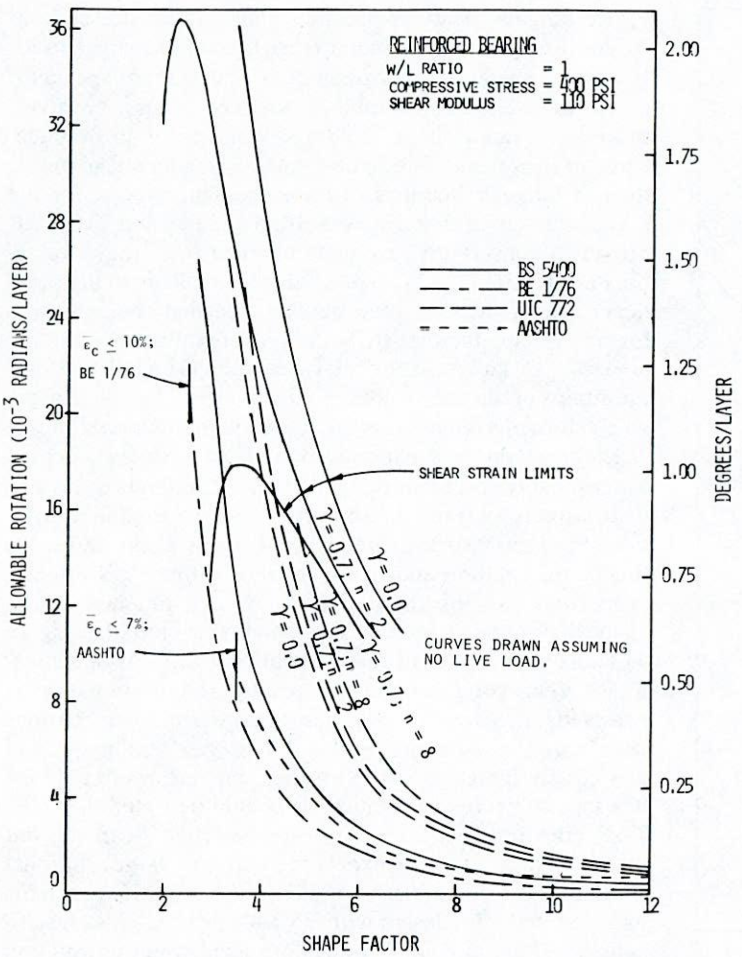


Figure 39(a). Maximum permissible rotation with 400 psi compressive stress.

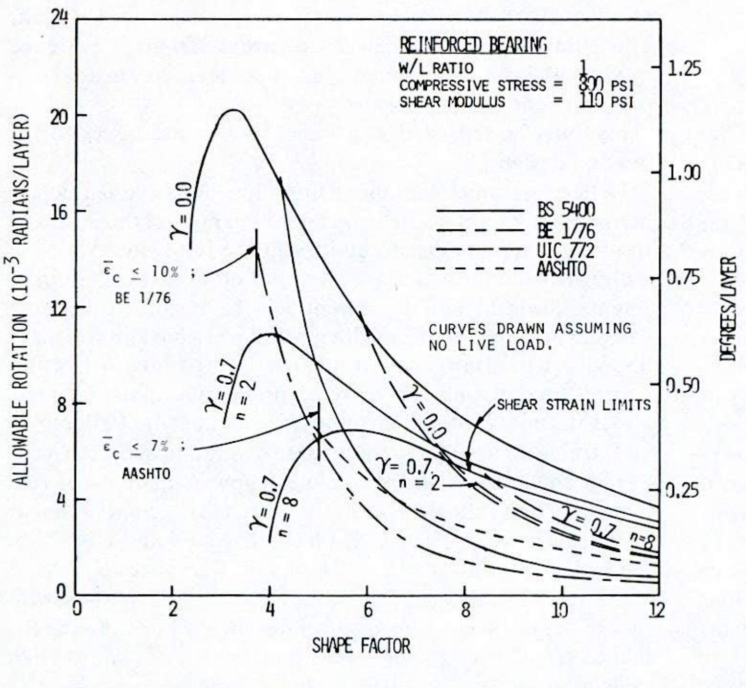


Figure 39(b). Maximum permissible rotation with 800 psi compressive stress.

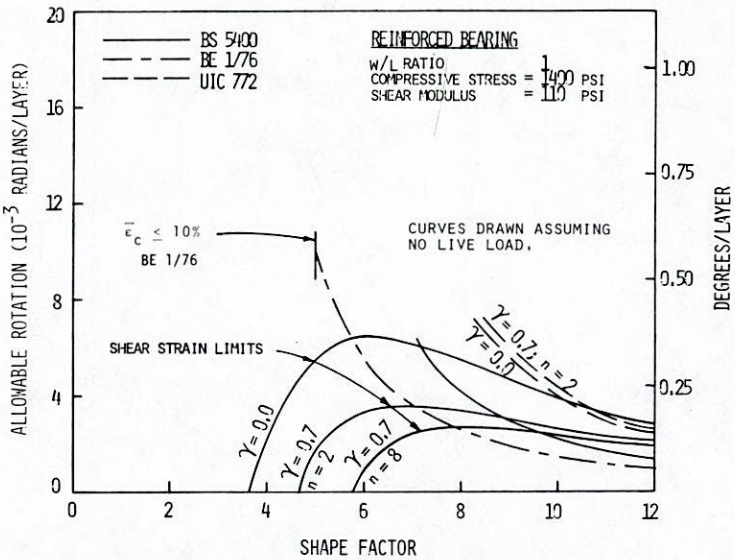


Figure 39(c). Maximum permissible rotation with 1,400 psi compressive stress.

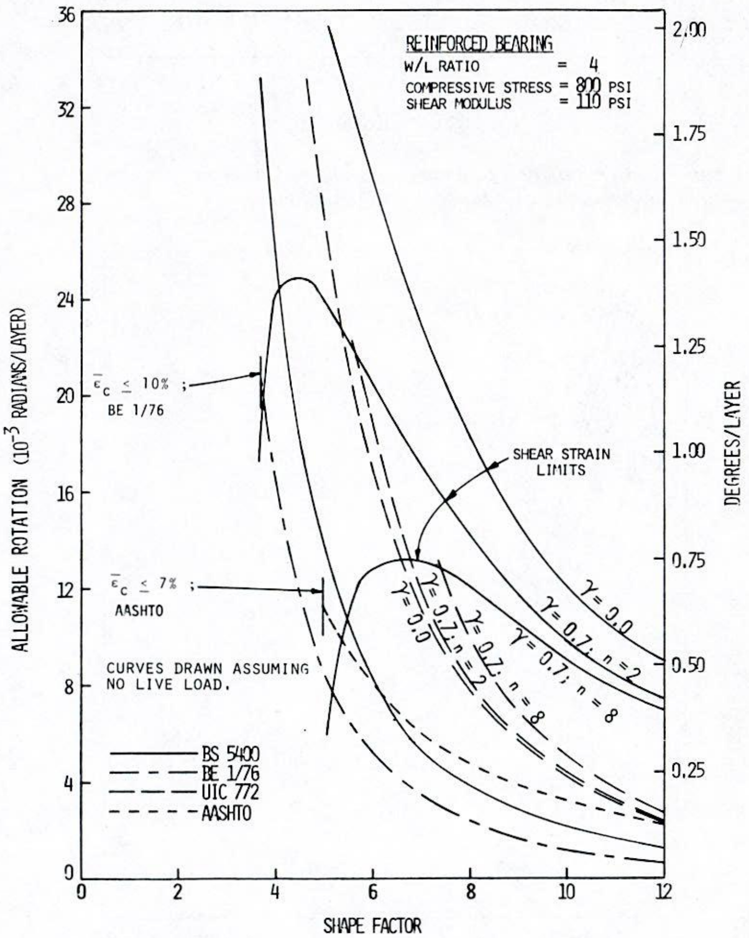


Figure 40. Maximum permissible rotation with a 1:4 rectangular bearing—800 psi compressive stress.

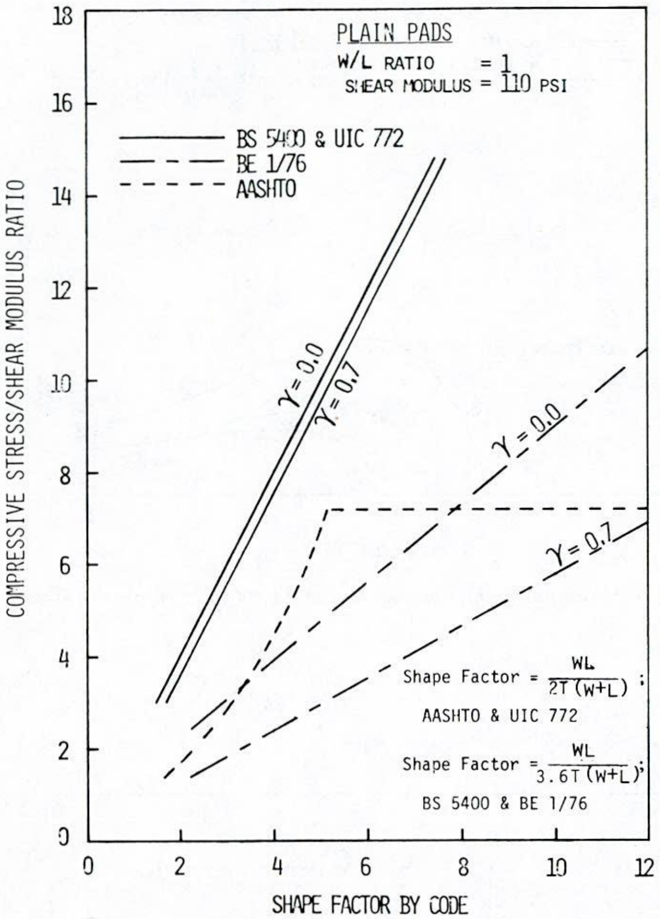


Figure 41. Comparison of compression allowed on plain pads by various design codes as a function of shape factor.

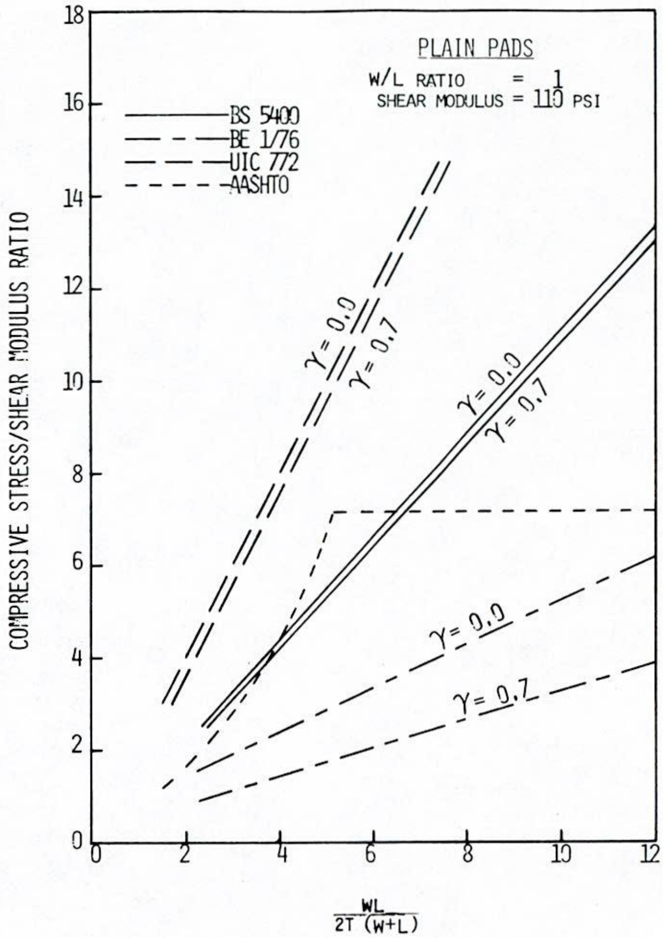


Figure 42. Comparison of compression allowed on plain pads by various design codes as a function of geometry.

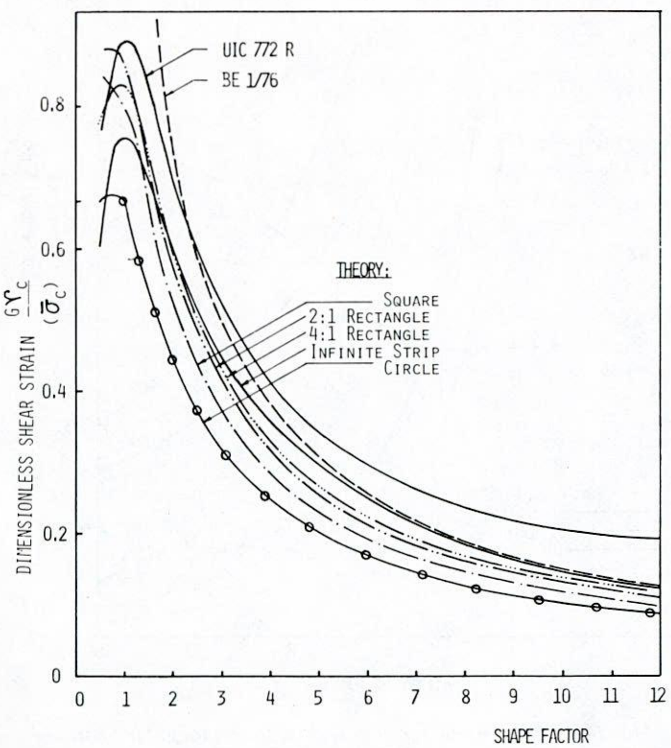


Figure 43. Shear strain due to compression vs. shape factor.

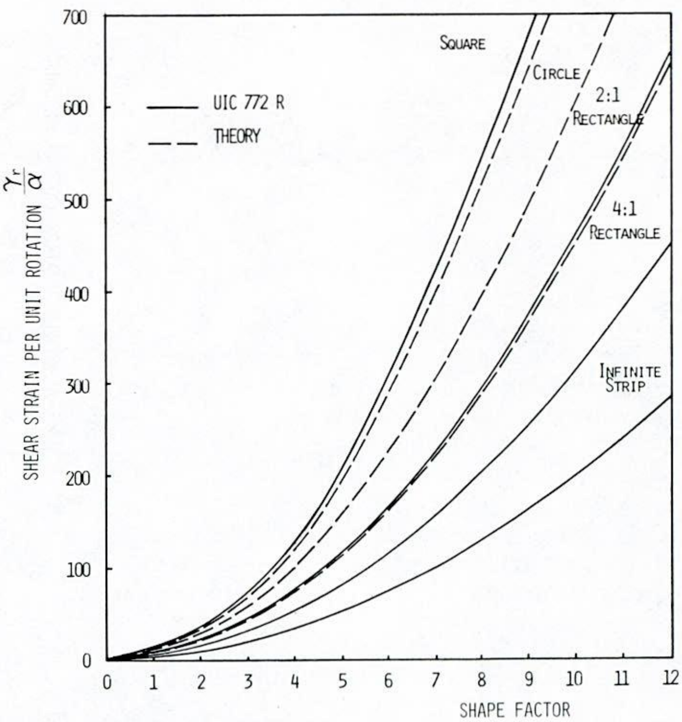


Figure 44. Shear strain due to rotation vs. shape factor.

Experiments on bearings have shown that tensile rupture of the reinforcement is an important mode of failure under compressive load, but very little effort has been devoted to determining the stresses in it. The shear stresses in reinforced bearings under compression can be integrated over the area to determine stress in the reinforcement for some shapes. This was done for a long thin strip and compared in Figure 45 to the design equations of UIC 772R and BE 1/76. The design equations are very conservative, giving an actual factor of safety against yield on the order of 3 or 4 against yield under service loads. However, holes in the plates or changes in shape introduce stress concentrations that must be superimposed on the theoretical stresses, and they may have a significant impact on the fatigue strength. This topic merits close attention but has received none.

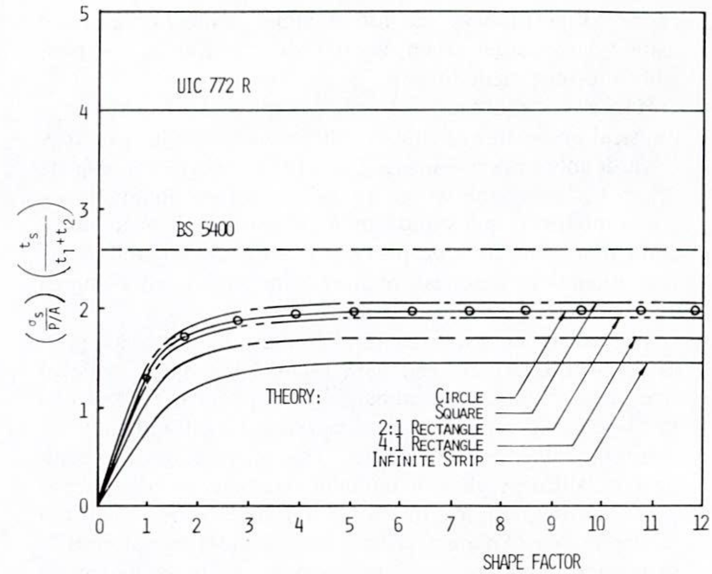


Figure 45. Dimensionless stress in reinforcement vs. shape factor.

CHAPTER SEVEN

CONCLUSIONS AND SUGGESTED RESEARCH

Discussions of individual aspects of elastomeric bearings have been presented in the preceding chapters. This chapter summarizes the major findings of the research, presents discussions of two possible sets of code provisions, and outlines the most pressing research needs.

SUMMARY OF MAJOR CONCLUSIONS

The investigation showed beyond doubt that elastomeric bearings can fulfill their requirements admirably if they are properly designed and constructed. Failures today are fewer than they were when elastomers were first used, and those that occur are generally associated with poor design, materials, or workmanship rather than any fundamental deficiency in the concept of elastomeric bearings. No evidence has come to light to cast doubt on the fundamental concept of elastomeric bearings, and they are now used by an ever increasing number of bridge engineers who appreciate that they are cheap to manufacture, easy to install, and need no maintenance. This satisfactory record has led to bigger bearings and more sophisticated applications, but these now require a better understanding of the behavior of the elastomer and the finished bearing than presently exists.

The material properties and physical behavior of the elastomer are radically different from those of most other civil engineering materials, and so many engineers do not understand them well. Material properties are discussed in greater detail in Chapter Three, but it should be emphasized that elastomers are complex polymer compounds which have mechanical properties that are sensitive to changes in compounding and vulcanization process. They are flexible in shear and in uniaxial tension or compression but very stiff in volumetric deformation. The material properties are nonlinear, and time and temperature dependent. Thus, factors such as creep and low temperature stiffening are important to the designer. The modes of failure of rubber in elastomeric bearings are not well understood, but it is generally believed that the development of tensile stress in the rubber is the predominant one. In particular, fatigue is of critical concern, and related problems such as internal rupture, ozone cracking, and mechanical crack growth are consequently important. Only polychloroprene (CR) and natural rubber (NR) have been used widely in bridge bearings. These rubbers are both strain crystallizing materials, and this enhances their fatigue life. Further, their economy and their history of good performance cause a natural hesitance to try new polymers. Their hardness nearly always lies in the range of 50–60 because softer rubbers are too flexible, while harder ones usually have larger creep, greater shear stiffness, and possibly shorter fatigue life.

Hardness is still the most widely employed measure of the physical properties of rubber, but its use impedes progress towards more precise bearing design. Rubber technologists favor it, despite the variability of the values obtained, because the test is quick and simple. However, shear modulus is the more important propertyfor bearing design, and, while it is related to hardness, it may vary significantly among compounds of the same hardness.

The elastomer is formed into bearings either as plain pads or as bearings reinforced with bonded laminates made of fiberglass or steel. The laminates develop tensile stress when the bearing is subjected to compressive load and they restrain the bulging of the rubber. This provides greater compressive stiffness while maintaining flexibility with respect to shear deformation. Rupture of the reinforcement is the usual failure mode when the bearing is loaded under monotonically increasing load. Steel reinforcement is typically much stronger than fiberglass fabric and thus steel reinforced bearings usually have a much higher ultimate load capacity. Holes in the bearing and reinforcement will hasten this rupture and may increase the tendency toward fatigue failure, so they should be discouraged until they are better understood. Unreinforced bearings behave differently because they depend on friction at the loaded surfaces to restrain bulging. This friction is unreliable and thus plain pads should be designed more conservatively than reinforced bearings.

Analytical techniques are an important tool to the designer. However, while the general behavior of elastomeric bearings is understood in principle, reliable and accurate methods of analysis are not yet available. The primary analytical tool used by designers is linear elastic theory with infinitesimal strains. As noted earlier, rubber is not a linear elastic material, and, further, bearings are commonly deformed to large strains, which violate the assumptions of infinitesimal strain theory and introduce secondary effects under combined loads. Despite these deficiencies, nearly all the analytical theories available to the designer are based on linear theory with some approximate corrections for the nonlinear effects. Further, only a limited number of experiments have been performed to verify the validity of these analytical methods. These methods are discussed in greater detail in Chapter Four.

The design of elastomeric bearings is based primarily on shape factor. This factor is only an approximate tool for estimating the behavior of a bearing because it does not fully account for different shapes in plan. However, it appears to be acceptable for present day design of reinforced bearings, and can also be used for plain pads if a correction is used to account for the difference in behavior between them and reinforced bearings.

Detailed discussions and comparisons of U.S. and foreign design specifications are contained in Chapter Six, but it should be noted that the present AASHTO specifications have several deficiencies.

Firstly, the 7 percent compressive strain limitation and the 800-psi stress limit do not appear to have a rational basis. But, although they are not derived from consideration of specific modes of failure, they generally keep stresses low enough to prevent trouble. The present AASHTO specification is very conservative for steel reinforced bearings with high shape factors and no shear or rotational deformations. It is less conservative for reinforced bearings with high shape factors and large shear and/or rotation, and it is somewhat unconservative for reinforced bearings of low to moderate shape factors and maximum shear and rotations. It is very unconservative for plain pads.

Secondly, the present AASHTO specification has no limitation on rotation of the bearings, although this is regarded as important in many other design codes. Nor does it have a rational serviceability criterion. The 7 percent compressive strain limitation may be interpreted as a serviceability limit, but it is not rational because serviceability damage to the bridge and joint sealants is likely to be controlled by relative deflections rather than absolute strain of the bearing. The specification also fails to address the strength of the reinforcement, despite the fact that it almost always controls static load capacity. The quality control of the manufacturing process and acceptance critria of elastomeric bearings are not defined, and finally the friction limits for controlling slip of the bearings are not rational.

In view of these limitations, a series of changes are proposed for the AASHTO specifications. Two separate proposals referred to as Methods A and B are offered, and they are included in Appendixes D and E. Method A is quick, simple, and safe for a wide range of applications, while Method B makes possible more refined designs at the expense of increased design effort. The researchers feel that in the long term the specifications would be most useful if both methods were included, providing bases for design both of run-of-the-mill and special-purpose bearings. However, it is apparent that the knowledge and control mechanism necessary for a good, reliable Method B are not presently available, so Method A alone is proposed for consideration by AASHTO now. As proposed, Method A contains a clause that permits the engineer to sanction designs to different rules, thus serving part of the purpose of Method B.

Method B is presented only in skeletal form because its details can be filled in only when the conflicts and unanswered questions outlined in earlier chapters have been resolved. They are summarized later in this chapter under "Research Needs."

The two methods are discussed in the following sections, and the limitations and research requirements for finalizing Method B are discussed under "Research Needs."

DISCUSSION OF METHOD A

Method A resembles the existing AASHTO Specifications with a few major exceptions. The subject matter is still divided into section 1.12 (Design) and section 2.25 (Construction), with the information on material located in 2.25. A separate material specification exists now (AASHTO M251-74), but it contains many clauses that duplicate or contradict those of the main Bridge Specification, and so is ignored here on the supposition that it will either be rescinded or rewritten in the future. Sections 1.12 and 2.25 have been reorganized into a more logical format in Method A.

In section 1.12 (Design), significant departures from the existing provisions are to be found in the following areas:

1. Plain pads and reinforced bearings—These are treated differently as concerns both allowable stress and deflection.

2. Material properties for design—The use of shear modulus rather than hardness as a measure of physical properties is encouraged, and a range of shear modulus values is given which correspond to specific hardnesses. These may be used when more specific information is not available.

3. Compression behavior—Strength is considered by introducing an allowable compressive stress that is limited directly as a function of bearing geometry and material properties. For serviceability, compressive deflection rather than strain becomes a design criterion and is limited so as to protect joints in the bridge deck.

4. Rotation is considered.

5. The strength of the reinforcement is addressed to ensure adequate performance at service loads.

6. The requirements for anchorage against horizontal slip have been changed.

Of these, the provisions for compression (1 and 3 above) are likely to have the most effect on design. The others are included because they form an integral part of a rational design procedure—although with present stresses and manufacturing techniques they seldom govern the design. Their importance is likely to increase as future developments permit higher allowable stresses.

Allowable compression is presently limited to the lower of 800-psi stress or 7 percent strain, with no distinction between plain pads and reinforced bearings. Strains are most often obtained from manufacturers' design aids, most of which also fail to distinguish between plain pads and reinforced bearings.

The proposed compressive stress limits are written in terms of the shape factor, which is evaluated in the traditional way for internal layers of reinforced bearings, but should be divided by a factor β (greater than 1.0) for the purpose of computing allowable compressive stress and compressive deflections. The result is that if a plain pad and a layer of a reinforced bearing have the same dimensions, the effective shape factor of the plain pad will be about one-half that of the reinforced bearing. With this definition of shape factor, and with G as the shear modulus, proposed compressive stress limits are:

1. The lower of GS/β or 800 psi on plain pads or fiber reinforced bearings.

2. The lower of GS/β or 1,000 psi on steel reinforced bearings.

3. However, stresses on fixed (i.e., no shear) reinforced bearings may be increased 10 percent above these values (i.e., an absolute maximum on any bearing of 1,100 psi).

These provisions permit higher stresses on bearings of higher shape factor or made from stiffer material. In practice this is also true of bearings designed to the existing provisions that are governed by the 7 percent strain limit. However, the GS/β limit is proposed because it is simpler, more direct and independent of manufacturers' design aids. It might appear from the provisions that use of a stiff material is desirable because a smaller bearing could be used, causing less horizontal force. This is not the case because the reduction in plan area would be compensated for by the increase in material stiffness and the horizontal force would remain the same. The only advantage would be the reduction in volume of rubber used. However, the use of hard material is discouraged because other aspects of its behavior are not fully satisfactory. Using higher stresses with higher shape factors and restricting the rubber hardness to the range of 50–60 are consistent with theoretical and experimental results.

Many engineers appear to believe that allowable compressive stresses on elastomeric bearings should be increased dramatically, and the authors anticipate that the stresses proposed in Method A may not meet the expectations of such engineers. The reasons underlying the choice thus merit discussion.

Firstly, changing the absolute maximum stress from 800 to 1,100 psi represents an increase of 37.5 percent, which by any standards is substantial. If the allowable compressive stress on steel columns were to be increased by 37.5 percent, the change would almost certainly be regarded as excessive, particularly if no experiments had been performed on columns at the higher stress level. Secondly, the mechanics of elastomeric bearings are difficult, are totally unlike those of other structural materials, and consequently are not well understood by many engineers. It is felt that this has contributed to the propagation of the belief that the bearings are currently grossly understressed. This view receives further support from the fact that bearing failures appear to be rare, and can mostly be traced to poor quality materials. However, several points should be borne in mind. Possibly the most important is that the criteria for failure have not all been clearly identified, and, until they are, refined designs close to limits are not possible. Failure by overstressing the reinforcement or by instability are reasonably well understood, but the stress which the rubber itself can carry remains a major unanswered question. Significant unresolved facets of this problem are posed by fatigue, combined stresses, and large deformation effects. Until these questions are addressed, the risks involved in a dramatic increase in allowable stress outweigh the benefits. The initial cost of bearings represents only a small fraction of the total price of the structure; yet, if they fail, the cost implications can be very serious. A number of examples of this situation are extant today. The question is further obscured by the fact that failure of the rubber is not a dramatic event. It is generally associated with creep or delamination, both of which lead to excessive deflections and loss of serviceability rather than total collapse. That such events occur over time means that in the field they may go unnoticed, thus enhancing the impression that no bearings are failing. However, the potential for damage to the total bridge may be serious.

A comparison with other specifications is possible, but the results should be viewed with caution. There is a wide variation in stresses allowed by other codes, which suggests that none of them have resolved all of the problems. Figures 46 and 47 show the allowable compressive stress divided by the shear modulus plotted against shape factor (defined in the traditional way) for plain pads and reinforced bearings.

Upper and lower bounds on the allowable stress are shown for most European codes, representing the case when the simultaneous rotation and shear are either zero or maximal. The new provisions appear to fall at the low end of the range

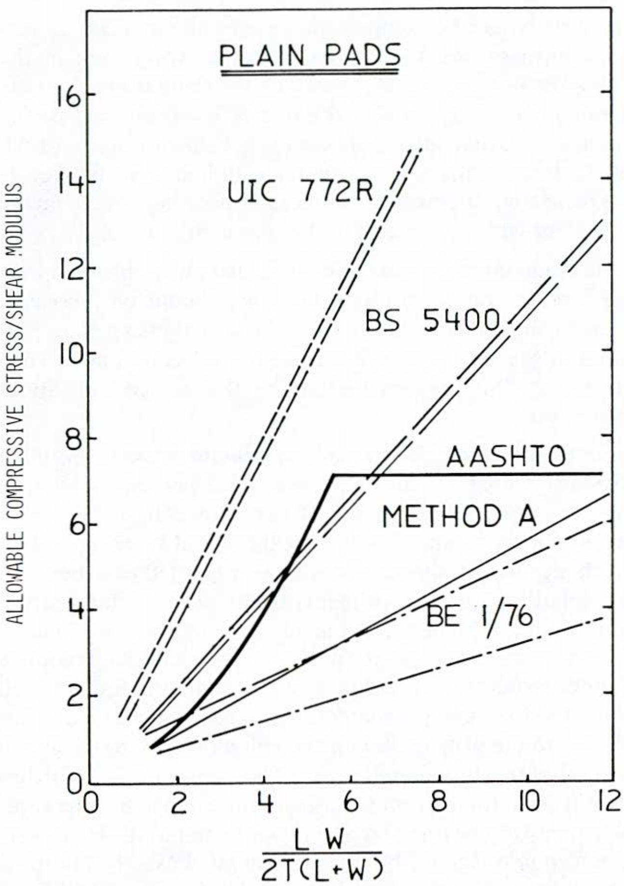


Figure 46. Comparison of proposed AASHTO Method A with other design codes—plain pads.

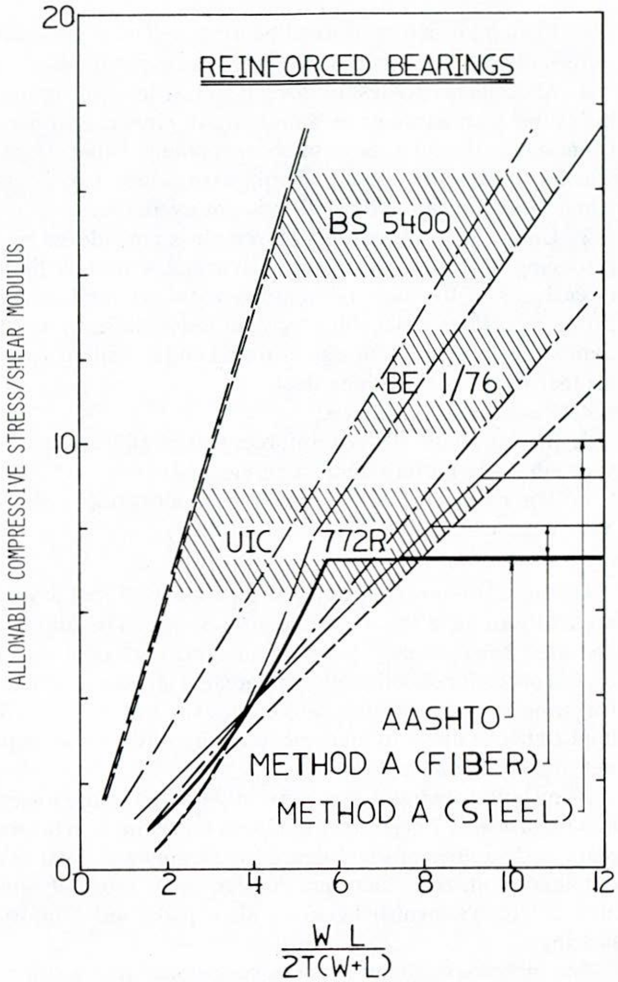


Figure 47. Comparison of proposed AASHTO Method A with other design codes—reinforced bearings.

in most cases, which is partly explained by the fact that they assume that maximum shear and rotation will be present. Also, some of the specifications include an allowance for impact (e.g., 50 percent in UIC 772R). Furthermore, BE 1/76 (and the Australian AS 1523 which is similar and so is not shown) bases its allowable compressive stress on the elongation at break, *EB*, of the rubber. Since AS 1523 demands a minimum *EB* of 575 percent, the BE 1/76 curve was drawn for that value. For a more realistic comparison with material used in the United States (using an *EB* of say, 400 percent), the values shown should be adjusted down proportionately.

The German specifications also permit stresses almost twice those proposed for AASHTO, but only for bearings of specific, regulated dimensions, with very high shape factors (between 9 and 13), relatively low rotation, and extremely stringent quality control standards. These involve extensive testing (which is very time consuming and expensive) and absolute adherence to the compound that is approved. As a result, only 4 companies are licensed to manufacture bearings and all use identically the same compound. Flexibility and freedom of choice of materials and dimensions are clearly considered by the German specification to be incompatible with reliable high performance.

The graphs also show that, contrary to popular opinion, there are specific instances (particularly at shape factors between 4.5 and 6) when the existing AASHTO provisions are among the *least* conservative.

The last reason why both the existing and proposed AASHTO provisions may appear conservative is that they permit the same compressive stress regardless of the presence or absence of other loadings, whereas most European codes place a limit on the total stress or strain caused by all three modes of deformation (compression, rotation, and shear). Clearly, under European rules, if rotation and shear are nonexistent, the allowable compression can be increased accordingly. This design approach is attractive because it allows higher stresses in special cases; for example, smaller rotations would be anticipated in continuous bridges, permitting a higher stress to be used in compression. However, to justify such a sophisticated approach, the associated techniques of theoretical mechanics, the experimental evidence to substantiate them, and the control over material properties must all be raised to a comparably high level of reliability. The authors believe that such developments would require a considerable research effort, and when they are completed they should be incorporated in a more sophisticated design method (i.e., Method B).

In section 2.25 (Construction) two major changes are introduced. Firstly, the problem of low temperature performance is addressed by permitting selection of an elastomer whose resistance to stiffening is appropriate to the temperatures prevalent at the site. This reflects the wide range of climatic conditions in the United States and should permit a wider variety of compounds (particularly of CR) to be used in warmer regions. The classification follows ASTM D-2000 and D4014. The second change concerns acceptance of the bearings, which is treated at two levels. The intention is to use Level I criteria which depend on standard material quality control tests and, in addition, to proof load each steel reinforced bearing. No measurements need be taken during the proof loading, so it can be performed quickly and cheaply, but it is an effective way of detecting bearings with serious defects in the laminations. Level I criteria represent minimal conditions for acceptance of any bearing, and would be used alone for acceptance of run-of-the mill units. Level II acceptance criteria are more stringent and are intended for use either as an arbitration tool in disputes over acceptance of ordinary bearings, or, in addition to the Level I criteria, as the basis for acceptance of more critical bearings. The tests for acceptance are likely to change as Method B is developed, for obvious reasons.

Other lesser changes are also introduced. Tapered elastomer layers are banned because their behavior is not well understood, and any out-of-level during installation is to be corrected with a grout bed or other suitable means. If the end slope of a steel or precast concrete girder exceeds the rotation capacity of the bearing, a suitable device (such as a convex or tapered top plate) should be introduced to reduce the imposed rotation. A clause has also been introduced mandating that bearings designed to act as a unit be fabricated as a single unit. Although this may appear to be unnecessary, cases have been reported of large bearings being fabricated in segments. The provision is included because the question is likely to arise only in the case of large bearings which are presumably critical to the bridge's behavior, and for which the consequences of inadequate performance may be severe.

DISCUSSION OF METHOD B

A skeletal form of Method B is contained in Appendix E. As presented it cannot be used for design because further research is needed to establish a number of limiting values. In particular, the numerical values for limiting shear stress and strain must be determined. There are many ways which Method B will differ from Method A, but discussion of them remains somewhat fruitless until the necessary limiting values are established. However, a few of the major differences are pointed out below.

The most significant changes lie in the definition of the allowable stresses and the selection, description, and quality control of the elastomer.

The allowable stresses follow reasoning similar to that found in most European codes. The shear strain is considered the most accurate measure of the degree of distress of the elastomer, and so the primary design criterion is a limit on the total shear strain due to compression, rotation, and shear. Whether the limiting value should be a constant or a function of the physical properties of the elastomer is not at present clear. Each individual mode of deformation (compression, rotation, and translation) would also be subject to an individual limit. Because elastomeric bearings have much greater capacity under loads that are monotonic rather than cyclic, the limits are likely to be more restrictive for cyclic loadings.

Because the stresses and deformations which will be permitted in Method B are likely to be larger than those permitted by Method A, a more refined stability criterion may be needed and design concepts, such as the reduced effective area (see Ch. 6), may be employed.

The specification of the elastomer may also be more restrictive in Method B. Ideally, material tests should be developed that would directly determine the suitability of a material for use as a bridge bearing. This would permit rapid appraisal of new materials as well as providing reliable quality control tests for existing ones. However, judging by the difficulties experienced to date in defining precisely those qualities needed in a bridge bearing elastomer, this task may prove extremely difficult. A useful step in the right direction would be to develop tests and/or other acceptance criteria that could be used to determine which compounds of CR or NR make suitable materials for the higher stress applications envisaged under Method B. These would undoubtedly have to include fatigue tests.

Method B would be based on shear modulus rather than hardness because the former varies less and reflects more accurately the properties pertinent to bearing behavior. Bulk compression may be included too, because its effects become important at the high shape factors which may be more common with Method B.

Appendix E and the foregoing discussion show that Method B is at present an attempt to look into the future as realistically as possible rather than to provide a sophisticated design method for use today. However, it serves the vital function of identifying the research which needs to be undertaken. Previous chapters (4, 5, and 6) suggest that theoretical developments in the field have been relatively orderly, although restricted to theories of infinitesimal deformation, whereas by comparison experimental work has been fragmented and lacking in clearly defined goals. Code developments such as Method B require an orderly series of experiments to verify existing theories and evaluate simplifications and code provisions. Random tests of "off the shelf" bearings will serve little purpose and may further complicate the interpretation of existing research. Thus, the following section outlines the areas where experimental and theoretical research is most urgently needed for the development of better, more refined design procedures.

RESEARCH NEEDS

A substantial body of research is needed before rubber products can be designed for civil engineering applications with the same confidence, reliability, and margins of safety as are common with more widely used materials. However, an ordering of research priorities is clearly needed for progress to be as rapid as possible, and in this section two categories of needed research are presented. Category I contains the most pressing problems, while those in II are slightly less so. Category I problems must be resolved before an appropriate Method B can be developed. Those in Category II are desirable, but not essential. A third category of research, Category III, could be defined, but will not be presented because this research is not essential to development of new and refined design procedures. The ordering within each category is arbitrary. In some cases, particularly concerning experimental work, some less urgent problems may be investigated at the same time as more pressing ones with very little extra effort. Clearly, researchers should take advantage of this situation whenever possible.

Category I

1. *Rotation*—Some design specifications suggest that rotation is the most critical parameter in the design of elastomeric bearings, since very small rotations may cause large shear strains in the rubber. However, virtually no experiments have been performed to determine an appropriate limit on rotation or the validity of its shear strain prediction. Experiments are needed to determine approximate rotation limits and substantiate theoretical predictions of rotational stiffness and stress in the elastomer.

2. *Combined Loads*—Elastomers are highly nonlinear materials, but design specifications almost invariably treat them as linear. They assume superposition applies and they compute stress and strain by linear theories. This may be grossly in error, particularly at higher stress levels and larger deformations. Thus, theoretical and experimental work is needed to evaluate:

   a. Combined rotation and compression.
   b. Combined compression and shear.
   c. Combined compression, shear and rotation.

In addition, study is needed to investigate the stability of bearings, the change in stiffness induced by combined loads, and the effects of nonlinear material properties on these combined load conditions.

3. *Failure Criteria*—Failure criteria of elastomeric bearings under static and dynamic loads are not well defined. Recent tests (*75*) have studied failure due to monotonically increasing compressive load, but fatigue and other criteria have been investigated only sporadically. However, fatigue and tension related effects are believed to be the most critical causes of failure. Several specific questions must be answered.

   a. Does fatigue life of the rubber depend on material properties such as elongation at break? Or is it independent?
   b. What is the failure criterion for the rubber under slowly applied monotonic load?
   c. Do very big bearings fail in the same way as small bearings under various loadings?
   d. Do holes or unusual shapes affect the strength or modes of failure? If so, how?

4. *Material Properties*—The material tests in the existing AASHTO Specifications are misleading because they appear to measure those properties which are necessary and sufficient to ensure that the bearing performs well in the field. In fact they do not do so, because those properties have never yet been clearly defined. Thus the tests are really general quality control tests that have a history of producing satisfactory bearings most of the time. Research is needed to isolate those material properties which lead to good behavior and to define tests to measure them reliably.

5. *Stability*—Further research is needed on stability, including an in depth evaluation of the effects of combined shear and compression on the material properties and stiffness of a bearing and the effect of various boundary conditions (such as elastically restrained bridges) on shear stability.

Category II

1. *Materials*—Additional materials research is needed such as the investigation of new polymers for bridge bearings.

2. *Friction and Slippage of the Bearing*—Research is needed to determine appropriate values for the friction between a bearing and its contact surface, and to identify the parameters which influence the friction force, particularly under cyclic loads.

3. *Cover*—The effect of edge cover on the strength and stiffness of self-reinforced bearings merits study.

4. *Size and Shape Effect*—The shape factor is a reasonable design tool, but it is not a theoretically correct indicator of bearing behavior because it does not account for difference in shape. Thus development and verification are needed of more precise theories that account for different plan geometries.

5. *Survey of Bearings in Service to Evaluate Their Performance*—Bridge engineers apparently seldom examine elastomeric bearings closely after they are in service. This, and the relatively short history of elastomeric bearings, raises uncertainty as to the effectiveness of existing design procedures. Thus, a few random inspections of bearings in service would be valuable in assessing the effectiveness of existing design provisions and changes such as those outlined in Method A.

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APPENDIX A

NOTATION

Symbol	*Description*
A	Plan area of bearing.
A_e	Effective (plan) area of bearing in compression, equal to area of reinforcement in a reinforced bearing, and area of elastomer in a plain pad
A_{re}	Reduced effective (plan) area of bearing in compression: A_{re} = (effective width) × (effective length-shear deformation in direction of length)
A_s	Plan area of bearing effective in shear

Symbol	*Description*
a	Half length of bearing
B	Ratio $\frac{\text{minimum compressive load}}{\text{maximum shear force}}$ in BE1/76
b	Half width of bearing
C_1 thru C_8	Dimensionless constants in design Method B
D	Diameter of circular bearing
E	Young's modulus (lateral displacement unrestrained)
E_c	Apparent Young's modulus in compression (accounts for restraint of lateral displacement)
E'_c	As E_c, but modified to account for bulk compression: $1/E'_c = 1/E_c + 1/K$
E_r	Apparent Young's modulus in rotation (accounts for restraint of lateral displacement)
f_c	Coefficient by which Young's modulus, E, must be multiplied to give apparent modulus in compression: $E_c = f_c E$
f_{c1}, f_{c2}	Components of f_c which address, respectively, unrestrained (uniaxial) compression and lateral deformation under constant compressive deformation: $f_c = f_{c1} + f_{c2}$
f_r	Coefficient by which Young's modulus, E, must be multiplied to give apparent modulus in rotation: $E_r = f_r E$
f_{s2}	Coefficient relating maximum hydrostatic stress to average compressive strain: $p_{\max} = f_{s2} E \bar{\epsilon}_c$
g_c	Coefficient relating maximum shear stress to average compressive stress: $\tau_{\max} = g_c \bar{\sigma}_c$
g_r	Coefficient relating maximum shear stress to applied rotation for that layer: $\tau_{\max} = g_r E\alpha$
G	Shear modulus of elastomer under static load
G_a	Apparent shear modulus of a finished bearing: $G_a = \dfrac{HT}{A\,\Delta_s}$
H	Horizontal force
i	Summation index
I	2nd moment of area of bearing in plan
k	Dimensionless material constant used in compressive force deflection relation of restrained rubber blocks—value depends on hardness
K	Material bulk modulus (in compression)
l	Instantaneous length of elastomeric element (in incremental theory, Ch. 4) or length of Euler column
L	Length of bearing (parallel to longitudinal axis of bridge)
M	Resisting moment of bearing under applied rotation
n	Summation index

Symbol	*Description*
p, $p(x,y)$	Hydrostatic compression stress in elastomer
$p_{\max}$	Maximum hydrostatic stress in elastomer
P	Compressive load on bearing
P_{cr}	Critical load for instability of bearing
P_G	Product of shear modulus and area effective in shear: $P_G = GA_s$
P_E	Euler load (buckling load when only flexure is considered)
Q_n	Coefficient in series solution for stresses caused by rotation, including bulk compressibility
r	Radial coordinate
R_n	Coefficient in series solution for stress caused by compression including bulk compressibility
S	Shape factor
S_i	Shape factor for layer i
t	Thickness of a single layer of elastomer
t_i	Thickness of ith layer of elastomer
t_1, t_2	Thickness of elastomer layers directly above and below reinforcement, in calculation of reinforcement stress
T	Total elastomer thickness
$u(x)$	Horizontal displacement in x-direction of elastomer at a point
$u_o(x)$	Horizontal displacement in x-direction of elastomer at mid-depth of layer
$v(y)$	Horizontal displacement in y-direction of elastomer at a point
$v_o(y)$	Horizontal displacement in y-direction of elastomer at mid-depth of layer
W	Width of bearing (perpendicular to longitudinal axis of bridge)
x,y,z	Cartesian coordinates (parallel and perpendicular to longitudinal axis of the bridge and vertical, respectively)
α	Total rotation of one loaded bearing surface relative to the other
β	Coefficient (≥ 1.0) by which nominal shape factor must be divided to give effective shape factor
$\gamma_c, \gamma_s, \gamma_r$	Shear strain in elastomer caused by applied compression, shear or rotation, respectively
Δ_c	Compressive deflection
Δ_s	Shear deflection
Δ_{SL}, Δ_{SW}	Shear deflection in length and width directions respectively
ΔV	Change in volume of element of elastomer
∇^2	Laplacian operator $(\frac{\partial^2}{\partial x^2} + \frac{\partial^2}{\partial y^2})$ in cartesian coordinates
$\bar{\epsilon}_c$	Applied compressive strain (assumed uniform): $\bar{\epsilon}_c = \Delta_c/T$

Symbol	Description
η	Ratio $\dfrac{\text{allowable total shear strain}}{\text{elongation at break of material}}$ Used by British Specification BE 1/76
θ_n	Coefficient in series solution for stress caused by compression, including bulk compressibility
λ	Extension ratio: $\dfrac{\text{stretched length}}{\text{unstretched length}}$ (Tension causes $\lambda > 1$)
μ	Coefficient of friction
$\bar{\sigma}_c$	Average compressive stress
σ_{cr}	Critical compressive stress for instability of bearing
σ_s	Direct stress in reinforcement
$\sigma_{xx}, \sigma_{yy}, \sigma_{zz}$	Direct stresses in elastomer in x, y, and z directions
τ_c, τ_r, τ_s	Maximum shear stress in elastomer caused by applied compression, rotation, and shear respectively
τ_{max}	Maximum shear stress in elastomer
τ_{ij}	Stress on face i in direction j of an element of elastomer
τ_{zx}, τ_{zy}	Shear stress on z face in x or y direction of an element of elastomer
$\bar{\tau}_{zx}, \bar{\tau}_{zy}$	Shear stress on z face in x or y direction of an element of elastomer adjacent to reinforcement

APPENDIX B

GLOSSARY

AASHTO—American Association of State Highway and Transportation Officials.

AREA—American Railroad Engineers Association.

ASTM—American Society for Testing and Materials.

Accelerated testing—testing under conditions different from those to be expected in the field but chosen so as to reveal long-term effects in a shorter length of time. In rubber technology generally involves testing at elevated temperatures.

Aging resistance test—test (e.g., ASTM D573) to determine resistance of material to changes in physical properties caused by longterm exposure to the atmosphere.

Antioxidant—chemical(s) in compound to reduce changes in physical properties caused by chemical reaction with oxygen.

Antiozonant—chemical(s) in compound to reduce changes in physical properties caused by chemical reaction with ozone.

Apparent modulus—see *modulus*.

Aspect ratio—ratio of plan dimensions (rectangular bearings only).

Bond—joint between elastomer and reinforcement. Cold bonding (seldom used) is achieved at room temperature using adhesives. Hot bonding is effected during vulcanization (q.v.) generally aided by adhesives as well. Good bond is critical to the good functioning of a laminated bearing.

Bulk modulus—see *modulus*.

Butyl rubber—isobutylene polymer which displays high hysteresis and better low-temperature properties than most neoprenes and natural rubber compounds. However, it creeps more than either.

Chlorobutyl—chlorinated modification of basic Butyl rubber.

Chloroprene—monomeric material formed by the reaction of vinylacetylene with hydrogen chloride. Generally polymerized to form polychloroprene (neoprene).

Commercial bearing—bearing made by commercial manufacturer using standard production methods. To be distinguished from specimens made by special methods for laboratory testing.

Compound—substance whose constituent elements are chemically bonded together.

Compressive strain—compressive deflection of elastomer under load divided by its unloaded thickness.

Constitutive relation—relation between stress and strain.

Cover, side top and bottom—layer of rubber lying outside the reinforcement. Provides corrosion protection for reinforcement, and bond, and reduces stress concentrations in the rubber.

Cross-links—chemical bonds made during vulcanization between long chain molecules of the polymer at points where they cross. Increased cross-linking reduces creep.

Crystallization—phase change of elastomer during which segments of the long chain molecules gradually become oriented with reference to each other. The condition is promoted by low temperature in all elastomers, and by tensile strain in some. It manifests itself as an increase in hardness and stiffness. Strain crystallization is generally advantageous because it acts as an arrester at a crack tip. Low-temperature

crystallization is not, because it causes an increase in shear modulus, which in some neoprenes can be quite dramatic.
Delamination—separation of elastomer and reinforcement, generally starting at the edges. More prominent in steel-reinforced bearings with no edge cover.
Dowels—steel rods cast into concrete to prevent relative shear movement between elements. They may either be welded to the bearing sole plate and be cast into the pierhead, preventing slip of the bearing on its support, or they may be cast into both girder and support, preventing translation of the girder. In the latter case, they must either pass through holes in the bearing (common) or lie outside it.
Durometer—instrument for measuring hardness (see below).
Durometer hardness—the reading obtained from use of the instrument. See also *hardness*.
Dynamic loads—see *loads*.
EB—see *elongation at break*.
Effective area—area of rubber effective in resisting compression. Equal to plan area of reinforcement in a reinforced bearing, plan area in a plain pad. See also *reduced effective area*.
Elastically restrained structure—structure (e.g., bridge) which is not connected rigidly to its supports at any point, and in which all lateral forces are transmitted to the substructure through flexible bearings.
Elastomer—any member of a class of polymeric substances (i.e., those made of giant molecules consisting of linked smaller molecules or monomers) possessing rubberlike qualities, especially the ability to regain shape almost completely after large deformation. Generally applied only to vulcanized materials.
Elastomeric bearing—a device constructed wholly or partially from elastomer, the purpose of which is to transmit loads and accommodate movements between a bridge and its supporting structure.
Elongation at break—percentage elongation of material at break in standard tensile test. Defined as $\left(\frac{\text{stretched length}}{\text{unstretched length}} - 1\right) \times 100$. Rubber used in bridge bearings generally has a value between 200 and 800, but preferably it should be at least 400.
EPDM (Ethylene Propylene Dimonomer)—man-made rubberlike material which is a candidate for use in bridge bearings. It is believed to have better low temperature, but worse creep properties than most neoprenes.
Filler—material not possessing rubberlike properties included in elastomer compound. Carbon black is most commonly used and its addition tends to increase hardness, shear stiffness, and bulk modulus to increase energy dissipation during cyclic loading, to delay slightly the onset of low-temperature crystallization, and, because it is cheaper than the raw polymer, to reduce the price of the compounded elastomer.
Finite strain—material deformation which is large enough that the commonly used linearized equations for computing strain do not give acceptably accurate results.
Hardness—mechanical property of a material which describes its resistance to indentation of a standard device (i.e., a durometer). Measured in degrees on several scales (International Rubber Hardness, Shore "A", Shore "B", etc.), each of which is based on the use of a different shaped indentor. Measurements show considerable scatter, and numerical values on all scales are about the same for materials suitable for bridge bearings (i.e., 45–65).
Homogeneous—Having the same character and properties at all points throughout the body.
Hot molding—process using heat and pressure by which elastomeric materials may be vulcanized and shaped. Reinforced bearings are always so made, whereas plain pads may, as an alternative, be extruded.
Hydrostatic stress—direct stress having the same value in all directions.
Hypalon—chlorosulfonated polyethylene, manufactured by du Pont. Tried but found unsuitable for use in bridge bearings because it has a low elongation at break and poor resistance to abrasion and cyclic load.
IRHD—international rubber hardness degrees (see *hardness*).
Infinite strip—see *strip*.
Isotropic—Having the same properties in all directions.
Isolation device—device which prevents the influence of an action from reaching the isolated entity (e.g., rubber bearings may be used to isolate a building from ground-borne acoustic noise or from seismic disturbances).
Keeper blocks—small blocks placed in the edge of a mold for the purpose of holding the reinforcement in place during hot molding.
Laminate—layer of reinforcing material bonded to rubber in order to prevent its lateral expansion.
Laminated bearing—see *reinforced bearing*.
Linear elastic—descriptor of materials which completely recover their original shape when loads are removed and in which stress is a constant multiple of strain. (By nature of the definition it is applicable only to the range of infinitesimal strains.).
Loads—forces acting on a body. Dynamic loads vary with time, static loads do not.
Modulus (shear, Young's, bulk, apparent)—proportional constant relating stress to strain, strictly applicable only to linear elastic materials:

$$\text{shear modulus} = G = \frac{\tau}{\gamma} = \frac{\text{shear stress}}{\text{shear strain}}$$

$$\text{Young's modulus} = E = \frac{\sigma}{\epsilon} = \frac{\text{direct uniaxial stress}}{\text{direct uniaxial strain}}$$

$$\text{bulk modulus} = K = \frac{p}{\Delta V/V} = \frac{\text{hydrostatic stress}}{\text{change in volume/original volume}}$$

Apparent modulus (in compression, rotation, or shear) = that modulus which a single material would have to possess in order for a bearing made of it to display the same overall mechanical properties as would a reinforced elastomeric bearing of the same dimensions.
Mooney scorch time—time for cross-links to form in a rubber mix when held at a given temperature, detected by a specific change in Mooney viscosity. This latter is measured from the torque required to rotate a metal rotor embedded in the mix.
NCHRP—National Cooperative Highway Research Program. Program set up and administered by the National Academy of Sciences via the Transportation Research Board to initiate and supervise research in highway topics.

Natural rubber (polyisoprene)—polymer occurring naturally in the sap of certain plants, particularly Hevea brasiliensis.
Neoprene—any of a class of elastomers made by polymerization of the compound chloroprene. Notable properties are good resistance to abrasion, oxidation, and chemical attack. Neoprene (capitalized) has been used by the du Pont company for many years to describe a particular product of the company.
Nonlinear—a system in which the response is not a constant multiple of the excitation.
Ozone resistance test—standard test (e.g., ASTM D1149) to determine the ability of a material to retain its original properties when subjected to high concentrations of ozone for a given period of time.
P-Δ effect—influence of vertical load on the lateral stiffness of a structure, wherein the vertical load, P, multiplied by the lateral deflection of its point of application, Δ, causes additional overturning moments, and so, a tendency to increase lateral deflection.
PTFE—see *polychlorotrifluorethyline*.
Pad—see *plain pad*.
Plain pad—a block made entirely of elastomer used as a bearing and in which resistance to lateral expansion of the elastomer is provided solely by friction at the loaded surfaces.
Polychloroprene—polymerized form of the monomer chloroprene (see *chloroprene*).
Polychlorotrifluoroethyline—synthetic polymer with a very low friction coefficient and good chemical inertness and stability.
Polyisoprene—see *natural rubber*.
Polymer—a material consisting of long chain molecules built up from a number of smaller molecular units (monomers). They are characterized by very high molecular weights. Rubbers, both natural and synthetic, are a class of polymers which have the potential for large elastic deformations under load.
Polymerization—the process of forming long chain molecules by causing a number of smaller molecules or monomers to join together.
Pot bearing—bearing made from a block of elastomer, which is completely enclosed in a shallow cylinder and on which a piston rides. It can carry more compressive stress than an elastomeric bearing but it is more expensive, and on its own permits no shear deformation.
Protective system—chemicals added to an elastomeric compound to increase its resistance to ozone and chemical attack.
Raw rubber—rubberlike polymer forming the basis of an elastomeric compound.
Reduced effective area—area common to upper and lower compressed faces when a bearing is subjected simultaneously to compression and shear. Defined for a rectangular bearing as (width of reinforcement) × (length of reinforcement-shear displacement).
Reinforced bearing—a bearing composed of alternate layers of elastomer and reinforcing material, integrally bonded during vulcanization.
Reinforcement—layers of fabric or steel interleaved between elastomer layers and bonded to them, which locally prevent lateral expansion of the elastomer.
Rollover—local deformations other than simple shear which occur at corners of an elastomeric bearing subjected to shear deformation. More pronounced in bearings with flexible (or no) reinforcement and, in severe cases, may cause the edge of the bearing to lift off its seat.
Rubber—natural or synthetic organic polymer notable for its ability to undergo very large deformations and then to return to its original shape on removal of the load. The term is applied both to the raw polymer and to the compounded and vulcanized elastomer.
Rubber bearing—see *elastomeric bearing*.
Rubber technologist—person specializing in the development and manufacture of rubber and rubber products.
Seating pad—block of elastomer used in building construction to ensure even load distribution between beams and their supports and to permit modest relative movements in a horizontal plane.
Shape factor—dimensionless geometric factor defined as $\frac{\text{area of one loaded surface}}{\text{total area free to bulge}}$. It gives a good idea of the compressive stiffness and strength of an elastomeric bearing, regardless of the shape in plan.
Shear lag—phenomenon found in flexural members with wide flanges wherein direct stress due to flexure is not constant across a plane parallel to the neutral one, owing to the influence of shear deformation.
Shore hardness—see *hardness*.
Small deformations—deformations of a structure which are sufficiently small that for all practical purposes strains are linearly related to displacements.
Sole plate—heavy steel plate bonded to the upper or lower surface (or both) of an elastomeric bearing, for the purposes of simplifying connection of the bearing to other elements and ensuring an even distribution of load between them.
Static loads—see *loads*.
Strain crystallization—crystallization of a polymer under the influence of tensile strain.
Strip—rectangular block of material in which one plan dimension is much larger than the other.
Tear test—standard test (e.g., ASTM D624) used to determine the resistance of the material to tearing.
TFE—see *PTFE*.
Thixotropic—having properties which return to their original values when the material is altered by deformation and then left undisturbed for a period of time.
Transition temperature—temperature at which a significant change occurs in material properties.
UIC—Union Internationale des Chemins de Fer (International Railway Union).
Unreinforced pad—see *plain pad*.
Viscoelastic—possessing the property that resistance to deformation is dependent on the rate of deformation.
Vulcanization—process of inducing cross-links in a polymer by chemical reaction. Requires heat, pressure, and often a vulcanizing agent which varies with the polymer used.

APPENDIX C

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APPENDIX D

PROPOSED REVISED AASHTO SPECIFICATION—METHOD A

TABLE OF CONTENTS

CODE

1.12.1 General

An elastomeric bridge bearing is a device constructed partially or wholly from elastomer the purpose of which is to transmit loads and accommodate movements between a bridge and its supporting structure. This section of the specification covers the design of plain pads (consisting of elastomer only) and reinforced bearings (consisting of layers of elastomer restrained at their interfaces by integrally bonded steel or fabric reinforcement). Tapered elastomer layers are discouraged. In addition to any internal reinforcement, bearings may have external steel load plates bonded to the upper or lower elastomer layers or both. Such load plates shall be at least as large as the elastomer layer to which they are bonded.

The materials, fabrication, and installation of the bearings shall be in accordance with the requirements of section 2.25 of this specification.

This design procedure is based on service loads, excluding impact.

COMMENTARY

1.12.1 General

The specification does not cover types such as pot bearings or those made from a mixture of elastomer and closely spaced random fibers. Tapered layers are discouraged because too little is known about their behavior.

CODE

1.12.2 Design Procedure

1.12.2.1 Definitions

A = Plan area of bearing (in.^{2})
D = Gross diameter of circular bearing (in.)
F_s = Shear force on bearing (lb)
G = Shear modulus of elastomer (psi) at 73°F
L = Gross length of rectangular bearing parallel to the longitudinal axis of the bridge (in.)
P = Compressive load on bearing (lb)

S = Shape factor of one layer of a bearing

$$= \frac{\text{loaded area}}{\text{effective area free to bulge}}$$

$$= \frac{LW}{2t(L+W)} \text{ for rectangular bearings}$$

$$= \frac{D}{4t} \text{ for circular bearings}$$

T = Total elastomer thickness of bearing (in.) = Σt_i

t_i = Thickness of ith elastomer layer (in.)

W = Gross width of rectangular bearing perpendicular to longitudinal axis of the bridge (in.)

$\alpha_L,(\alpha_W)$ = Relative rotation of top and bottom surfaces of bearing about an axis perpendicular (parallel) to the longitudinal axis of the bridge (radians)

β = Modifying factor having a value of 1.0 for internal layers of reinforced bearings, 1.4 for cover layers, and 1.8 for plain pads

Δ_c = Instantaneous compressive deflection of bearing (in.)

Δ_s = Shear deflection of bearing (in.)

ϵ_{ci} = Compressive strain of ith elastomer layer (change in thickness divided by unstressed thickness)

σ_c = Average compressive stress on bearing caused by dead load and live load without impact

= P/A (psi)

COMMENTARY

1.12.2 Design Procedure

1.12.2.1 Definitions

A modifying factor, β, is introduced because the rubber in outside layers is restrained laterally by friction rather than bonding, and, except in bearings with impractically small shape factors, some slip occurs causing more vertical deflection and higher shear stresses in the rubber than would occur with bonded layers. The proposed provisions imply that a plain pad of shape factor S behaves like a reinforced bearing of shape factor $S/1.8$. The precise value of β depends on the contact surface between the rubber and load surface. The frictional resistance varies widely, but $\beta = 1.8$ is representative of practical plain pad applications.

In the past, vertical deflections have been estimated using design aids that have generally not distinguished between plain pads and reinforced bearings. The proposed provisions allow charts developed for reinforced bearings to be used for both provided the shape factor for plain pads is calculated using $\beta = 1.8$.

The allowable compressive stress (section 1.12.2.3) is now expressed as a single function of S/β for all bearings, but use of the appropriate value for β causes the allowable stress on a plain pad to be 56 percent of that of a reinforced bearing layer of the same dimensions.

Cover layers of reinforced bearings are treated similarly except that $\beta = 1.4$.

The increased computed deflection and the lower allowable stress on plain pads are supported by theory and tests, and have parallels in many foreign codes.

Holes are strongly discouraged in steel-reinforced bearings and are forbidden in fabric-reinforced bearings. However, if they exist, their effect should be accounted for when calculating the shape factor because they reduce the loaded area and increase the area free to bulge. Suitable formulas are:

$$S = \frac{LW - \frac{\pi}{4}\Sigma d_i^2}{t[2L + 2W + \pi \Sigma d_i]} \text{ for rectangular bearings}$$

$$S = \frac{D^2 - \Sigma d_i^2}{At[D + \Sigma d_i]} \text{ for circular bearings}$$

where d_i is the diameter of the ith hole in the bearing in inches.

CODE

1.12.2.2 Material Properties for Design

The properties of elastomeric compounds depend on their constituent elements. Where shear modulus or creep deflection properties are specified or known for the elastomer of which the bearings are to be made, they should be used in the design. Otherwise the values used shall be those from the applicable range given in Table 1.12.2.2A which provide the least favorable results. Values for intermediate hardness may be obtained by interpolation. The grade of the elastomer shall be selected on the basis of the requirements of section 2.25 and the environmental conditions. The shear modulus shall be determined using the test specified in section 2.25 of this specification. Material with a nominal hardness greater than 60 shall not be used in reinforced bearings.

Table 1.12.2.2A

HARDNESS (SHORE 'A')	50	60	70
Shear Modulus at 73°F (psi) (MP_a)	 85-110 (0.60-0.77)	 120-155 (0.85-1.10)	 160-260
$\left(\frac{\text{creep deflection}}{\text{instantaneous deflection}}\right)$ at 25 years	25%	35%	45%

COMMENTARY

1.12.2.2 Material Properties for Design

Materials with nominal hardness of greater than 60 are prohibited for laminated bearings because they generally have a smaller elongation at break, greater stiffness, shorter fatigue life, and greater creep than their softer counterparts. This inferior performance is generally attributed to the larger amounts of filler present. Plain pads up to 70 hardness are permitted because of their satisfactory use in past practice. However, evidence exists to suggest that the fatigue life of very hard pads may be considerably shorter than those made of softer compounds, and consequently their use is discouraged.

Shear modulus is the most important mechanical property for design and ideally it, rather than hardness, should be used to specify the material. However, hardness is retained for the time being because of its widespread use and ease of measurement.

It is important to use a value of G which reflects as closely as possible the material behavior, because it influences strongly the shear strain in the rubber caused by compression and the shear force caused by shear displacement, both of which are critical quantities. Thus, use of compounds with known properties is explicitly advocated. Where this is not possible the least favorable value should be selected from the applicable range in Table 1.12.2.2A, which covers the variations to be expected between different compounds of the same hardness. Thus, if a bearing is specified only as 50 durometer, and no limits are imposed on G, the compressive strength must be based on a G of 85 psi and the horizontal forces, on the low temperature properties of a material which has a G of 110 psi at 73°F. The resulting horizontal forces on the supports are larger than would be the case if the material properties had been well defined and may lead to extra expense in the substructure. Specifying the material by hardness rather than the more relevant shear modulus thus imposes a slight penalty.

Shear modulus increases as the elastomer cools but by an amount which varies very widely from one material to another. Information on specific compounds should be obtained from individual manufacturers. However, it should be noted that compounds of Neoprene generally stiffen more than those of Natural Rubber.

Creep varies from one compound to another and is generally more prevalent in harder elastomers, but is seldom a problem if high quality materials are used in reinforced bearings. The figures given are representative of Neoprene, but should be a safe estimate for Natural Rubber. The problem is much more serious for other elastomers such as Butyl. Field experience and tests suggest that plain pads creep more than reinforced bearings, and that variable (i.e., dynamic) compressive loads exacerbate the effect. The reasons are not clear, but they may be associated with increasing slip at the interface.

CODE

1.12.2.3 Compressive Stress

Unless shear deformation is prevented, the average compressive stress, σ_c, in any layer shall not exceed GS/β, nor shall it exceed 1,000 psi for steel reinforced bearings, or 800 psi for fabric reinforced bearings or plain pads. In bearings containing layers of different thickness, the value of S used shall be that for the thickest layer. Allowable compressive stress may be increased by 10 percent where shear translation is prevented.

COMMENTARY

1.12.2.3 Compressive Stress

The loads which can be safely placed on most structural elements are so defined as to prevent both instability and material distress to any of the materials of which they are made. The previous stress limits for elastomeric bearings did not do this, but the proposed ones attempt to. The provisions of this section address indirectly the question of distress in the elastomer, on the assumption that in a well-made bearing the elastomer will fail before the bond.

Elastomers are virtually indestructible when subjected to hydrostatic compression, and so it is the tensile and shear stresses and strains which are believed to be critical. These arise from imposed compression, shear deformation, and rotation.

Shear strains in the elastomer cause diagonal tension, although the large shear strains occurring in elastomeric bearings cause difficulties in computing the resulting tensile strain. However, this maximum tensile strain should clearly be limited to a proportion of the elongation at break of the elastomer, if tensile rupture is to be avoided.

European codes generally define shear strains from all causes and restrict their sum to a limiting allowable value. The proposed specification is less explicit, but, by limiting the average compressive stress to a proportion of GS and setting values for permissible simultaneous rotation and shear deformation, the maximum tensile strain is limited implicitly. As an example, a square laminated bearing with the maximum allowable compression, rotation, and shear would cause a peak tensile strain in the rubber of approximately 220 percent, or 63 percent of the minimum elongation at break specified by AASHTO for 60-durometer materials.

The European approach is not advocated here because knowledge is lacking in nonlinear mechanics (for adding large strains from different load cases) and in materials (for deciding what tensile strain limit is appropriate for fatigue loads). Without developments in these areas, design formulas of greater complexity than those offered here are not justified. Furthermore, many of the European codes which permit higher stresses do so contingent only on superior material properties, extremely stringent quality control, and extensive testing programs on actual bearings under static and dynamic loadings.

The upper limits of 1,000 psi for steel and 800 psi for fabric reinforced bearings are imposed because adequate test data (particularly in fatigue) are not available at higher stresses.

The 10 percent allowable stress increase for fixed or restrained bearings is based in principle on the total shear strain concept.

CODE

1.12.2.4 Compressive Deflection

Compressive deflection, Δ_c, of the bearing shall be so limited as to ensure the serviceability of the bridge.

Instantaneous deflection shall be calculated as

$$\Delta_c = \sum_i \epsilon_{ci} t_i$$

Values for ϵ_{ci} shall be obtained from design aids based on tests such as presented in Figures 1.12.2.4A and B, by testing, or by rational analysis.

The effects of creep of the elastomer shall be added to the instantaneous deflection when considering long-term deflections. They shall be computed from information relevant to

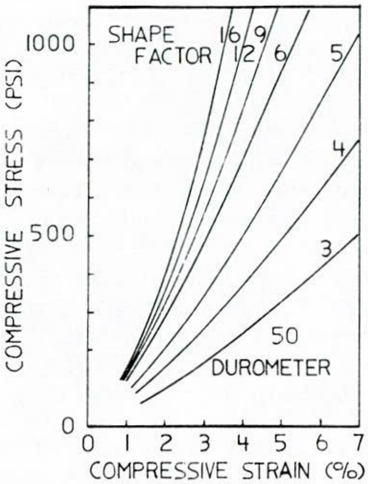

Figure 1.12.2.4A. Compressive stress–strain curves for 50-durometer elastomer (reproduced with the permission of the EI du Pont de Nemours Company).

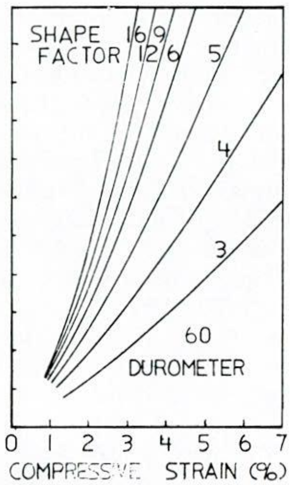

Figure 1.12.2.4B. Compressive stress–strain curves for 60-durometer elastomer (reproduced with the permission of the EI du Pont de Nemours Company).

the elastomer compound used if it is available. If not, the values given in section 1.12.2.2. shall be used as a guide.

COMMENTARY

1.12.2.4 Compressive Deflection

Limiting instantaneous deflections is important to ensure that deck joints and seals are not damaged. Long-term deflections should be accounted for as well when considering joints and seals between sections of the bridge which rest on bearings of different design, and when estimating redistribution of forces in continuous bridges caused by support settlement. Provided high quality materials are used, the effects of creep are unlikely to cause problems.

Possible equations for analytical computations of deflections may be found in Ref. (*1*). A maximum relative deflection across a joint of ⅛ in. is suggested. Joints and seals that are sensitive to relative deflections may require limits that are tighter than this.

CODE

1.12.2.5 Rotation

The relative rotation between top and bottom surfaces of the bearing shall be limited by

$$L\,\alpha_L + W\alpha_W \leq 2\Delta_c \text{ for rectangular bearings}$$

$$D\,\sqrt{\alpha_L^2 + \alpha_w^2} \leq 2\Delta_c \text{ for circular bearings}$$

COMMENTARY

1.12.2.5 Rotation

Rotation may be accommodated either by deformation of the bearing or by attachment of a rocking device.

The underside of the bearing must be set horizontal (except where pairs of inclined bearings are used) in order to prevent shear forces due to permanent loads. If, in the absence of a rocking device, the underside of the girder is not horizontal, its slope must either be included in the calculation of rotation or appropriate measures (such as tapered shim plates or a grout bed) must be used to correct it.

The proposed rotational limit prevents net upward displacement of any point on the bearing when it is subjected to simultaneous compression and rotation. This in turn prevents reversal of stress (from compression to tension) which is critical to the fatigue resistance of the elastomer.

The rotation should also be limited so that the combined effects of compression, shear and rotation cause neither excessive shear strain in the elastomer nor lift-off at one edge of the bearing. Exact criteria for these conditions are not currently available, but the existing evidence suggests that neither will occur if the proposed provisions are met.

CODE

1.12.2.6 Shear

The shear deformation shall be taken as the maximum possible deformation caused by creep, shrinkage, post-tensioning, and thermal effects computed between the installation temperature and the least favorable extreme temperature, unless a positive slip apparatus is installed.

The bearing shall be designed so that

$$T \geq 2\Delta_s$$

The shear force induced by shear deformation is approximated by

$$F_s = G\,\frac{A}{T}\,\Delta_s$$

Variations of G with temperature shall be taken into account.

COMMENTARY

1.12.2.6 Shear

Shear deformations in excess of 0.5T appear to cause roll-over at the bearing corners and, in steel reinforced bearings, the potential for delamination.

The build-up of shear force is calculated more realistically by an approach in which the increments of force in successive time steps are summed, each force increment being computed using the appropriate increment of shear deformation and the instantaneous value of G which depends on temperature. However, enough unknown influences exist (such as stress relaxation and rates of temperature change) to render such a method unreliable at present, so the simpler, more conservative method is retained.

Different elastomers display widely differing increases in shear modulus at low temperature. Temperature sensitivity depends strongly on many factors, such as raw polymer, compounding agent, vulcanization procedure, etc. As a result no useful range of stiffness can be given at low temperature and the use of the correct data is imperative. It is generally believed that natural rubber compounds are less influenced by low temperatures than polychloroprene.

Shear deformations in the elastomer caused by irreversible movements of the bridge girders may be relieved by temporarily lifting the girders and allowing the bearing to return to its undeformed shape.

CODE

1.12.2.7 Stability

To ensure stability, the total thickness of the bearing shall not exceed the smallest of:

L/5, W/5, or D/6 for plain pads
L/3, W/3, or D/4 for reinforced bearings

COMMENTARY

1.12.2.7 Stability

A reinforced bearing is relatively stiff in rotation and flexible in shear, causing its buckling load to be significantly lower than that computed using the bending stiffness alone. Surprisingly stocky bearings will fail by instability rather than material distress. A buckling theory exists, and, if used in conjunction with additional assumptions, slenderness limits can be developed below which the effects of instability may be neglected. Reasonable agreement is obtained with the geometric proportions which have traditionally provided satisfactory bearings.

The thickness limit for reinforced bearings has been changed from $\frac{W}{2}$ to $\frac{W}{3}$ to reflect the increase in allowable stress and the possibility of shear deformation in the lateral direction.

If, in the future, allowable compressive stresses are raised, problems of instability will require closer attention. A formula for calculating buckling stress explicitly may be included, but present analytical methods are unable to include the effects of bending in the reinforcement, and the only tests reported used very slender bearings and low stresses. Furthermore, "elastically restrained" bridges (in which no point is fixed rigidly against translation) are more susceptible to instability than are conventionally restrained ones. Thus work needs to be done before such provisions can be included.

CODE

1.12.2.8 Reinforcement

The reinforcement shall satisfy the requirements of section 2.26, and its resistance in pounds per linear inch at working stresses in each direction shall be not less than

$1400\ t_i$ for fabric
$1700\ t_i$ for steel

For these purposes, t_i shall be taken as the mean thickness of the two layers of elastomer bonded to the reinforcement if they are of different thickness.

The resistance per linear inch is given by the product of the material thickness and the allowable stress on the net section. The steel thickness should be appropriately increased if material is removed by cutting holes in it.

The determination of the material resistance at working stresses shall take into account the effects of fatigue loading and the stress concentrations caused by holes (if any) in the reinforcement. Holes are discouraged for steel-reinforced bearings and prohibited for fabric-reinforced bearings.

COMMENTARY

1.12.2.8 Reinforcement

The reinforcement must be adequate to sustain the tensile stresses induced by compression of the bearing, which increase with compressive load. With the present load limitations (1,100 psi maximum), the minimum steel plate thickness practical for fabrication will usually provide adequate strength.

For bearings reinforced with layers of fiberglass fabric in which alternate elastomer layers are ⅛ in. and ⅜ in. thick, the provisions require that fabric reinforcing must be able to resist a working load of 350 lb/in.

Holes in the reinforcement cause stress concentrations which have a harmful effect under fatigue conditions. It is not well understood, and so holes are discouraged. The required increase in steel strength to account for material removed when cutting holes is a separate issue from the stress concentration caused by the hole. Fiber reinforcement can carry no shear stress within its own plane, and so the stresses in the reinforcement cannot spread around a hole in the same way possible with a plate, and holes are thus not permitted. The allowable tensile stress in a steel plate with holes subjected to fatigue loading may be estimated by a rational method, or, as an alternative, by use of Category D in Table 1.7.2A1 of this specification.

CODE

1.12.2.9 Anchorage

If some combination of loads exists which causes a shear force greater than 1/5 of the simultaneously occurring compressive force, the bearing shall be secured against horizontal movement. If the bearing is attached to both top and bottom surfaces, the attachment should be such that no tension is possible in the vertical direction.

COMMENTARY

1.12.2.9 Anchorage

Friction coefficients between rubber and other materials appear to reduce with increasing normal stress. However, $\mu = 0.20$ appears to be a reasonable lower bound in the range of stresses of interest for contact with clean steel or concrete.

CODE

1.12.2.10 Stiffeners for Steel Girders

Steel girders seated on elastomeric bearings must have flanges that are stiff enough locally not to risk damage to the bearing. Any necessary stiffening may be accomplished by means of a sole plate or by vertical stiffeners welded to the girder web and flanges. The stiffening requirements of this section do not replace any others in this specification, but should be read in conjunction with them.

Single-webbed girders symmetric about their minor (vertical) axis and placed symmetrically on the bearing need no additional stiffening if

$$\frac{b_f}{2t_f} \leq \sqrt{\frac{F_{yg}}{3.4\sigma_c}}$$

where b_f = total flange width
t_f = flange thickness
F_{yg} = yield stress of girder steel

COMMENTARY

1.12.2.10 Stiffeners for Steel Girders

Local deformations of girder and bearing must be small enough to prevent overstress of the bearing, overstress of the girder flange and excessive deflection of the bridge deck. The first of these three governs with commonly used materials and allowable stresses.

A sole plate rather than stiffeners is recommended because it requires minimal welding and introduces the possibility of using a bearing wider than the girder flange.

Examples of girders calling for special analysis are box girders, or girders which are seated eccentrically on the bearing.

CODE

1.12.2.11 Installation

Misalignment in bridge girders due to fabrication tolerance, camber, or other source shall be considered in the bearing design or shall be accounted for with tapered sole plates or by a device which prevents eccentric loading on the bearing.

Bearings which are used in pairs shall be placed along an axis perpendicular to the longitudinal axis of a beam.

Bearings which are to be attached to a beam or the support pier shall use a positive attachment detail.

COMMENTARY

1.12.2.11 Installation

Inclined ends of the bridge girder due to camber or fabrication tolerances induce rotation in the bearing which causes large strains and must be considered in the design unless otherwise accounted for. Tapered steel plates or curved bearing surfaces as shown in Figures 1.12.2.11 A and B may be used to take up this rotation without inducing significant eccentricity on the bearing. Grouting may be used to provide a level bedding surface.

Attachment of the bearing is best performed with steel plates bonded to the top and bottom of the bearing. The plates should be bonded during vulcanization and may then be fixed to the pier or girder by bolts, keys or other mechanical means. Dowels may also be inserted through the bearing, but they require holes which reduce the strength of the bearing. Adhesive bonding is an undesirable method for attaching the bearing to the pier or girder, because cold bonding of rubber is not always reliable.

CODE

1.12.3 Alternate Design Procedures

The design of bearings by procedures other than those outlined above shall be permitted, at the discretion of the engineer. Such procedures shall take into account the stresses and deformation in the bearing determined from a rational analysis and the design shall be based on the material properties pertinent to the elastomer of which the bearing is to be made. Performance shall be verified by test using the standards of Level II certification given in section 2.25.7, and, in addition, the effects of instability and fatigue shall be investigated.

COMMENTARY

1.12.3 Alternate Design Methods

Methods used by codes in other countries fall into this category and are discussed in Ref. (1). However, it should be noted that there is wide disagreement in design procedures throughout the world and contradictions exist within the research literature. Thus, discretion is recommended. Furthermore, it should be noted that the quality control standards in other countries are more stringent than in the U.S.A., and the use of foreign specifications should be contingent on maintaining the appropriate standards during manufacture.

COMMENTARY

REFERENCES

1. J. F. Stanton and C. W. Roeder, "Elastomeric Bearings—Design, Construction, and Materials." *NCHRP Report 248* (August 1982).
2. Department of the Environmental (UK), Highways Directorate, "Design Requirements for Elastomeric Bridge Bearings." Technical Memorandum (Bridges) No. BE 1/76 (Feb. 13, 1976).
3. International Union of Railways, "Code for the Use of Rubber Bearings for Rail Bridges." UIC Code 772R. (Jan. 1, 1969).

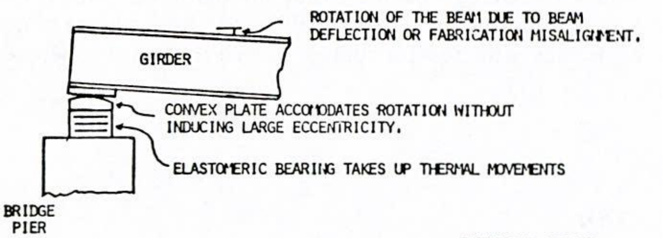

Figure 1.12.2.11A. Curved bearing surface.

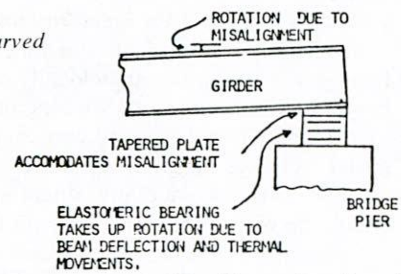

Figure 1.12.2.11B. Tapered external plate.

CODE

2.25 ELASTOMERIC BEARINGS

2.25.1 Scope

Elastomeric bearings as herein defined shall include unreinforced pads (consisting of elastomer only) and reinforced bearings with steel or fabric laminates.

COMMENTARY

2.25 ELASTOMERIC BEARINGS

2.25.1 Scope

This specification does not cover pot bearings, bearings from polymers other than virgin natural rubber or virgin polychloroprene, or mechanical bearings. AASHTO specification M251-74 does not apply to bearings designed in accordance with this specification, sections 1.12 and 2.25.

CODE

2.25.2 General Requirements

Bearings shall be furnished with the dimensions indicated. They shall be composed of the specified elastomer type, grade, and shear modulus (or hardness); shall be adequate for the specified design load; shall be tested at the appropriate level; and shall satisfy any special requirements of the designer. In the absence of more specific information, bearings shall be Grade 3, 50-durometer elastomer, adequate for 1,000 pounds per square inch Design Compression Stress, and be tested to Level I.

COMMENTARY

2.25.2 General Requirements

The designer and the contractor must provide sufficient information to permit manufacture and certification of the bearing. This requires additional information when a higher level certification is required. The design load is required because it is needed in some of the test procedures.

CODE

2.25.3 Materials

2.25.3.1 Properties of the Elastomer

The raw elastomer shall be either virgin polychloroprene (Neoprene) or virgin polyisoprene (Natural Rubber). The elastomer compound shall exhibit one of four grades of low temperature behavior. The four grades are as follows:

Grade 0—Suitable for use where the temperature never falls below 41°F (5°C) for more than 12 hours.

Grade 2—Suitable for use where the temperatures may frequently drop below 32°F (0°C) for periods less than 12 hours and may occasionally persist at that temperature for a day or two.

Grade 3—Suitable for use where the temperature may drop below 32°F (0°C) for up to 2 weeks.

Grade 5—Suitable for use where the low temperature may drop down to −40°F (−40°C) frequently for short durations and may remain below 5°F (−15°C) continuously up to 2 months.

An elastomer of higher grade number may be substituted for any lower grade.

The elastomer compound shall also meet the minimum requirements of Table 2.25.3.1A except as otherwise specified by the Engineer. Compounds of nominal hardness between 50 and 60 may be used and test requirements interpolated. If the material is specified by its shear modulus a consistent value of hardness shall also be supplied for the purpose of defining limits for the tests in Table 2.25.3.1A. If the hardness is specified, the shear modulus must fall within the range of Table 1.12.2.2A in Section 1.12.2.2. When test specimens are cut from the finished product, a 10 percent variation in physical properties shall be permitted. All material tests shall be carried out at 73°F ± 4°F (23°C ± 2°C) unless otherwise noted.

COMMENTARY

2.25.3 Materials

2.25.3.1 Properties of the Elastomer

At present only polyisoprene (Natural Rubber) and polychloroprene (Neoprene) are permitted. This is because both have an extensive history of satisfactory use. In addition, much more field experience exists with these two materials than with any other, and almost all of it is satisfactory.

The low-temperature grading system addresses the problem of stiffening of the elastomer at low temperatures. Special compounding is needed to avoid the problem but it increases cost and in extreme cases adversely affects some other properties. These adverse effects can be minimized by choosing that grade of rubber which corresponds to the conditions prevailing at the site. The grades follow the approach of ASTM D2000 and D4014 and require more stringent low temperature testing for higher grades.

Table 2.25.3.1A outlines the minimum properties of the elastomer. The standards are sometimes different for polychloroprene and natural rubber, which appears irrational because in other ways the requirements resemble a performance specification. However, the present state of knowledge is inadequate to pin down precisely those material properties needed to assure good bearing behavior, so the tests are intended to achieve a generally good quality material rather than specific properties. Natural rubber and chloroprene have different strengths and weaknesses, so different tests are indeed appropriate. (Natural Rubber generally creeps less, suffers less low-temperature stiffening, and has a better elongation at break—but polychloroprene has better chemical, ozone, and aging resistance.)

Harder elastomers have a greater shear stiffness and thus exert much larger pier forces due to thermal expansion than materials of low hardness, unless the plan area of the bearing is reduced proportionately. This causes the bearing to be rather high with respect to its lateral dimensions, possibly leading to instability problems, and so generally is impractical. Further, it is generally believed that 70-durometer material has a considerably shorter fatigue life and greater creep than its softer counterparts. Thus, when larger compressive stiffness is required, it is recommended that reinforced bearings of softer elastomer with thinner layers and higher shape factor be used. Hardness is maintained as a material property because it is widely used in rubber technology and is easy to measure. However, measurements are sensitive to who takes them, and hardness generally gives only a rough indication of mechanical properties. The shear modulus is a much better indication, but is more difficult to measure.

CODE

2.25.3.2 Steel Laminates

Steel laminates used for reinforcement shall be made from rolled mild steel conforming to ASTM A36, A570, or equivalent, unless otherwise specified by the Engineer. The laminates shall have a minimum nominal thickness of 16 gage. Holes in plates for manufacturing purposes shall be considered in design.

COMMENTARY

2.25.3.2 Steel Laminates

It is intended that a mild steel of a well-defined ASTM Standard be used. However, no single ASTM grade is always suitable because A36 steel is not rolled to gage thicknesses which are frequently used, and other grades are not rolled to 1/16 in., 1/8 in., etc., thickness, which may also be used. The minimum thickness is intended to ensure that the steel will not deform excessively during sandblasting or molding of the bearing. For large bearings, thicknesses greater than 16 gage may be needed, and individual manufacturers should be consulted. In most cases, satisfaction of this requirement will ensure satisfaction of section 1.12.2.8.

CODE

2.25.3.3 Fabric Reinforcement

Fabric reinforcement shall be woven from 100 percent glass fibers of "E" type yarn with continuous fibers. The minimum thread count in either direction shall be 25 threads per inch (10 threads per cm). The fabric shall have either a crowfoot or an 8 Harness Satin weave. Each ply of fabric

Table 2.25.3.1A

MATERIAL PROPERTY	ASTM STANDARD	TEST REQUIREMENTS	NATURAL RUBBER 50 *Duro*	 60 *Duro*	 70 *Duro*	POLYCHLOROPRENE 50 *Duro*	 60 *Duro*	 70 *Duro*	UNITS
Physical Properties	D2240	Hardness	50 ± 5	60 ± 5	70 ± 5	50 ± 5	60 ± 5	70 ± 5	Shore A Points
	D412	Min. tensile strength	2250 (15.5)	2250 (15.5)	2250 (15.5)	2250 (15.5)	2250 (15.5)	2250 (15.5)	psi MPa
		Min. ultimate elongation	450	400	300	400	350	300	%
Heat Resistance	D573 at specified temp.	Specified temperature of the test (22 hrs)	158 (70)	158 (70)	158 (70)	212 (100)	212 (100)	212 (100)	°F °C
		Aging time	168	168	168	70	70	70	Hours
		Max change in durometer hardness	+10	+10	+10	+15	+15	+15	Shore A Points
		Max change in tensile strength	−25	−25	−25	−15	−15	−15	%
		Max change in ultimate elongation	−25	−25	−25	−40	−40	−40	%
Compression Set	D395 Method B at specified temp.	Specified temperature of test degrees	158 (70)	158 (70)	158 (70)	212 (100)	212 (100)	212 (100)	°F °C
		Max permissible set	25	25	25	35	35	35	%
Ozone Resistance	D1149	Partial pressure of ozone during test	50	50	50	50	50	50	mPa
		Duration of test	100	100	100	100	100	100	Hours
		Tested at 20% strain 100°F ± 2°F (37.7°C ± 1°C) mounting procedure D518, Procedure A	no cracks	no cracks	no cracks	no cracks	no cracks	no cracks	
Low Temperature Properties	D2137 Method A	Low temperature Brittleness Test (required for Grade 3 and 5 only) at −13°F (−25°C) for Grade 3 and −40°F (−40°C) for Grade 5	no failure	no failure	no failure	no failure	no failure	no failure	
	D1415 or D2240	Low temperature stiffness (required for Grades 2, 3 and 5 only). Conditioned for 22 hours at 14°F (−10°C) for Grade 2, −13°F (−25°C) for Grade 3 and −40°F (−40°C) for Grade 5. Max change in hardness	+15	+15	+15	+15	+15	+15	Shore A Points
	D1229	Max low temperature compression set (required for Grades 2, 3 and 5 only) when tested at 25% compression for: 22 hours at 32°F (0°C) for Grade 2, 7 days at 14°F (−10°C) for Grade 3, 14 days at each of 14°F (−10°C) and −13°F (−25°C) for Grade 5.	65	65	65	65	65	65	%

shall have a minimum breaking strength of 800 lb/in. (140 KM/m) of width in each thread direction.

COMMENTARY

2.25.3.3 Fabric Reinforcement

Fiberglass is the only fabric proven to perform adequately as reinforcement, and only one grade is currently permitted. Polyester has proved too flexible, and both it and cotton are not strong enough. High performance fabrics, such as Kevlar, polyethylene, may be suitable for use in the future. The strength of the reinforcement governs the compressive strength of the bearing when minimum amounts are used, so if stronger fabric with acceptable bond properties is developed, the stress limits of section 1.12.2.8 may be reconsidered. However, thorough testing over a wide range of loading conditions will be needed prior to acceptance.

CODE

2.25.3.4 Bonding Adhesive

The vulcanized bond between fabric reinforcement shall have a minimum peel strength of 30 lb/in. (5.2KN/m). Steel laminated bearings shall develop a minimum peel strength of 40 lb/in. (6.9 KN/m). Peel strength tests shall be performed by ASTM D429 Method B.

COMMENTARY

2.25.3.4 Bonding Adhesive

Adequate bond is essential if the reinforcement is to be effective. It is particularly important at the edges of the bearing.

CODE

2.25.4 Fabrication

Bearings with steel laminates shall be cast as a unit in a mold and bonded and vulcanized under heat and pressure. The molds shall have standard shop practice mold finish. The internal steel laminates shall be sandblasted and cleaned of all surface coating rust and mill scale before bonding, shall be free of sharp edges and burrs, and shall have a minimum edge cover of ⅛ in. (3 mm). External load plates (sole plates) shall be protected from rusting by the manufacturer, and preferably shall be hot bonded to the bearing during vulcanization. Bearings which are designed to act as a single unit with a given shape factor must be manufactured as a single unit.

Fabric-reinforced bearings may be molded and vulcanized in large sheets and cut to size. Cutting shall be performed so as to avoid heating the materials and produce a smooth finish with no separation of the fabric from the elastomer. Fabric reinforcement shall be single ply at the top and bottom of the reinforced bearings and double ply for internal reinforcement layers. Fabric shall be free of folds and ripples and shall be parallel to the top and bottom surfaces. Plain pads may be molded or extruded, and vulcanized in large sheets and cut to size. Cutting shall not heat the materials, and shall produce a smooth finish to ANSI 250.

Flash tolerance, finish, and appearance shall meet the requirements of the latest edition of the Rubber Handbook as published by the Rubber Manufacturers Association, Inc., RMA F3 and T.063 for molded bearings and RMA F2 for extruded bearings.

COMMENTARY

2.25.4 Fabrication

Bearings which are designed as a single unit must be built as a single unit, because the shape factor, bearing stiffness, and general behavior will be different if built in sections.

Sandblasting and thorough cleaning of steel laminates are important to assure good bond. Edge cover is primarily needed to prevent corrosion of the reinforcement and ozone attack of the bond. However, it has been suggested that the cover decreases the probability of delamination or fatigue problems by reducing the stress concentrations at the exposed outer surface.

Bonding during vulcanization has been more successful in the past in producing good bonds, and so is required for bonding of internal laminates. However, practical difficulties may arise in hot bonding of external plates, thus hot bonding is strongly recommended for them, but not required.

CODE

2.25.5 Tolerances

Plain pads and laminated bearings shall be built to the design dimensions and these specifications with the following tolerances:

	(in.)	(mm)
1. *Overall Vertical Dimensions*		
Design Thickness 1-¼" (32 mm) or less	−0, +⅛"	(−0, +3)
Design Thickness over 1-¼" (32 mm)	−0, +¼"	(−0, +6)
2. *Overall Horizontal Dimensions*		
36" and less (.914m)	−0, +¼"	(−0, +6)
over 36" (.914m)	−0, +½"	(−0, +12)
3. *Thickness of Individual Layers of Elastomer* (Laminated Bearings Only) at any point within the bearing	±20% of design value but no more than ±⅛" (± 3 mm)	
4. *Variation from a Plane Parallel to the Theoretical Surface* (as determined by measurements at the edge of the bearings)		
Top	slope relative to the bottom of no more than .005 radians	
Sides	¼"	(6 mm)
5. *Position of Exposed Connection Members*	⅛"	(3 mm)
6. *Edge Cover of Embedded Laminates or Connection Members*	−0, +⅛"	(−0, +3 mm)
7. *Size of Holes, Slots, or Inserts*	±⅛"	(3 mm)
8. *Position of Holes, Slots, or Inserts*	±⅛"	(3 mm)

COMMENTARY
2.25.5 Tolerances

Some of the tolerances have been changed to relative values, because a specific tolerance such as 1/16 in. may be overly large for a small bearing and unrealistically small for a large bearing.

CODE
2.25.6 Marking and Certification

The manufacturer shall certify that each bearing satisfies the design specification, and supply a certified copy of test results. Each reinforced bearing shall be marked in indelible ink or flexible paint. The marking shall consist of the order number, lot number, bearing identification number, and elastomer type and grade number. Unless otherwise specified in the contract documents, the marking shall be on a face which is visible after erection of the bridge.

COMMENTARY
2.25.6 Marking and Certification

In the short-term, marking simplifies the identification of the correct bearings at the job site. In the long-term, it may permit the removal of bearings after a number of years of service to check the change in material properties over time. It also helps in settling disputes.

CODE
2.25.7 Bearing Tests and Acceptance Criteria

The acceptance criteria shall be at 2 levels. Level I acceptance shall be applied to all bearings. Level II acceptance criteria shall, at the discretion of the Engineer, be applied to more critical or unusual bearings. It shall also be used to resolve differences over the acceptance of bearings to which only Level I tests shall have been applied.

Level I criteria require that the bearing be manufactured according to this specification and any additional requirements specified by the Engineer. The manufacturer shall proof load each steel reinforced bearing with a compressive load 1.5 times the maximum design load. If bulging patterns imply laminate placement which does not satisfy design criteria and manufacturing tolerances or if bulging suggests poor laminate bond, the bearing shall be rejected. If there are 3 separate surface cracks which are greater than 0.08-inch (2 mm) wide and 0.08-inch (2 mm) deep, the bearings shall be rejected.

Level I criteria require that the elastomer satisfy the minimum properties of Table 2.25.3.1A, except as otherwise specified by the Engineer. Tensile strength elongation at break, durometer hardness, bond strength, and ozone resistance shall be tested for each production lot of bearings. Tensile strength of fabric-reinforcement shall be tested for each lot of fabric. Other material tests shall be performed whenever there is a change in the type or source of raw materials, elastomer formulation or production procedures, or as required by the Engineer.

Level II certification requires that all Level I conditions are satisfied, except that individual conditions may be waived by the Engineer if Level II certification is used as an arbitration of disputes. As a minimum, shear modulus and compressive stiffness shall be determined in accordance with ASTM D4014. The shear modulus may be determined by testing a piece of the finished bearing as specified on D4014, or at the discretion of the Engineer, a comparable non-destructive test may be performed on the complete bearing. Compressive stiffness tests shall be performed on the complete bearing. The shear modulus shall fall within ±15 percent of the value specified in the design document or within the limits of section 1.12.2.2 if no value for shear stiffness is specified. The compressive stiffness shall vary by no more than ±10 percent from the median value of all bearings, nor ±20 percent from the design value, if specified. However, a compressive stiffness and a shear stiffness shall not both be specified for the same bearing.

COMMENTARY
2.25.7 Bearing Tests and Acceptance Criteria

Two levels of acceptance criteria are proposed. Level I consists of a series of material certification tests plus assurances that the bearing has been manufactured to design specifications and tolerances. Each steel reinforced bearing must be proof loaded as an additional check of the manufacturing process. Poor laminate bond or misplaced laminates will show up in the bulging pattern and thus these bearings should be rejected. It should be noted that the proofload test may result in rejection of a small number of bearings which are in fact acceptable, but this is believed to be a small price to assure the elimination of major fabrication deficiences. It is particularly important because only hardness and external dimensions can be checked with any ease once the bearing has been delivered.

Level I certification is simple and inexpensive, but it is not comprehensive enough to guarantee good bearing behavior. Good behavior depends on the strength and stiffness of the bearing. Level I testing focuses on elastomer hardness, material properties, and a proof load to check fabrication. Hardness is only a very approximate measure of bearing shear stiffness, and the value varies from one operator to another. However, in view of the satisfactory performance of most bearings designed by existing procedures with existing quality control, Level I certification is believed to be adequate for most bearings at present stress levels. Level II certification is proposed for more critical situations. It will focus on actual bearing behavior in addition to Level I conditions. Shear and compressive stiffness tests will be performed on selected bearings to assure that they fall within accepted limits, and additional testing such as fatigue or tests to failure may be required at the discretion of the engineer. Level II testing will be moderately expensive and time consuming and thus is intended for very important bearings in critical structures. It is also intended to be a possible method for resolving differences which arise in Level I certification. For example, if a dispute arises over the true hardness value of a material, Level II techniques (such as a shear test) can be used to determine its acceptability.

Note the shear modulus determined by Level II certification is not a true material property, but is adequate for practical purposes. A minimum specimen size of 3 in. × 3 in. × ¼ in. is recommended for the shear test.

CODE

2.25.8 Installation

Bearings shall be placed on level surfaces unless they are placed in opposing pairs. Any misalignment in the support shall be corrected to form a level surface. Welding of steel girders or base plates to the exterior plates of the bearing is not permitted unless there is more than 1½ inches of steel between the weld and elastomer. In no case shall the elastomer or elastomer bond be exposed to instantaneous temperatures greater than 400°F (204°C).

COMMENTARY

2.25.8 Installation

Bearings must be placed on a level surface because gravity loads are resisted as a shear force in the bearing when they are inclined. In addition, inclined bedding surfaces increase the likelihood of the bearing walking out from the load. Thus, inclined bedding surfaces are prohibited unless the bearings are placed in opposing pairs as is recommended, for example, for seismic design. Welding of steel sole plates of the bearing to base plates or steel girders may also be used, but high local temperatures during welding will break down the bond between the elastomer and laminate, so great care is needed. A few seconds at 400°F or greater may damage the bearing. Longer periods at lower temperatures may also prove critical.

APPENDIX E

OUTLINE OF PROPOSED REVISED AASHTO SPECIFICATION—METHOD B

This appendix contains a skeleton on which can be built a design method more sophisticated than Method A. It is tentative and must be regarded as subject to extensive change. For easy presentation it has been organized in a manner similar to Method A and in some cases its provisions are likely to be the same. In other cases the provisions differ, and are given only in principle, with numerical values held in abeyance until the research necessary to verify them can be done. Symbols (e.g., C_1, C_2, etc) are given in their stead. In areas of greater uncertainty, even the principles expressed are tentative.

1.12 DESIGN

1.12.1 General

As method A except:

- Only for reinforced bearings.

1.12.2 Design

1.12.2.1 Definitions

As method A with possible changes:

- Dimensions used for defining the shape factor to be between the overall ones and those of the reinforcement.
- Thickness modification factor, β, to be revised, possibly to become a function of shape factor.

1.12.2.2 Material Properties for Design

- Material properties to be defined in terms of shear modulus.
- Hardness values alone not acceptable.

1.12.2.3 Compressive Stress

- Limit compressive stress $\sigma_c \leq C_1$ GS.
- Limit total shear strain $\gamma_c + \gamma_r + \gamma_s \leq C_2$ (C_2 may be a constant or a function of rubber properties).
- Possibly use different limits for dead (static) and live (dynamic) components in the above.

1.12.2.4 Compressive Deflections

Existing method A approach is very approximate and highly variable, and extension and experimental verification of existing theories may be warranted.

1.12.2.5 Rotation

- Limit stress on tensile side of bearing

$$L\alpha_L + W\alpha_L < C_3\Delta_c$$

- Prevent lift-off at lightly loaded edge.
- Prevent compressive overstress (see 1.12.2.3) by

$\gamma_c + \gamma_r + \gamma_s \leq C_2$

- Possibly use different limits for dead (static) and live (dynamic) components in the above.

1.12.2.6 Shear

- Limit shear strain alone by $\Delta_s \leq C_4T$ (C_4 may depend on bearing properties such as cover thickness, rigidity of reinforcement, etc., which influence rollover, or it may be a constant as defined in Method A).
- Limit total shear strain (see 1.12.2.3)

  $\gamma_c + \gamma_r + \gamma_s \leq C_2$
- Prevent problems in slender bearings by

  $\Delta_{SL} < C_5L$

  $\Delta_{SW} < C_5W$
- Include in shear deformation the translation of girder end rotation multiplied by depth to neutral axis.
- Account for variations in G with temperature in shear force calculation, e.g.

  $F_s = \max |F_{sr}|$

where

$$F_{sr} = \sum_{i=1}^{r} G_i \left(\frac{A}{T}\right) \Delta_{si}$$

and

G_i = shear modulus during ith increment of shear deformation.

Δ_{si} = amplitude of ith increment of shear deformation.

1.12.2.7 Stability

- Limit allowable stress to a safe fraction of the elastic buckling stress: i.e., $\sigma_c <$ function of geometry and material properties.

  The function is likely to have the form proposed by Haryngx and Gent, but may need modification to account for different types of top and bottom restraint, such as freedom to translate (e.g., in an "elastically restrained structure") or freedom to rotate (e.g., when a rocker plate is placed on top of the bearing). Bending flexibility of the reinforcement may also need to be considered.
- As a safe but more restrictive alternative:

  $T < C_6L$

  $< C_7W$ similar to method A

  $< C_8D$

1.12.2.8 Reinforcement

As method A with possible changes:

- Required strength to be a function of plan geometry.
- Separate limits for static and dynamic components of load.

1.12.2.9 Anchorage

As method A with possible changes:

- Friction coefficient to vary with compressive stress and, perhaps, other parameters such as contact surface.
- Friction coefficient may be a function of the ratio

$$\frac{\text{dynamic compression}}{\text{static compression}}$$

1.12.2.10 Stiffeners for Steel Girders

As method A.

1.12.2.11 Alternate Design Procedures

Since the preceding sections of 1.12.2 would represent the state of the art, there seems little cause for including a clause permitting design by "other rational analysis." Possibly "design by test" could be permitted, but since the bearings are likely to be big and fatigue tests would be essential, the provision would be seldom used because of the practical difficulties.

1.12.2.12 Other Provisions

Possibly provide guidance on:

- Bearings inclined in pairs
- Bearings subject to uplift

2.25 CONSTRUCTION

2.25.1 General

- Reinforced bearings only.

2.25.2 Basis of Purchase

As method A.

2.25.3 Materials

2.25.3.1 Elastomers

- Possibility of using elastomers other than NR and CR.
- Properties to be defined by shear modulus not hardness.
- Temperature grade system to be maintained but refined.

2.25.3.2 Steel

- Minimum thickness to be related to bearing dimensions.

2.25.3.3. Fabric

- Other materials and weaves to be considered.
- Performance tests to be defined.

2.25.3.4 Bonding Adhesive

As method A.

2.25.4 Fabrication

As method A.

2.25.5 Tolerances

- To be reviewed for large bearing sizes.

2.25.6 Marking and Certification

As method A.

2.25.7 Bearing Tests and Acceptance Criteria

- Level II tests to be refined in the light of practical experience.

2.25.8 Installation

- Possibly provide guidance on efficient positive attachment details.

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